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Wrocław University of Technology

Automotive Engineering

Piotr A. Wrzecioniarz, Wojciech Ambroszko,
Aleksander Górniak

ENERGY EFFICIENT DESIGN OF POWERTRAIN AND BODY

Wrocław 2011

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Introduction

Over 100 years the motor car industry is developing all over the world. Millions of vehicles are produced every year. Hundreds of millions are used, serviced and at last recycled. Hundreds of millions of human beings exploit all kinds of vehicles: on-road and off-road, cars and utility vehicles. People who are working in oil, chemical, mechanical and electronic industry producing parts, elements, systems and all kinds of fluids needed for exploitation, experts from financial, business, marketing, health, insurances and other branches are busy all the time fulfilling requirements coming from the market.

For running this big business all the time in whole world a lot of people are trained. Understanding of specific phenomena connected with this branch of industry overspreads on the earth. This growing industry also needs more and more engineers familiar with vehicle problems. Vehicle engineering courses on different levels appeared in many universities. Globalization also in this field of interest has influence for exchange of ideas and methods of training.

This text book for students also will be used for this purpose. In was quite difficult to prepare next one because in mentioned 100 years appeared thousands of articles, reports, publications as well as hundreds books, encyclopaedic definitions, university publications. Additionally in the last years, at the beginning of XXI century also thousands of pages of text appeared in the internet. Also in past decades in divided world there were two or even more authors of the same ideas and drawings. It is very difficult to find now the real author of the text, because every new information immediately are repeated many times. Today in open exchange of information this problem arises more intensively. We must also remember that at every university available material is selected looking for own historical requirements of local industry, expectations of the job market, future trends and international collaboration as well as the idea of leading person. Also program proposed in this text book is taking into account those requirements as well as EU expectations, because of the financial support of our work.

This text is also accompanied by additional over 582 slides prepared for lectures. So this text and those presentations are the full set of the course presented for Vehicle Engineering at the Wrocław University of Technology for polish and overseas students. Specific subject “Energy Efficient Design of Powertrain and Body” is coming from long term specialization of WUT.

So in the first chapter vehicle dynamics is described. Vehicle dynamics is a branch of engineering which covers all kinds of accelerations caused by all kinds of vehicle movement. These accelerations which are a result of forces actuated on a vehicle are affected by solutions used in brakes system, suspension system, steering system, aerodynamic package, etc.

In the chapter 2 power train system is briefly described. The main elements of the system are mentioned. Fundamental information about friction clutches as well as torque converters are presented. Different types of gearboxes from classical up to continuous variable are described. Short sub-chapter presents propeller shafts and produced contemporary universal joints. Different types of differential ends the chapter.

Chapter 3 covers steering system design. Forces acting on the steered wheel are described. Steering system layouts and elements used in contemporary cars as well as power steering systems are presented.

Brake systems are presented in chapter 4. Starting from brake types, through brake system elements and power brakes solutions up to brake proportioning. Safety systems connected with brakes are also briefly described.

Chapter 5 is covering suspension systems. Firstly suspension geometries are presented, afterwards wheel alignment angles are described and shown on examples, next the pitch and roll centre is explained and designated for some suspension systems. Suspension variables wheel movement depended are presented and the relation is explained. This big chapter is finished with anti-features explanation and presentation of all springs and dampers used in production cars.

In the chapter 6 the overall road load impact on fuel economy is described.

Chapter number 7 shows different applications of CAD/CAM/CAE in automotive industry. Nowadays there is no area of interest in industry without computer aided solutions. This chapter only shows some possibilities existing, thanks to development in last decades.

Noise, vibration and harshness problems are briefly mentioned in chapter nr 8. Practical solutions of reduction of those disadvantages connected with vehicle exploitation are described.

Different materials used in automotive industry as well as tendencies connected with replacing classical materials by new ones are discussed in chapter nr 9. Mass reduction, fuel consumption reduction, reduction of production and recycling costs require still a lot of work and is a great challenge for future employee in research, development and competence centres.

Chapter nr 10 is connected with fundamental information about vehicle usage, servicing and End of Life Vehicle.

The whole text is ended by chapter 11 in which short description of future trends in automotive industry is presented

We know that we did not covered all areas connected with automotive industry. Full description of all mentioned aspects would take more than thousand pages. So at the end of the text big bibliography is given. A lot of information in internet helps all students to find additional data needed at university and later on in engineering practice.

This text was prepared on the basis of over 30 year of experience in teaching different problems connected with vehicle engineering in WUT, but also on our collaboration with universities in Germany, Italy, Great Britain, Greece, Spain, Portugal, in post-soviet countries and also on the basis of permanent collaboration with world brands located near Wrocław collaborated with us e.g. Volvo, Wabco, Mercedes, Rehau, Continental, Bosch, GKN, TÜV, TÜVPOL.

We would like to thank also our students who heard all our lectures in polish language in last 20 years and overseas students from all European countries as well as from China, South Korea, Brazil and other countries who participated in lectures, seminars and laboratories, who added their culture, knowledge and enthusiasm to international exchange of knowledge. Thank you very much for Prof. Wiesław Fiebieg from WUT , who is also leading his company "Wibroakustyka", who prepared for us some details connected with noise, vibrations and hashness problems.

We hope that this introductional information in written text and in presentations prepared for lectures will allow to come to fascinating area of Vehicle Engineering and that students will apply this knowledge in real life practice.

We would like also to invite future students to work together with us for permanent improvement of our job. We hope that younger colleagues will continue the efforts also in the future, collaborating with everybody all around the world.

1. VEHICLE DYNAMICS

Vehicle dynamics is a branch of engineering which covers all kinds of accelerations caused by all kinds of vehicle movement. These accelerations which are a result of forces actuated on a vehicle are affected by solutions used in brakes system, suspension system, steering system, aerodynamic package, etc.

Before describing dynamic problems, the vehicle statics should be considered.

1.1 Vehicle statics

The dynamics of the car is strongly affected by the Centre of Gravity (CoG) position. Both equations for longitudinal and lateral weight transfer include CoG height. Therefore it is vital to find the position the centre of mass of a vehicle to calculate the dynamic forces.

$$\Delta W_{Lat} = \frac{W \cdot A_{lat} \cdot h_{CoG}}{t} \tag{1.1}$$

$$\Delta W_{Long} = \frac{W \cdot A_{long} \cdot h_{CoG}}{l} \tag{1.2}$$

ΔW_{Long}- longitudinal weight transfer

ΔW_{Lat}- lateral weight transfer

W – load

A_{long} - longitudinal acceleration / deceleration

A_{lat} - lateral acceleration

h_{CoG} - Center of Gravity height

t – track

l - wheelbase

Practically in laboratory or workshop to determine CoG height, the vehicle front or rear needs to be jacked. When reaching specific angle, the differences in wheel loads and simple geometric calculations will determine this value. There are several rules that need to be obeyed.

- The vehicle should be standing on 4 corner weights
- The suspension needs to locked (suspension movement would affect the calculated value)
- Front/rear of the car should be lifted
- The wheels that are standing on the scales need to be secured from rolling off the scales
- Any load that could move must be secured
- Fuel tank full or empty
- Driver inside the car or not

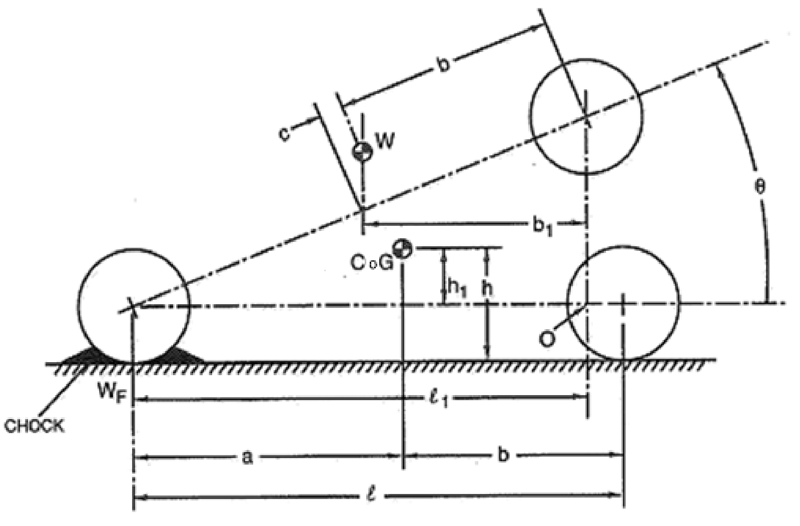

Figure 1.1. Main relations for determining CoG [8]

Data required to calculate the CoG height:

W - weight of the vehicle

W_F - weight on front wheels with rear elevated

b – horizontal distance from rear axle to CoG

l – wheelbase

R_{LF} - front axle height above ground

R_{LR}- rear axle height above ground

$tan\theta, cos\theta$ – tangent and cosine of the angle to which the rear is elevated

From the geometry l_1 can be calculated

$$l_1 = l \cdot cos\theta \tag{1.3}$$

From the moment about point O

$$W_F \cdot l_1 = W \cdot b_1 \tag{1.4}$$

$$b_1 = \left(\frac{W_F}{W}\right) \cdot l \cdot cos\theta \tag{1.5}$$

$$\frac{b_1}{b+c} = cos\theta \tag{1.6}$$

$$c = \left(\frac{W_F}{W} \cdot l\right) - b \tag{1.7}$$

The distance h_1 above line connecting wheels centres can be calculated from:

$$h_1 = \frac{W_F \cdot l - W \cdot b}{W \cdot tan\theta} \tag{1.8}$$

If the wheels used have the same diameter – the axle height for the rear and front may be different. For the same diameters the CoG height can be easily calculated.

$$h = R_L + h_1 \tag{1.9}$$

In case of different wheels sizes – for the front and rear of the car – the equation is a little more complicated.

$$R_{L\,CG} = R_{LF}\left(\frac{b}{l}\right) + R_{LR}\left(\frac{a}{l}\right) \tag{1.10}$$

$$h = R_{L\,CG} + h_1 \tag{1.11}$$

After calculating CoG height all dynamic forces calculations can be made.

1.2 Coordinates system

Before going into specific solutions a whole systems, main principles and occurrences need to be explained.

Main coordinate system defines vehicle movement. It has to be noted that in real situation roll and pitch occur about specific axes – geometry dependent (fig. 1.2).

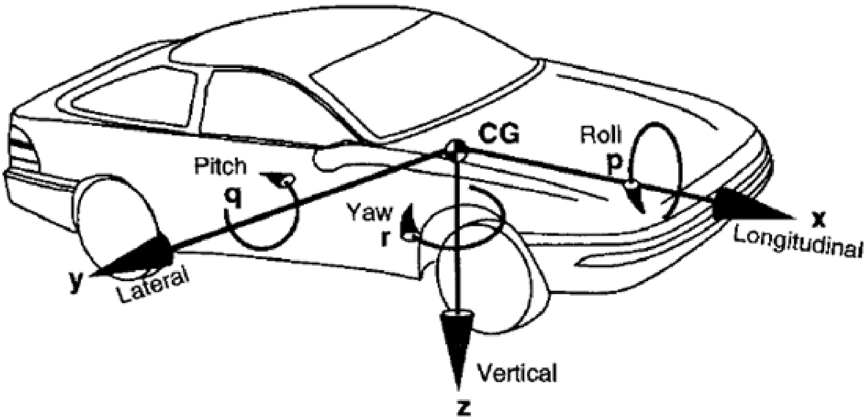

Figure 1.2 SAE axis system: x – longitudinal velocity, y – lateral velocity, z – normal velocity, p – roll velocity, q – pitch velocity, r – yaw velocity

1.3 Problem of vehicle movement

Vehicle movement is being affected by movement resistances. There are several types of resistances which act either on the wheel or the whole vehicle.

$$\mathbf{F= F_r+F_a+F_h+F_i+F_p} \tag{1.12}$$

F_r- rolling resistances

F_a- aerodynamic resistances (further on this subject in 1.3.5)

F_h- hill resistances

F_i- inertia resistances

F_p- pulling resistances

Let’s consider all of them.

1.3.1 Rolling resistances

Rolling resistance occur when tyre rolls on a surface in straight line. The resistances are caused by deformations of both tyre and road – which is mainly caused by road irregularities.

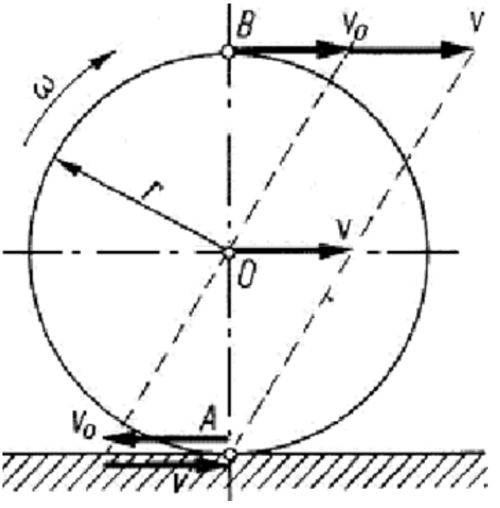

Figure 1.3 Rolling wheel

Main equation for rolling resistances.

$$F_t = G \cdot f_t \tag{1.13}$$

Rolling resistance coefficient (f_t) is determined differently for different vehicle speeds. For up to 140km/h.

$$f_t = {f_t}^0 \cdot (1 + cV^2) \tag{1.14}$$

$$F_t = {f_t}^0 \cdot (1 + cV^2) \cdot G \tag{1.15}$$

${f_t}^0$ values have been measured during road test and are gathered in a table.

Table 0.1 Main friction coefficient

Road surface	Rolling resistance coefficient (f_R')
Asphaltic concrete	0,012
Smooth concrete	0,014
Coarse concrete	0,015
Very fine - gradient granitic cube	0,016
Rolled gravel	0.02
Tarmacadam	0,025

Road paved with stone cubes (good quality)	0,033
Road paved with stones	0,04
Unpaved road	0,05÷0,14
Wet, sandy road	0,08÷0,15
Dry sand	0,15÷0,3
Grass surface	0,06÷0,110
Snow	0,04÷0,15

For speed exceeding 140km/h rolling resistance coefficient is determined by:

$$f_t = \frac{0{,}02}{0{,}01^{0{,}64}} + \frac{v^{3{,}7}}{0{,}1p^{2{,}08} \cdot 142 \cdot 10^7} \tag{1.16}$$

p – tyre pressure [kPa]

v – vehicle maximum speed [km/h]

As it can be seen from the equation – rolling resistance depends on the speed and pressure in the tyres. Figure shows the relation between these two factors.

Factors affecting rolling resistance:

- tire temperature,
- tire inflation pressure,
- velocity.

As already mentioned, the rolling resistance occurs due to energy dissipation caused by tire deformation. Hence, the type of a tire, its construction and material has significant influence on rolling resistance. Bias-ply tires generates greater rolling resistance comparing with radial- ply tires. Moreover, the thickness of sidewalls and tread as well as carcass construction also affects the magnitude of rolling resistance. namely, the thicker the sidewall and tread, the greater resistance will be generated. Furthermore, the force increases with respect to the increasing number of carcase plies. The material which tires are made of is also important in terms of rolling resistance generation. It appears that tread made of butyl rubber compounds produces the largest amount of rolling resistance, although this kind of material ensures best traction performance and road holding properties. Tires made of natural rubber compound produces smallest amount of rolling resistance, but are less effective in terms of vehicle traction and road holding comparing with synthetic or butyl materials [16].

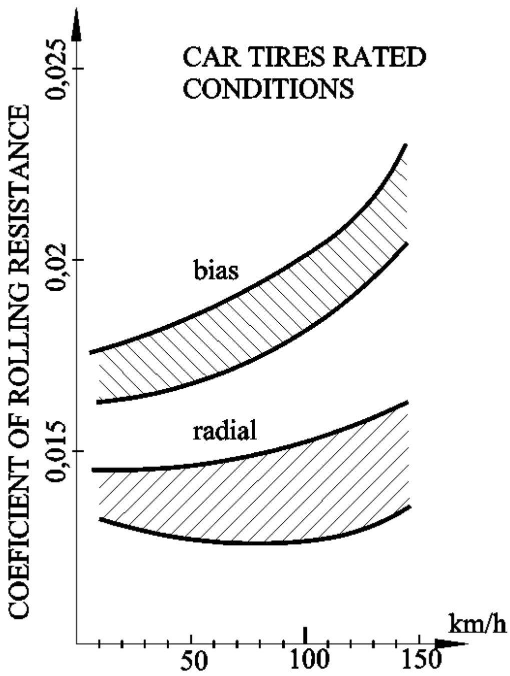

Figure 1.4 Influence of speed of a vehicle on coefficient of rolling resistance for different type of tires [16]

The rolling resistance is also dependant on the pavement structure. The force increases with respect to increasing porosity of the road. The smoothes surface produces the smallest amount of rolling resistance, however traction and road holding performance will be reduced in this case. Therefore, obtaining compromise between tire and the driving condition is crucial especially in case of high performance vehicle.

Tire temperature

It is obvious that, during driving, tires will change their temperature, and in consequence affecting the rolling resistance. The temperature of a tire will change the pressure of air inside the tire cavity, and therefore change the inflation pressure. The relationship of a internal temperature, pressure and velocity is depicted in *figure 1.5*.

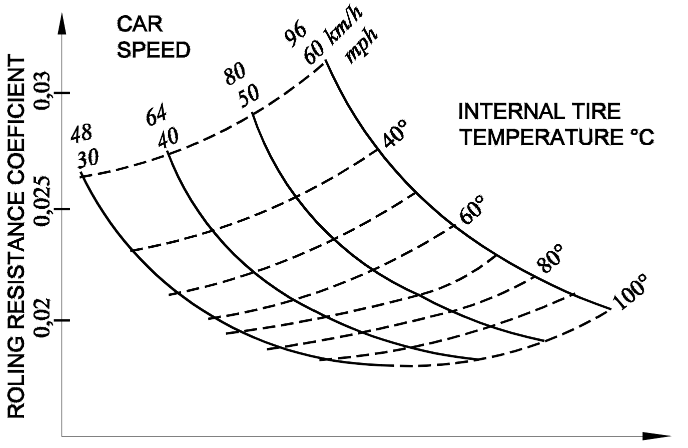

Figure 1.5 The relationship of internal temperature, pressure and velocity affecting the coefficient of rolling resistance [16]

Moreover, the temperature has significant influence on the properties of rubber tire. The rolling resistance decreases with respect to increasing shoulder temperatures of a tire. This is depicted in *figure 1.6*. It is also important to notice that the tire's shoulder temperature, not the ambient, is a basic factor for the rolling resistance coefficient determination.

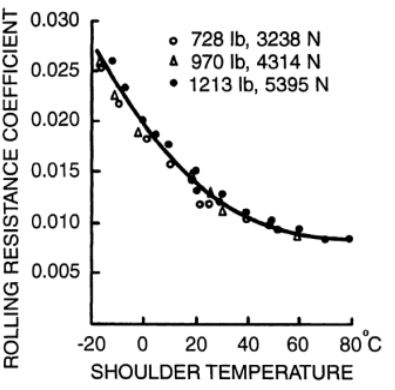

Figure 1.6 Influanc of shoulder temperature on rolling resistance coefficient [16]

Tire inflation pressure

In general, the rolling resistance increases with decrease of tire inflation pressure. However, this is only when the car drives on a hard surface. The magnitude of rolling resistance is dependent on both road and tire deformability. For example, considering sandy surface, great inflation pressure will not be favourable as the tire will penetrate the surface, and in consequence the resistance will increase. In oppose, less inflected tires will reduce ground penetration in consequence reducing the rolling resistance. The value of rolling resistance with respect to the surface kind and inflation pressure is depicted in *figure 1.7.*

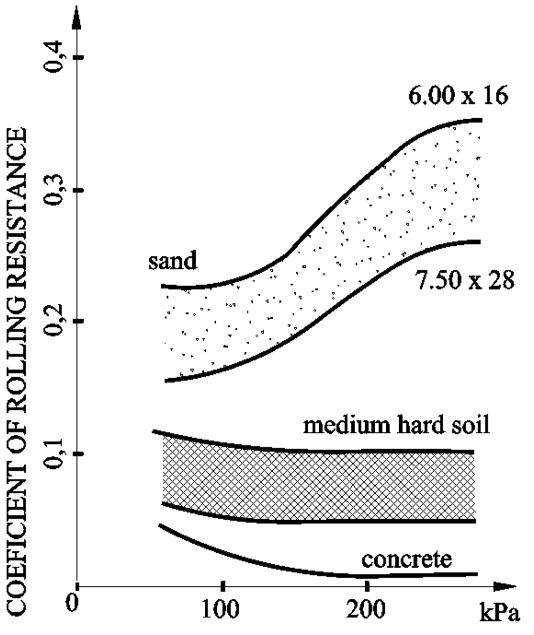

Figure 1.7 The value of rolling resistance with respect to the surface kind and inflation pressure [16]

Deflection of the tire also depends on its construction (size, sidewall stiffness etc.) as well as the load applied. The test has shown that the inflation pressure has much significant influence on rolling resistance of the bias and bias – belted tires than the radial – ply tire. It also should be noted that the test were carried out with constant inflation pressure. The pressure was regulated in order to maintain required values designated as letters ranging from *A* to *F*. It is shown in *figure 1.8 [16]*

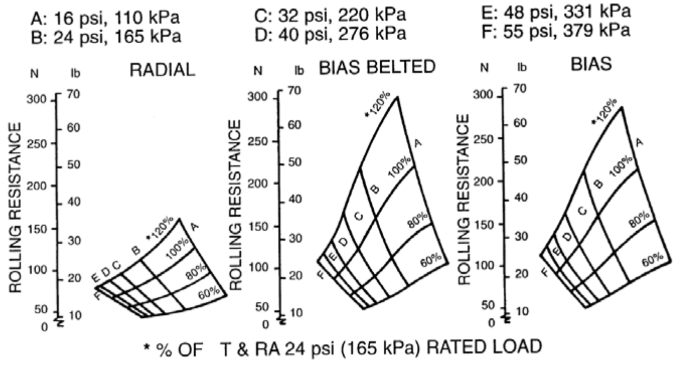

Figure 1.8 The comparison of loaded bias, bias – belted and the radial – ply tires in terms of their rolling resistance [16]

<u>Velocity</u>

The driving speed has different influence on different tire types. In this case rolling resistance occurs due to the fact that the increasing speed causes increase of work in deformation the tire as well as the structure vibration. Another factor affecting rolling resistance originating from driving speed is phenomenon called standing wave. The standing wave occurs because tread does not recover from distortion immediately after it leaves the surface. Of course this phenomenon arises only when the level of threshold velocity, which can be determined by means of *equation 1.17*, is breached*.* The standing wave significantly increases the loss of the energy and in consequence provokes great tire temperature increase [1].

$$V_{max} = \sqrt{\frac{\tau_{Ct}}{\rho_t}} \tag{1.17}$$

Where:
τ_{Ct} is a circumferential tension in the tire
ρ_t is a density of tread material

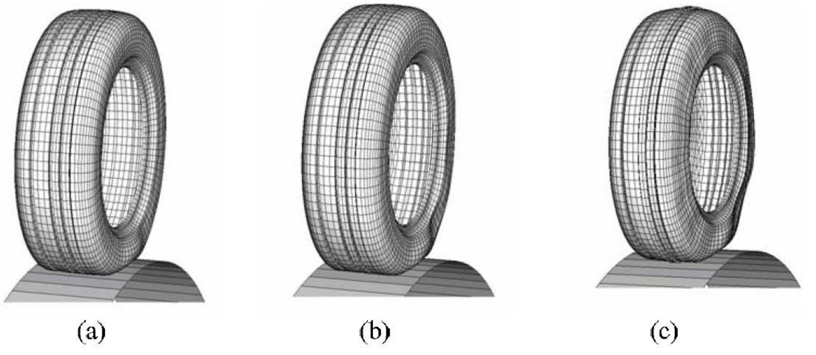

Figure 1.9 Simple model of standing wave: a) 160km/h, b) 250 km/h, c) 245 km/h [42]

1.3.2 Hill resistances

During driving uphill, vehicle need to overcome the hill resistance. It is a result of energy required to rise the vehicle. Very simple equation for hill resistance force is weigh and hill angle dependent.

$$F_w = mg \cdot sin\alpha \tag{1.18}$$

Below figure 1.10 showing fundamental well known relations.

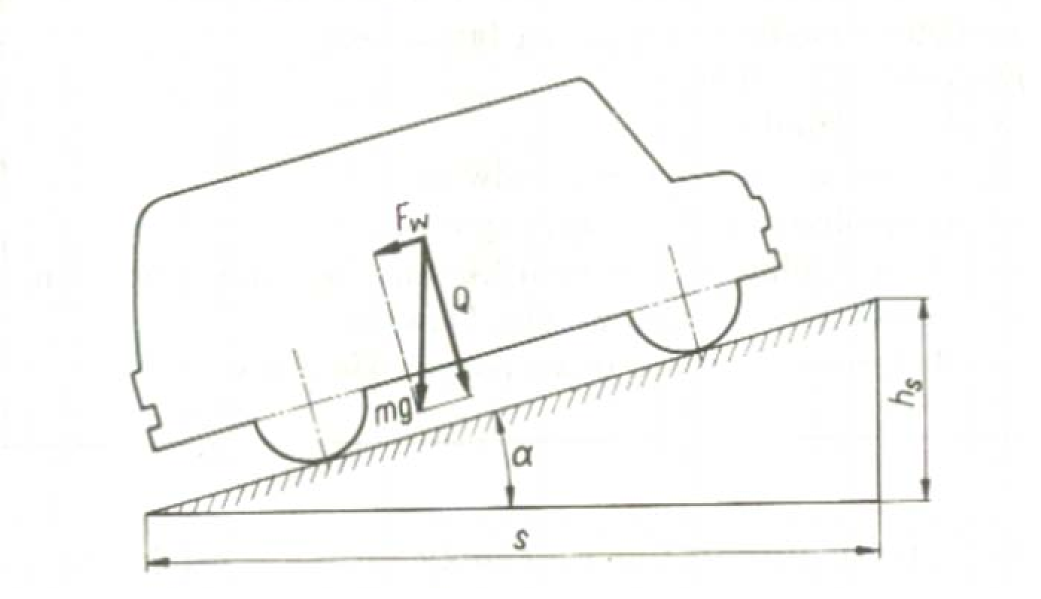

Figure 1.10 Hill resistances

In practise road signs inform the driver how steep is the hill in percentage – knowing this value hill resistances can be calculated with the following equation.

$$100 \cdot tg\alpha = 100\frac{h_s}{s} = w \tag{1.19}$$

and thus

$$F_h = mg \cdot \frac{w}{100} \tag{1.20}$$

1.3.3 Inertia resistances

Vehicle consist of various mechanisms which rotate during movement. Inertia of those elements cause resistances during starting the movement and when increase in speed is required.

Basic inertia forces are shown on fig. 1.11

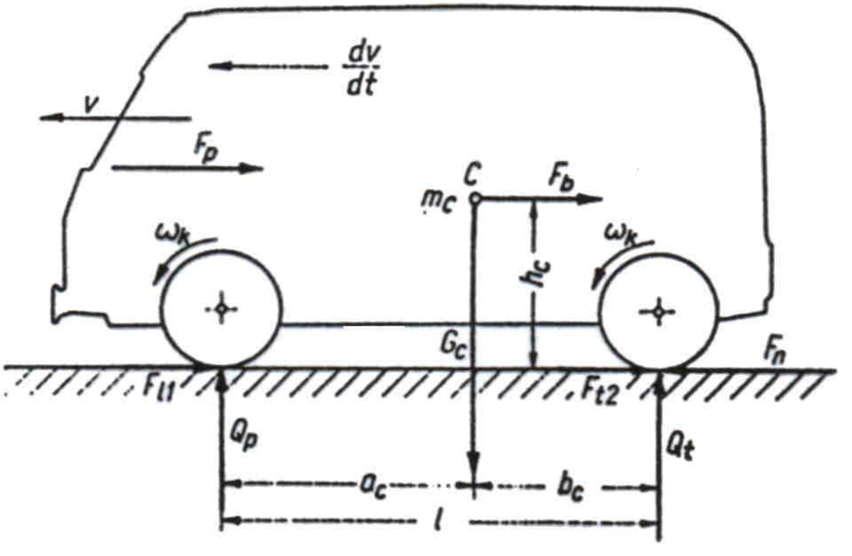

Figure 1.11 Inertia resistances

Equation describing inertia resistances has a form:

$$F_i = m_c\frac{dv}{dt} + I_{kf}\frac{d\omega_k}{dt}\cdot\frac{1}{r_d} + I_{kr}\frac{d\omega_k}{dt}\cdot\frac{1}{r_d} + I_s\frac{d\omega_k}{dt}\cdot\frac{{i_g}^2\cdot {i_d}^2}{r_d\cdot\eta_m} \tag{1.21}$$

m_c – mass of the vehicle

I_{kf} – front wheels moment of inertia – all rotating masses connected with wheels in reference to wheel axis

I_{kr} – rear wheels moment of inertia – all rotating masses connected with wheels in reference to wheel axis

I_s – moment of inertia – all engine elements masses in reference to crankshaft

η_m – powertrain efficiency

r_d – wheel dynamic radius (deflected)

i_g – gearbox ratio

i_d – final drive ratio

For practical calculation - coefficient is used which describes inertial resistances of all listed above vehicle elements. After this simplification the equation has a classical form:

$$F_i = \delta \cdot m_c \cdot \frac{dv}{dt} \tag{1.22}$$

Where

$$\delta = 1 + \frac{I_k}{m_c \cdot r_d{}^2} + \frac{I_s \cdot i_g{}^2 \cdot i_d{}^2}{m_c \cdot r_d{}^2} \tag{1.23}$$

1.3.4 Pull resistances

In case of pulling a trailer, pull resistances occur. These resistances are a sum of all mentioned above resistances referred to the pulled object – usually a trailer.

$$F_p = F_r' + F_a' + F_h' + F_i' \tag{1.24}$$

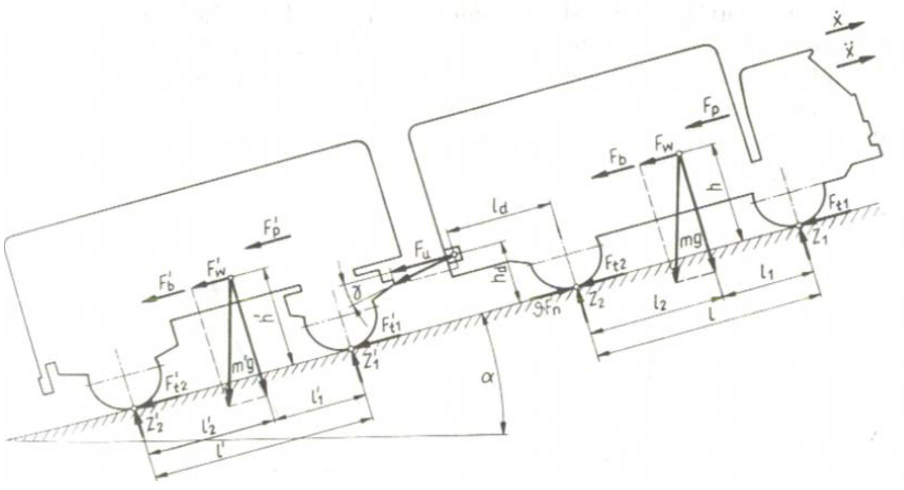

Figure 1.12 Pull resistances

1.3.5 Aerodynamics resistances

An aerodynamics can be defined as a study of the interaction between moving vehicle's body and the atmosphere. The forces which are caused by this interaction are called aerodynamic forces. The forces are highly depended on the air density and the relative velocity of the air and the body. The aerodynamic force which is responsible for opposing the forward motion is called the aerodynamic drag. The drag has tremendous influence on vehicles speed and fuel consumption due to the fact that the engine has to overcome this force during driving. The force directed vertically is called the aerodynamic lift. The lift force is considered as positive when directed upwards. The lift occurs due to difference of pressure on top and underneath of a car. The positive lift reduces the friction between the tires and roads surface, hence changing the steering and propelling characteristics of the vehicle. Negative lift also called a downforce (force directed downwards) improves the stability performance of a car as well as enables tighter cornering. The negative lift however, provokes increase of undesirable rolling resistance.

A vehicle aerodynamics can be separated on external and internal. In case of internal aerodynamic the air is directed through the body in order to provide engine cooling, a passenger compartment ventilation, air conditioning and heat management. However, an internal flow will be only briefly describe in this module. The external aerodynamics is not only combined with all aerodynamic forces and moments but also with such features like maintaining dirt free windows, aero-acoustics or rain water flow.

Mechanics of air flow around a vehicle

Assuming that air is incompressible fluid, the automotive aerodynamics is governed by Bernoulli equation (1.25). This equation determines the relationship between air speed and pressure. Another words, when the pressure increases, the velocity must decrees at this same point. Moreover, thanks to this equation it is easy to determine change of flow properties on two distant points.

$$p + \frac{1}{2}\rho {V_d}^2 = const \tag{1.25}$$

Generally, the Bernoulli equation is a sum of a dynamic and static pressure. A dynamic pressure dictates the magnitude of a drag and lift force. For a car travelling on a road the atmospheric pressure is considered as a static pressure.

$$p_{static} + p_{dynamic} = p_{total} \tag{1.26}$$

$$p_{dynamic} = \frac{1}{2}\rho V_d^2 \tag{1.27}$$

Where:
ρ is air density,
V_d is driving speed,

At the point where the flow velocity is zero (indicated as A in *figure 1.13)* the dynamic pressure is cancelled and consequently only static pressure is considered. This point is called a **stagnation point** and pressure that occurs there is called **impact pressure** or **stagnation pressure**, which is the highest pressure (in that case) that may occur for given vehicle. Beyond a stagnation point the air molecules are pushed upwards and downwards the vehicle. A location of the stagnation point is depended on cars construction. During driving the flow does not always approaches vehicle in its symmetry plane, hence the location of stagnation point changes with respect to the direction of the air path. The path of a fluid (ideal) particle is called a streamline. The shape obtain by means of those streamlines is called a streamline picture.

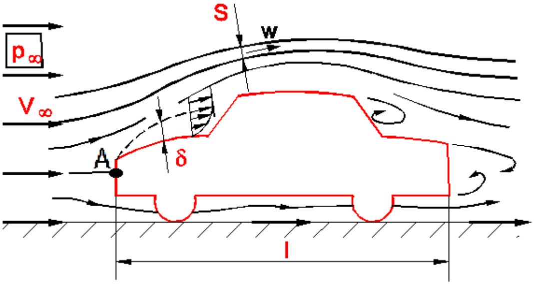

Figure 1.13 Schematic flow around a vehicle [9]

Assuming that the flow separation from the body does not occur, the viscous effect in the fluid is restricted to a thin layer called a boundary layer, beyond witch the flow is inviscid. Within the boundary layer the velocity decreases from the value of the external inviscid flow until it reaches the immediate vicinity of the body where the air molecules are stopped, and so called no-slip condition is fulfilled. *Figure 1.14* shows the flow in the boundary layer on a flat plane. As it can be seen the external flow has parallel streamlines and constant velocity (V_∞) and pressure (p_∞) before reaching the plane. When the plane is reached, the boundary layer begins to be formed. In the front of the plane the flow distortion does not occur and the streamlines are almost parallel to the plane. This kind of flow is called laminar boundary layer. Within the laminar boundary layer The air molecules that are close to the plane and moves with smaller speed are impacting the air molecules which are higher with respect to the plane. In consequence the flow speed is decreased. The thickness of the boundary layer increases with respect to distance (x) and the kinematic viscosity of the fluid (v). After some distance (x_{kr}) the distortion occurs and the flow becomes unstable (the developing transition zone is formed), accordingly the boundary layer transmits to the turbulent stage.

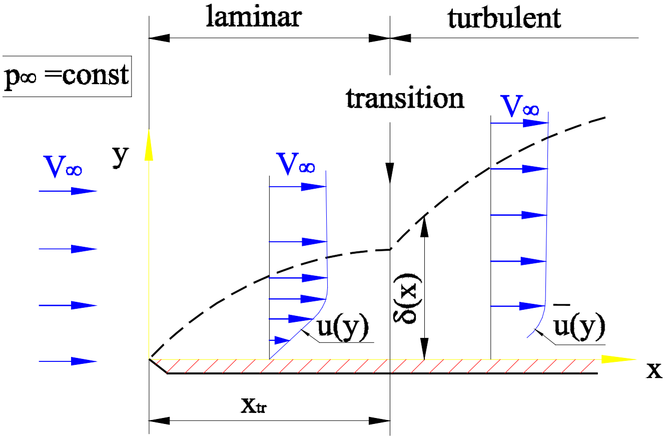

Figure 1.14 The boundary layer on a flat plane [9]

The transition from laminar to turbulent boundary layer is governed by the value of Reynolds number presented *in equation 1.28*. For a flat plane the transition occurs for the Re number of $5 \cdot 10^5$. This value however, applies only for negligible pressure gradient. a pressure decreasing in the flow direction causes a stabilisation of the laminar boundary layer, whereas the pressure increasing in the flow direction causes an earlier transition to the turbulent flow. The transition is also depended on such a features like for example surface roughness, body deformations etc. Summarising, for medium Reynolds number, the transition from laminar to turbulent boundary layer occurs in the region of minimum pressure. As the Re number increases the transition point moves upstream.

$$Re = \frac{V_\infty \cdot x}{\upsilon} \tag{1.28}$$

The turbulent boundary layer represents large distortion and chaotic molecule movement, however it is attached to the body, therefore it is less sensitive on cars body design, hence smaller form drag is produced. Due to the flow attachment, the larger friction occurs effecting in larger friction drag.

Both laminar and turbulent boundary layer are dependent on pressure distribution around a vehicles body. Let's consider a curved shape presented in *figure 1.15*. This shape can be a schematic representation of a rear end of a vehicles roof. It can be assumed that At the point (*I*) the flow speed is high and accordingly the pressure there must be low. The conditions are changing as the flow goes

farther downstream. At the point (*II*) the pressure is greater than at the point (*I*) and consequently the air speed must decrease. hence the flow is directed from lower towards higher pressure. It can be only accomplished by means of loss of the kinetic energy of air molecules. If the pressure increase is not radical the boundary layer does not detaches from the surface. However, when the pressure changes rapidly, the process of developing boundary layer can be insufficient in terms of sustaining the movement of air molecules in the surface vicinity. In that case the flow separates from the body. The point where the flow gets separated is called separation point indicated as (*A*) in *figure 1.15*. Beyond the separation, at the (*III*) point, the air molecules are pushed backwards due to adverse pressure gradient (an increase of pressure in the direction of flow). Another words the higher pressure pushes back the air molecules towards the lower pressure. The separation generates a type of drag called pressure drag, therefore, large emphasis is given to ensure attached flow as long as possible. Both laminar and turbulent boundary layer are subjected to the separation phenomenon, however, turbulent boundary layer is capable of withstanding much steeper pressure gradient due to the fact that the turbulent mixing ensures intensive momentum transport from the outer flow towards a vehicles body. The separation does not occur for pressure decrease in the flow direction (see *figure 1.16*).

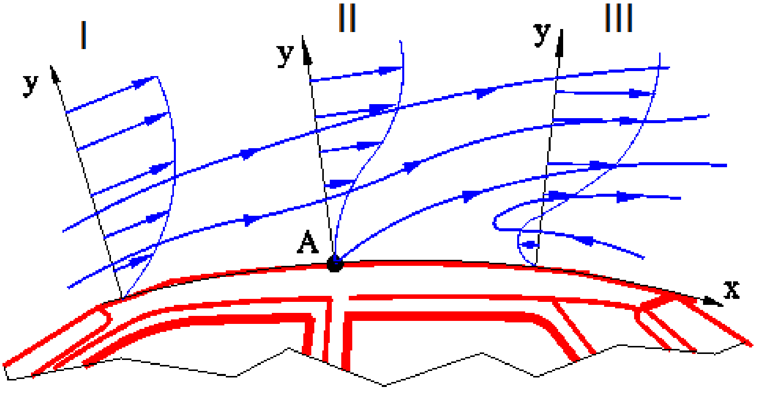

Figure 1.15 Laminar and turbulent boundary layer separation

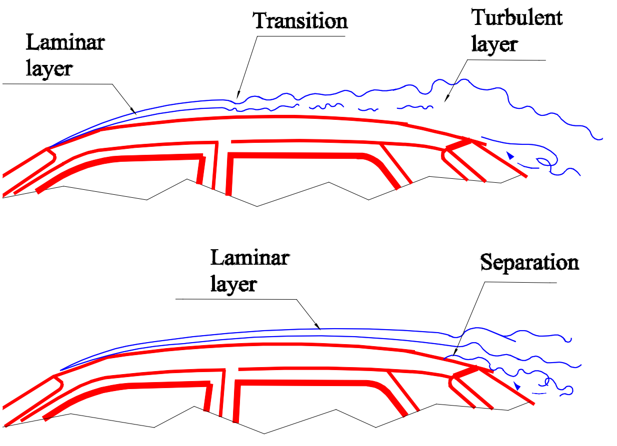

Figure 1.16 Flow separation

As a summarisation consider the flow around a vehicle installed in a wind tunnel depicted in *figure 1.17*. In order to visualise the streamline the smoke streams has been introduced. Upstream of the vehicles the streamlines have the same velocity. When the flow reaches front of the body the static pressure and velocity of the flow changes according to Bernoulli in consequence the streamlines gets separated and one of the streamlines impacts at the stagnation point. Most of them is led by the bonnet construction upwards and smaller part of the flow goes underneath of the car. The static pressure exceeds the ambient value effecting in generation of force which turns the airflow. Increasing pressure in the flow direction provokes the separation, hence the flow separates from the body near vertical radiator grill and point (a). however After some distance on the bonnet, the reattaches at point (b). The pressure again increases in front of the windscreen generating next separation of the flow at (c). The reattachment occurs farther downstream on the windscreen at (d) and separates again at the top corner (e) then reattaches farther downstream on the roof at (f). The pressure on the roof decrease below ambient and accordingly the velocity must increase. The velocity can be estimated based on streamlines concentrations. Region with streamlines close to each other represents greater flow velocity than in case of region with low streamline concentration. At the rear of the vehicle, as the air flow long enough against the adverse pressure gradient, the final separation occurs at point (g), however the flow can reattach again. It is depended on the shape of the rear end of the car. Considering the car presented in *figure 1.17* partial and unsteady reattachment may occur at point (h) [9].

A flow separation must occur at some point. It is beneficial in terms of economics to provide attached flow as long as possible. Generally, the smaller the area after separation, the lower the drug. Due to large pressure difference on the front and on the rear of the vehicle a pressure drug is generated along with large wake and various kinds of vortices. Furthermore, *figure 1.18* shows the flow around a vehicle with typical locations of separation.

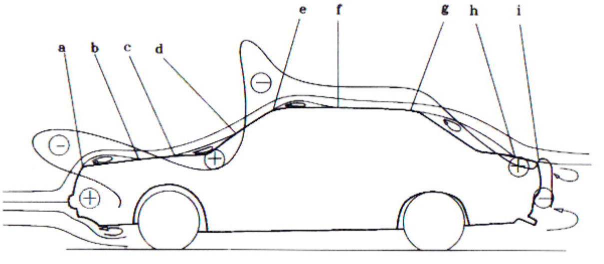

Figure 1.17 The central line pressure distribution resulting in flow separation [41]

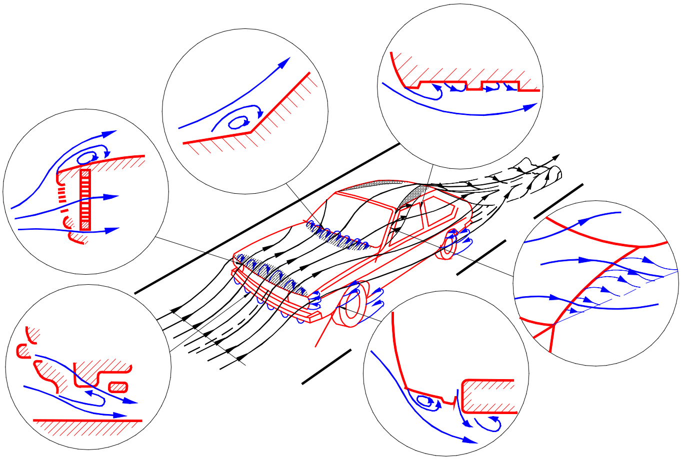

Figure 1.18 Flow around a vehicles and typical location of flow separation [9]

Pressure distribution on a vehicle

A moving vehicle is subjected to the variably distributed pressure which the air stream develop on the body. The variable pressure distribution is a reason of occurrence all aerodynamic forces. The positive pressure is directed towards the centre of a vehicle while the negative pressure is directed in opposite direction. There are three areas that has influence on minimising drag force. First, there is a low pressure area on the edge of the bonnet which provokes the flow reattachment after separation on vertical radiator grill. Then, there is a low pressure area responsible for reattachment on the roof line and farther downstream, on the end of the boot where the separation occurs.

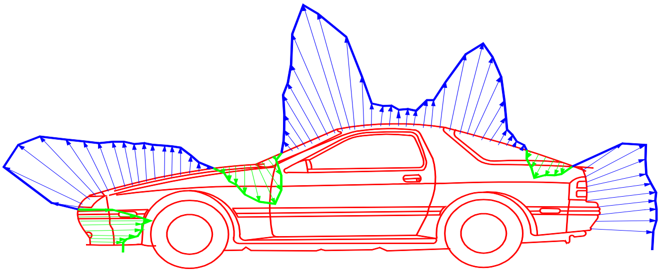

Figure 1.19 Pressure distribution plotted on the vehicle

The pressure distribution is dependent on a car construction and speed. Therefore, when describing pressure distribution on a vehicle body, it is convenient to use non dimensional coefficient called a pressure coefficient which is denoted by Cp. the difference of a local static pressure at any point and a static pressure of free stream is highly dependent on a free stream dynamic pressure. Hence the ratio of those pressures, presented in equation 1.29 is independent from the flow speed. Moreover, the pressure coefficient can be used for calculation of a pressure at any free-stream or driving speed.

$$C_p = \frac{p - p_\infty}{\frac{\rho V_\infty^2}{2}} = 1 - \frac{V}{V_\infty} \tag{1.29}$$

Where:
p and V are pressure and velocity at any point on a vehicle,
p_∞ and V_∞ a free stream pressure and velocity,
ρ is the air density.

Considering the flow around a vehicle, the stagnation point occurs when the flow velocity is zero hence the Cp is 1. On the other hand the Cp is zero when the p is equal to p_∞ and the velocity V is equal to V_∞. The Cp is negative when p is less then p_∞ and V is greater than V_∞. Usually The Cp is presented graphically on the vehicle outline.

Aerodynamics forces: drag, side force, lift force, pitching, yawing and rolling moments, crosswind sensitivity

Every vehicle, during motion, is subjected to forces and moments occurring due to air flow around a vehicle and consequently variously distributed pressure. The magnitude of those forces and moments has to be known already during a vehicles designing. Preliminary, the vehicle design is investigated in terms of aerodynamics by means of the Computational Fluid Dynamics (CFD) software [9]. Then the scaled-down model is tested in wind tunnel. In order to obtain results which are independent from dimensions the forces and moments coefficients are introduced. However, in order to obtain valid results for both scaled- down and full scale model the Reynolds number similarity has to be fulfilled. Those are coefficients enable universal comparison of vehicles' performance in terms of aerodynamics. It is important to notice that moments reference centre is located in the middle between front and rear axle. sometimes however, moments are referred to a centre of gravity of the vehicle, while forces are combined with centre of pressure (COP). The centre of pressure (COP) is a theoretical point of force application and usually it is distant from the centre of gravity. The types of forces and moments can be separated with respect with wind angle of attack (β). *Figure 1.20* shows the schematic behaviour of car subjected to frontal wind β=0 deg. There are three components associated with this kind of flow, i.e. drag and lift force as well as pitching moment.

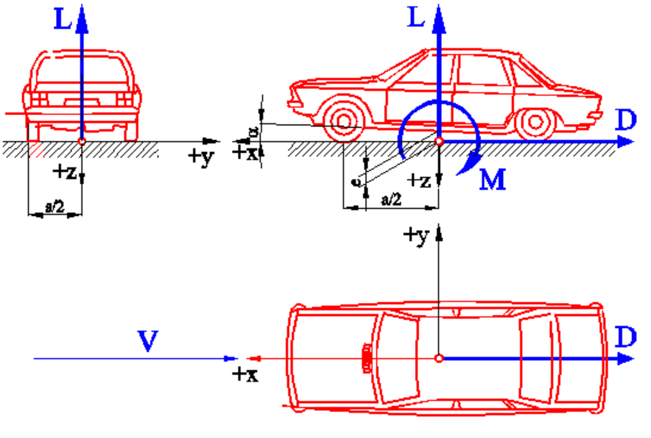

Figure 1.20 The direct flow influence on a vehicle

Drag force

The **drag force** resists the motion forward, hence this force is directed parallel to the centreline of a vehicle. The positive value of drag is directed backwards the vehicle. The drag force is most commonly denoted as ***D*** and can be determined by means of *equation 1.30*.

$$D = \frac{1}{2}\rho V^2 C_D A \tag{1.30}$$

Where:
ρ is the air density
V is the flow velocity
A is the frontal area
$\mathbf{C_D}$ is the drag coefficient

In order to describe the aerodynamic performance of a car and compare different cars a non-dimensional drag coefficient is introduced ($\mathbf{C_D}$). Generally, the smaller the coefficient the better, however it does not mean that the lower amount of drag will be produce. For example the Renault Scenic has lower drag coefficient than Renault Megane, however the frontal area of scenic is greater and consequently grater drag is produced. Accordingly, using the product of drag coefficient ($\mathbf{C_D}$) and the frontal area (**A**) , of a vehicle is more reliable in terms of vehicle aerodynamics comparison. The product $\mathbf{C_D}$ **A** is usually called an aerodynamics factor and it is published along with the drag coefficient [9].

The magnitude of a frontal area is measured as a projection of a vehicle's cross section enlighten on parallel to the car's central line plane. It can be also calculated using *equation 1.31*. The frontal area of a vehicle is depicted in *figure 1.21.* [41]

$$A = w_p H B \tag{1.31}$$

Where:
$\mathbf{w_p}$ is a filling factor
H is a height of a vehicle
B is a width of a vehicle

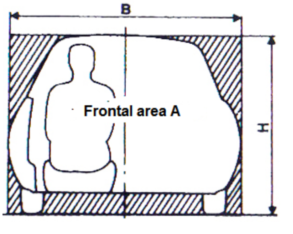

Figure 1.21 Frontal area of a vehicle [41]

In general, the drag can be express as a sum of a friction and pressure drag, however the frictional drag is most often significantly lower comparing with the pressure drag. Large emphasis is given for designing the vehicle which produces as low drag as possible. Construction of both front and rear is crucial in terms of generating forces and moments which impose the cars movement. For example, it has been measured that boundary angle of rear window is 30 degrees for flow to separate. Therefore, the body construction has to be optimised in this matter.

Friction drag

As it was already explained drag is a force resisting forward motion. Hence the drag can be result of molecular friction which provokes a shear stress acting everywhere on the surface of a body, assuming that separation did not occur. This kind of drag is called friction drag (D_F). The friction drag is relatively small as it can reach up to 30% of total drag. It is the friction drag which provokes either the laminar or turbulent boundary layer in dependence on the magnitude of Reynolds number. The turbulent boundary layer creates much greater friction drag, therefore it is beneficial in terms of economy to maintain the laminar boundary layer as long as possible.

Pressure drag

The pressure drag (D_P) occurs due to extremely large pressure gradient on the rear of a vehicle. This pressure gradient causes the flow separation. In another words, the pressure drag results from the fact that in front of the car the pressure is significantly higher than in the rear. It is common to explain the pressure distribution around the object with the aid of the circular cylinder. The pressure distribution around the cylinders is shown in *figure 1.22*. In the front of the object the flow behaves as inviscid, further downstream negative pressure occurs however, due to flow separation. The point of the flow separation is depended on Reynolds number so also on the flow velocity as well as the fluid conditions (i.e. density). Laminar boundary layer occurring for low Reynolds number the separation point of located close to the maximum thickness of the cylinder. This condition is depicted in *figure 1.22 (b).* This separation causes large wake region behind the object and large drag coefficient. When the Reynolds number is great enough the transmission occurs in front part of the cylinder [9]. As it was already explained in section (1.*3.5.1 Mechanics of flow around a vehicle)* a turbulent boundary layer separates later comparing with laminar stage. Therefore, the wake is narrow and the drag coefficient is much lower see *figure 1.23.*

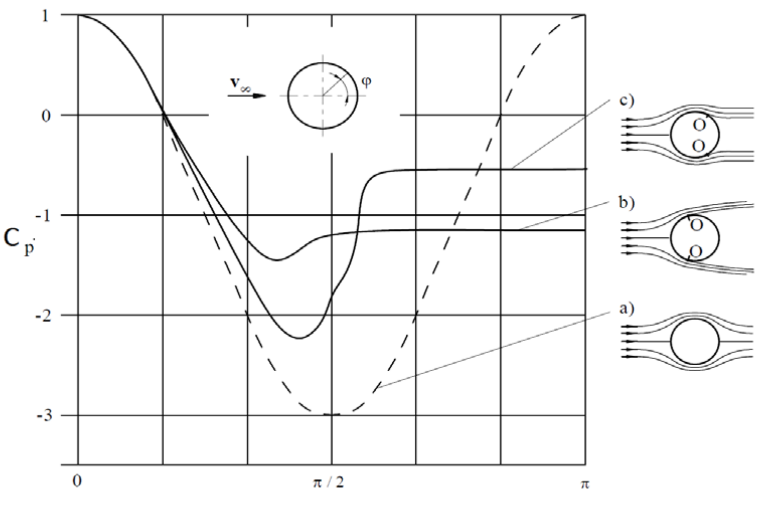

Figure 1.22 Pressure distribution around the circular cylinder: a) inviscid flow, b) subcritical flow, laminar boundary layer, c) subcritical flow, turbulent boundary layer [43]

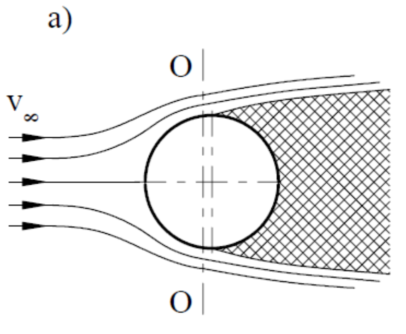

Figure 1.23 The flow around a cylinder: a) laminar separation, b) turbulent separation, [43]

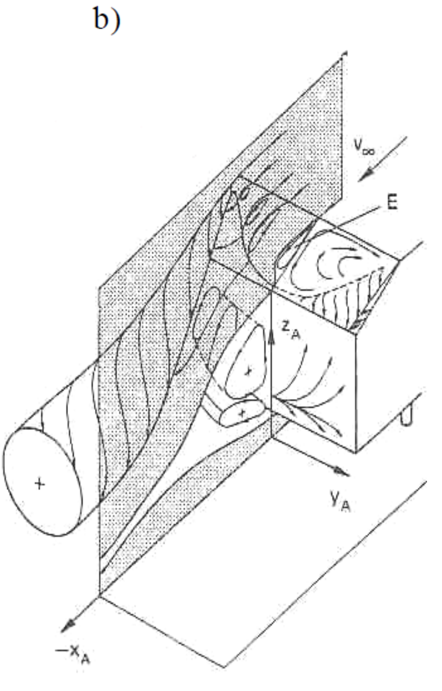

Figure 1.24 Three dimensional high-drag vortex system [44]

Trailing vortex drag

The drag, explained above is two-dimensional, hence it does not entirely reflects the reality. It is obvious that in reality the flow around the vehicle is tree-dimensional, therefore the consequence of pressure difference between sides, underside and top side of the vehicle should be under consideration. The air tends to flow from the high pressure towards the lower pressure in the wake. In consequence the vortices are developed. Those are so called trailing vortices. Production of the vortices is done with cost of large amount of energy , hence it is considered as part of drag. The drag occurring during vortices is called trailing vortex drag or induced drag. *Figure 1.24* shows the three dimensional flow around a vehicle along with developed vortices [9]. Considering the three dimensional flow it can be seen that the air is drawn by the vortices backwards the car . In consequence behind the vehicle the low pressure which pulls the flow down is created. The reduced pressure on the rear of the car is accompanied by increase of drag. Moreover, form momentum consideration it can be noticed that air drawn to the rear, causes a corresponding reaction which will pulled whole vehicle backwards [9].

Lift force and pitching moment

Besides drag, The parallel air flow direction results in lift force as well. This force is responsible for reducing the tire- road interaction, and in consequence the friction. The positive lift value is directed upwards the vehicle. The lift occurs due to the fact that the pressure underneath is much higher than on top of the car. It is because the molecules of air, which ware led under the vehicle has shorter distance to coverer comparing with molecules that were led upwards. Hence, as predicted by the Bernoulli equation, the flow velocity underneath the car does not have to be as significant as it is in case of flow on top of the body. Accordingly, the lift is dependent on the underbody flow and pressure distribution. The lift force can be determined by means of *equation 1.32.*

$$L = \frac{1}{2}\rho V^2 C_L A \tag{1.32}$$

Where

ρ is the air density
V is the flow velocity
A is the frontal area

C_L is the lift coefficient

In equation above, the frontal area is used, despised the fact that lift is more associated with top surface of the car. Obtaining a negative lift, or another words down force, improves stability, handling as well as acceleration and braking performance which is extremely important specially in sports cars. The lift is controlled by applying spoilers, underbody pans, or enabling easer flow outlet from underneath of a car by forming rear of an underbody as a diffuser. This will be discussed in grater details in section (1*.3.5 Aerodynamic aids).* The lift force is combined with pitching moment which is also result of parallel air flow direction. The pitching moment can be express with *equation 1.33*

$$M = \frac{1}{2}\rho V^2 C_M A l \tag{1.33}$$

Where
ρ is the air density
V is the flow velocity
A is the frontal area
$\mathbf{C_M}$ is the pitching moment coefficient
l is the reference length, usually wheelbase

When the vehicle's nose is pulled upwards then the pitching moment has a positive value. Conversely when the nose is pushed downwards the moment is negative. From vehicle handling point of view it is better when the moment is positive (nose up) due to the fact that the rear axles are subjected to grater load comparing with front axle and therefore provides safer oversteer characteristics.

Knowing the wheelbase (**l**) and position of COP it is possible to calculate the lift distribution on front and rear axle, and furthermore on each wheel. as it was already mentioned reference point of pitching moment is located in the midway between front and rear axle. Moreover, total lift can be express as a sum of lift on frond axle ($\mathbf{L_f}$) and lift on the rear axle (**Lr**) (see *figure 1.25*). Hence equilibrium *equation 1.34* is

$$L = L_F + L_R \tag{1.34}$$

$$-\frac{1}{2}L_F + M + \frac{1}{2}L_R = 0 \tag{1.35}$$

hence

$$M = \frac{1}{2}L_R - \frac{1}{2}L_F \tag{1.36}$$

Furthermore, since the coefficient of lift C_L and pitching moment C_M are known, the lift distribution can be determined using following equations:

$$C_{L_F} = \frac{1}{2}C_L + C_M \tag{1.37}$$

$$C_{L_R} = \frac{1}{2} C_L - C_M \tag{1.38}$$

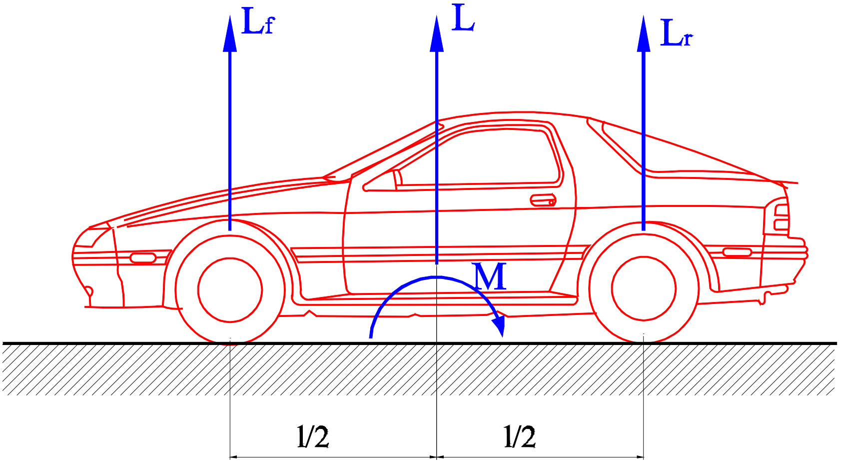

Figure 1.25 Lift force brake down

Crosswind

When the flow angle of attack is different than zero (β≠0), another words the flow is not parallel to the central line of the vehicle, three components arises, i.e. side force, pitching and rolling moments. *Figure 1.26* shows the schematic behaviour of vehicles subjected to this kind of flow.

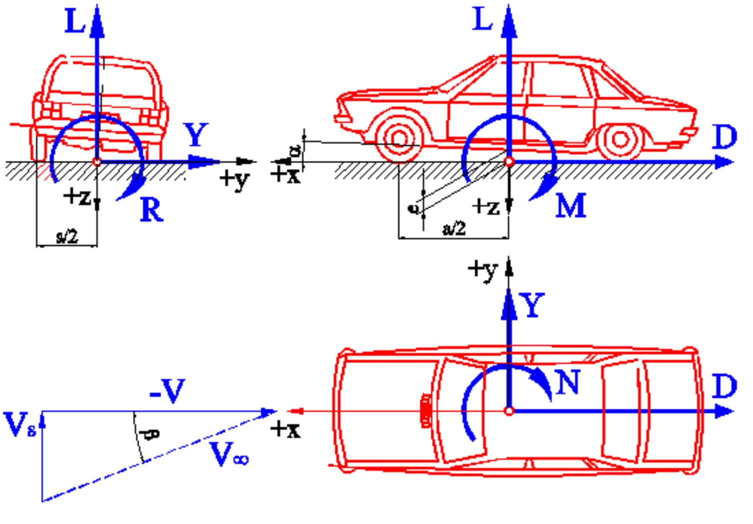

Figure 1.26 The crosswind influence on a vehicle [9]

Side force

The side force occurs when the air stream is directed in angle β which is not parallel to the central line of the vehicle. The magnitude of this force is depended on the angle of attack (β). The positive value of side force is directed towards a starboard (right side) a vehicle. The side force can be calculate by means of *equation 1.39*.

$$S = \frac{1}{2}\rho V^2 C_S A \tag{1.39}$$

Where:
ρ is the air density
V is the flow velocity
A is the frontal area

C_S is the side force coefficient

The aerodynamic side force arises due to large difference of pressure on both sides of the vehicle. Generally, the leeward side will experience a negative pressure whereas flow impinging the windward side will generate positive pressure. As it has been already mentioned the magnitude of the side force depends on the flow angle of attack (so called yaw angle β). Low yaw angle are accompanied by occurring of very negative pressure (suction) on leeward side of the front end of the car and leeward A-pillar. Farther downstream on the leeward side only law negative pressure occurs. For larger yaw angles flow separates at the leeward side of the bumper and A-pillar. Therefore, lower negative pressure occurs in this region. The airflow impinging the windward side generates low positive pressure on front of a car. Further downstream however, this positive pressure transfers into low negative pressure.   the magnitude of the positive pressure depends on the yaw angle as well. It increases with increase of the yaw angle. Hence, for high yaw angle, significant increase of pressure at the leeward side can be observed [9].

Yawing and rolling moments

As any other force the side force is a base for moment occurrence. in this case those moments are yawing moment (**N**) and subsequently explained rolling moment (**R**). Generally the yawing and rolling moment can be calculated by means of analogues *equation 1.40.*

$$N = \frac{1}{2}\rho V^2 C_N A l \tag{1.40}$$

ρ is the air density
V is the flow velocity
A is the frontal area

C_N is the yawing moment coefficient

Both yawing and rolling moments are considered to be positive when the vehicle yaw or roll away from the wind direction. The rolling moment “tries” to overturn the vehicle subjected to the crosswind. The magnitude of this moment depends on the pressure distribution around a vehicle and consequently on the shape of the car.

Crosswind sensitivity

During real driving condition, a vehicle may be subjected to the significant lateral wind acting constantly for some distance. This is consider to be a static case. The dynamic case arises when the lateral wind impinges a vehicle rapidly and the interaction is rather short, for example when exiting a tunnel. A car subjected to crosswind tends to deviate from its original trajectory. Generally, the deviation from the course is rather effect of dynamic case due to the fact that during long time of constant crosswind the driver has greater possibility of adjusting a vehicle to appropriate path.

Vehicle’s crosswind sensitivity is usually characterised by magnitude of the deviation from the original path. The crosswind sensitivity is determined by means of test during which a tested vehicle drives near to sets of fans deployed in a row as depicted in *figure 1.27.*

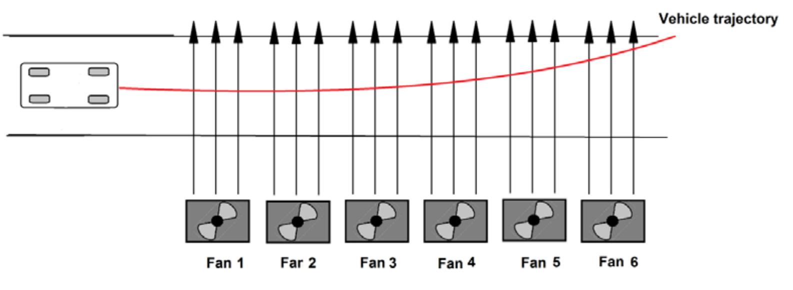

Figure 1.27 Schematic representation of crosswind testing

Usually the test is performed with fixed steering wheel position (“*fixed control*”). The tested vehicles drives through an area under influence of crosswind generated by the fans. As a result the car partially losses its stability. The magnitude of deviation from course is dictated obviously by mass of the vehicle, its size, and mostly by position of the centre of pressure with respect to the centre of gravity. *Figure 1.28* shows schematically the consequences of location of COP with respect to the centre of mass. As it can be seen if the COP is behind the centre of gravity the nose of the vehicle will be rotated towards the flow direction which is undesirable due to increased cross wind sensitivity and consequently worst stability performance. Conversely, if the COP is before the centre of gravity, then the vehicles nose will be pushed towards the leeward side which results with a stabilising effect.

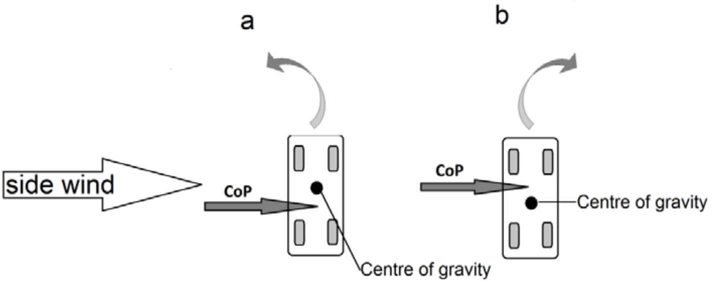

Figure 1.28 Effect of relative position of the CoP and the centre of gravity

C. C. MacAdam [40], based upon vehicle crosswind testing, proposed a mathematical representation of yaw rate provoked by applied constant side force (*equation 1.41*).

$$\frac{r}{S} = \frac{(c+d)}{mV(c+\zeta)} \tag{1.41}$$

This dependence concerns certain fundamental vehicle properties such as neutral steer point, side force centre of pressure, vehicle centre of mass and the tire yaw damping moment. Bearing in mind that aerodynamic side force can be expressed by means of *equation 1.39*, and multiplied by slip angle (**α**) which occurs due to this force ($S = 0{,}5\rho V^2 C_S A\alpha$), the equation above can be rearranged to express the passive crosswind sensitivity in terms of yaw rate response per angle of aerodynamic slip angle:

$$\frac{r}{\alpha} = \frac{\frac{1}{2}V^2 C_S A}{mU}\,\frac{(c+d)}{(c+\zeta)} \tag{1.42}$$

Knowing the geometrical dimension of a vehicle the equation can be rearrange in order to express the crosswind sensitivity in terms of completely aerodynamic properties of the vehicle. [40]

$$\frac{r}{\alpha} = \frac{\frac{\frac{1}{2}\rho V^2 C_S A}{mU} \cdot \left[c + \left(a - \frac{L}{2} + \frac{L C_Y}{C_S}\right)\right]}{c + \xi} \tag{1.43}$$

Where :
a is a distance from mass centre to front axle
b distance from mass centre to rear axle
nsp is a neutral steer point ("centre of tire forces")
m is a vehicle mass
CoP is a aerodynamic centre of pressure
c is a distance from nsp forward to mass centre = $(bC_r - aC_f) / (C_f + C_r)$
d is a distance from mass centre forward to CoP
U is a vehicle speed
V is speed of wind generated by the fans
r is a vehicle steady turning yaw rate response
S is a constant, aerodynamic side force
ζ is a moment arm proportional to the tire force yaw damping moment about the nsp(
$\zeta = (a+b)^2 C_f C_r / (C_f + C_r) m U^2$)
α is a slip angle
C_f effective total tire cornering stiffness of front axle
C_r effective total tire cornering stiffness of rear axle
C_Y is a aerodynamic yaw moment coefficient
C_S is a side force coefficient
L is a wheel base of a vehicle (*L= a+b*)
A is a frontal area of a vehicle
ρ is an air density

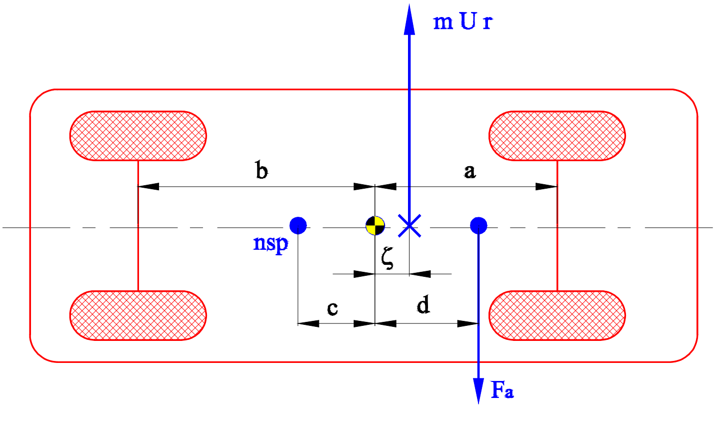

Figure 1.29 Static turning response of a vehicle under constant aerodynamic crosswind force [40]

Drag components

The aerodynamic drag changes due to loss of the pressure in the cooling duct, interference with the body, oblique flow to the front wheels. Hence, the drag can be separated on three main components which corresponds to the various region of a vehicle. This components can be farther separated as depicted in *figure 1.30*. Of course various region of a car will generate various amount of drag force. Understandably, it is impossible to eliminate such elements like external mirrors, windscreen washer, windscreen wiper, etc. in overall vehicle design. Every of those elements affects the flow around a vehicle and consequently generate drag.

Generally, the body drag is a consequence of constructional aspect of a car. For example, sharp corners of a fore body or inappropriate adjust slope of the backlight will significantly enlarge the body drag. The underbody of a vehicle generates some amount of additional drag. From aerodynamic point of view the underbody is usually considered to be extremely rough flat plane which is a source of drag occurrence, therefore large emphasis is given for smoothing it, which is very beneficial especially in sports cars [9].

The protuberance drag is provoked by any of protruding elements e.g. external mirrors and mud flaps and antennas as well as underbody roughness, wheel wells cavities, drip- rails windows recesses. Each of those elements affects the flow by generating vortices or provoking the flow transmission from laminar to turbulent which enlarges a frictional drag. In order to reduce the protuberance drag, all interfering elements are design to be as much aerodynamic as possible. For example the wheel are covered by a spats, and mirrors represents more aerodynamic shape. The internal drag represents rather negligible contribution in total drag as it reaches only 10 % [9].

As it can be seen in *figure 1.30* the internal drag is a sum of heating + ventilation system, engine cooling system and component cooling. However, only flow through a radiator represents significant influence on internal drag magnitude. Generally, the drag occurs due to drop of the stagnation pressure which farther results in turbulences. It has been determined that the internal drag can reach from 2% up to 10% of total drag.

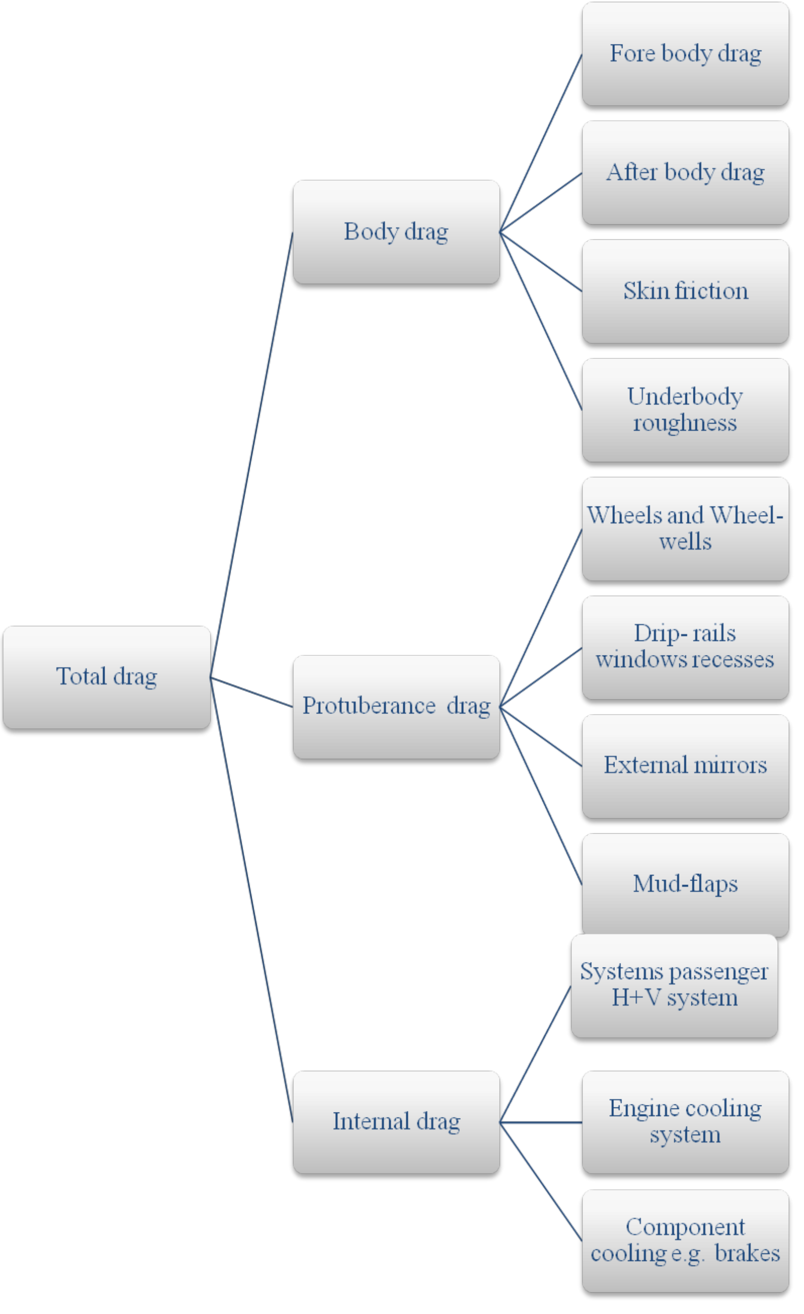

Figure 1.30 Breakdown of an aerodynamic drag

Aerodynamic aids

Generally, the aerodynamic aids are features which are responsible for improving aerodynamic performance of a vehicle i.e. reduce drag, improve stability and control lift force. The main aerodynamic aids are front spoilers which fixed on the front of the car reduces flow underneath a vehicle and therefore reduces positive lift, rear spoilers which regulates, in some extend, the flow separation and increasing the downforce on the rear. All of those are clearly exhibited specially in sports cars due to the fact that such designs are crucial in terms of obtaining best vehicle performance. In passenger cars aerodynamic aids represents smaller importance. Hence, in this case, installing this elements is targeted more in vehicles appearance then in aerodynamic performance. All types of “add on” devices must be designated for particular car model and tested by means of wind tunnel test. Adding random equipment may cause opposite effect than desired. Furthermore, passenger cars are simply too slow and mostly does not need aerodynamic aids. This same result is obtained by design of main body. For example, a vehicle’s nose “banded” downwards represent similar effect as front spoiler. The rear spoiler represent this same effect as higher boot lit [45]. Obviously, professional sport equipment is different from this available for normal passenger cars. For example the rear spoilers, and in some cases even front spoiler, are replaced by aerofoils.

The main objective of a front spoiler is to reduce the frictional drag under the car as well as decrease of lift (creating desirable downforce) on front of the car and increase of amount of air delivered to the cooling system. Generally, spoiler reduces a ground clearance of a car. Theoretically, in consequence the velocity at this point must increase, then farther downstream the flow slows down. Accordingly, as predicted by Bernoulli equation, the pressure under the spoiler is lower comparing with the rear of a car. The low pressure on frond end of the car results in generation of downforce. The spoiler effectiveness depends on its rake, its position with respect to the front axle, and the size of gap it creates. The relationship between spoiler design and lift and drag coefficient are depicted in *figure 1.31.*

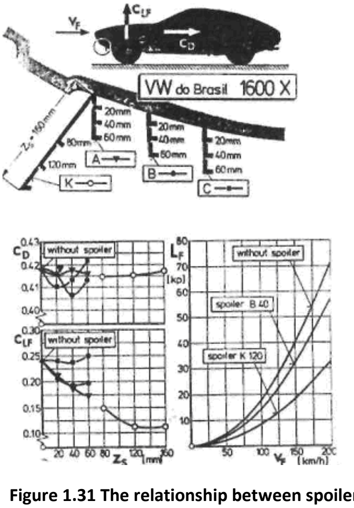

Figure 1.31 The relationship between spoiler design and lift and drag coefficient [9]

There are two kind of rear spoilers: roof spoiler and boot spoiler. The difference between them is that roof spoiler represent rather negligible influence on lift force. It is use as an elongation of a square back and upright hatch back roof. The roof spoiler ensures that airflow separates horizontally from the spoiler preventing flow of going downstream over the backlight. The cars of a fast back or hatch back has their spoilers located on the boot lit. Mostly it is slightly bended upwards end of the boot. In some cases however, the spoiler is “additional” part. The rear spoiler changes the flow trajectory leading it upwards resulting in higher pressure generation which pushes down the rear of the vehicle. Furthermore, spoilers tends to be reason of flow separation earlier upstream. Understandably, the shape, height and location of boot spoiler has influence on drag and lift performance. This dependence is depicted in *figure 1.32*. The drag is reduced only by a small size of spoiler. The drag increases with increase of size of spoiler. However larger spoilers produces larger downforce. During tests it has been determined that rear spoiler affect pressure under the vehicle which is crucial in terms of downforce generation. Spoilers selection is a compromise between obtaining better drag performance and obtaining greater downforce. Therefore it is to convenient describe a spoiler performance as a lift-to-drag ratio. The decrease of lift is follow by decrease of the trailing vortices, however it can be compensate by increase of pressure drag [45] and [9].

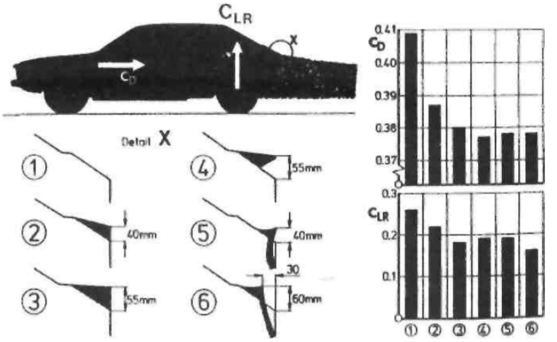

Figure 1.32 The rear spoiler influence on drag and rear axle lift [9]

1.3.6 Engine power designation

As we know from the previous chapter, vehicle have to overcome numerous resistance in order to start or change its movement. All those resistances can be represented as power of resistances. This is the key value for selecting the right engine for the vehicle.

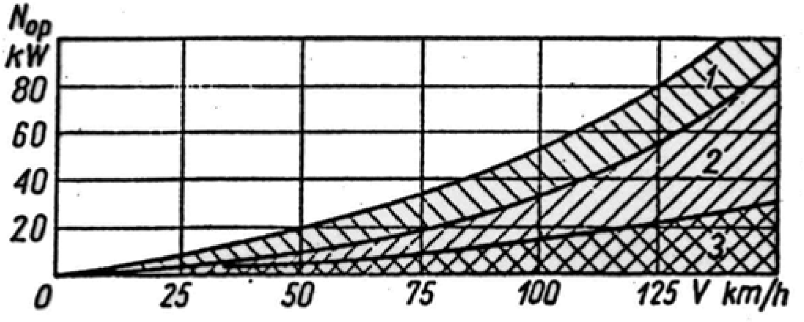

Figure 1.33 Resistance power in relation to speed: 1 – hill resistances, 2 – aerodynamic resistances, 3 – rolling resistances

For this situation resistance power and engine power can be shown as a vehicle power balance:

$$N_e \cdot \eta_m - N_h - N_a - N_r = 0 \tag{1.44}$$

N_e – engine power

η_m – powertrain efficiency

N_h - hill resistance power

N_a – aerodynamic resistance power

N_r – rolling resistance power

Relation between power resistances and engine as well as vehicle speed is presented on fig. 1.34.

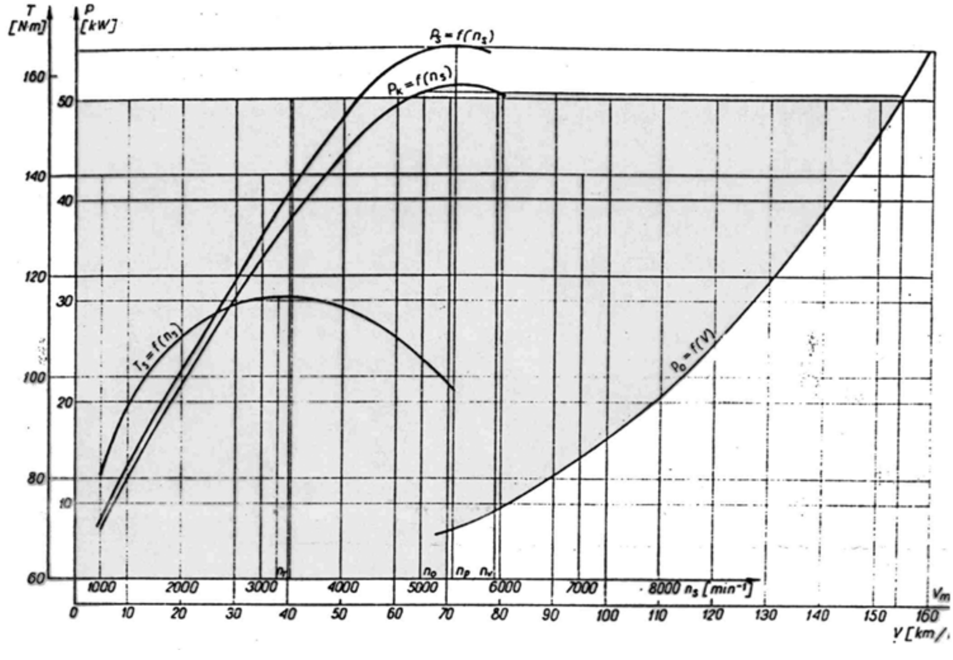

Figure 1.34 Power on wheels on a specific gear and resistances power in relation to engine speed and vehicle speed

1.4 Cornering

When the driving car enters a curve, significant centrifugal (inertia) forces starts to act horizontally on the vehicle mass centre (see *figure 1.35).* Furthermore, the centre of mass of whole vehicle can be considered as sum of front and rear axles mass centre. this forces cause wheels slip angle. The angle is different for the front and rear axle as it depends on for example local mass or tires condition (inflation thread wear). This, in consequence affects vehicles handling. The term handling is referred as ability of the vehicle to imply the responsiveness to the driver input or the easy of control [15]. The forces occurring during cornering causes various response like for example understeering and oversteering which are rather the properties of a vehicle as it mainly depends on a wheel slip angle.

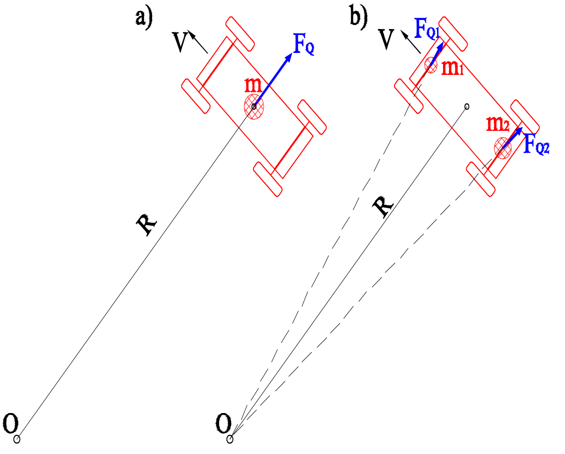

Figure 1.35 Schematic representation of forces occurring due to cornering [12]

1.4.1. Low speed cornering

During low speed cornering the wheels are under influence of rather negligible lateral forces, hence they roll without slip angle. If there is no slip angle on the rear axle, the centre on turn must be located on the projection of rear axle. Ideally, in case steering system presented in *figure.1.36*, the perpendicular from each of front wheels should meet at this same point (centre of turn). If they do not, then the wheels will influence each other trying to impose their trajectory. In consequence each wheel will suffer some scrap escalating wear and deterioration of handling [15]. The steer angles can be expressed by means of equations below. Those equations however, are only valid if the steering system has proper geometry i.e. all lines intersect at turn centre.

$$ct\alpha_1 = \frac{R-0{,}5b_k}{L} \tag{1.45}$$

$$ct\beta_1 = \frac{R+0{,}5b_k}{L} \tag{1.46}$$

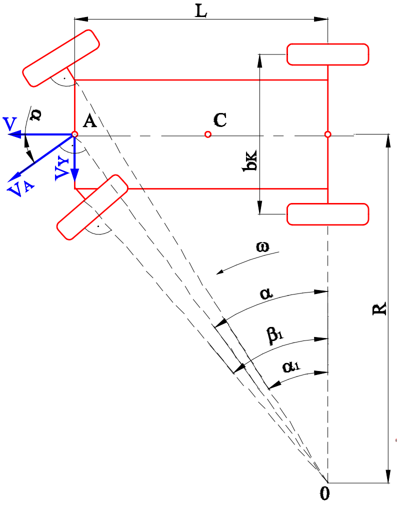

Figure 1.36 Geometry of a turning vehicle [12]

The average of both angles developed on each wheels is called as **Ackerman angle** and it can be determined by *equation 1.47.*

$$tg\alpha = \frac{L}{R} \tag{1.47}$$

Where:
L is the wheels base of the vehicle.
R is the turning radius.

As it can be seen the steering angle of both wheels are different. The difference however, is always constant. The magnitude of those angles is strongly dependant on the wheels base of a vehicle and the radius of the turn. The incorrect Ackerman geometry does not have great influence on handling and stability, but it causes significant tire wear. However it does affect the centering torques in staring system. The steering system with correct Ackerman geometry provides the steering torque to increase along with the steer angle in consequence facilitate the maneuvering. In contrast, if the Ackerman geometry is adjust incorrectly, the torque increases initially but can diminish at some point. This will result in steering more deeply into the turn and in consequence negatively affecting handling performance [15].

1.4.2. Highs speed cornering

As it was already mentioned steady state performance of a car travelling with low and constant speed through a curve with constant radius the inertia forces are negligible. However, when the car turns at moderate or high speed, the effect of centrifugal force can no longer be neglected. The centrifugal force acting on tires provokes occurrence of slip angle on each wheel. This is especially important due to the fact that the vehicle handling performance is greatly dependant on the relationship between slip angle on front and rear tires.

The schematic representation of a slip angle of a tire provoked by side force *(F_Y)* is depicted in *figure 1.37.* The side force causes the deflection of a tire. It should be emphasized that slip angle does not mean that the tire has slipped or slide. As it can be seen in *figure 1.37 b* the tire is divided on five parts ranging from 1 to 5. Furthermore, each part contacts the surface at corresponding points (A_1, A_2.....). Each part is shifted aside for a value of k=A_1B_1=A_2B_2=A_5B_5. In consequence the tire is deflected with angle of (δ). This causes the deflection of a whole wheel an accordingly the wheel without being turned provokes the change of vehicle trajectory [12].

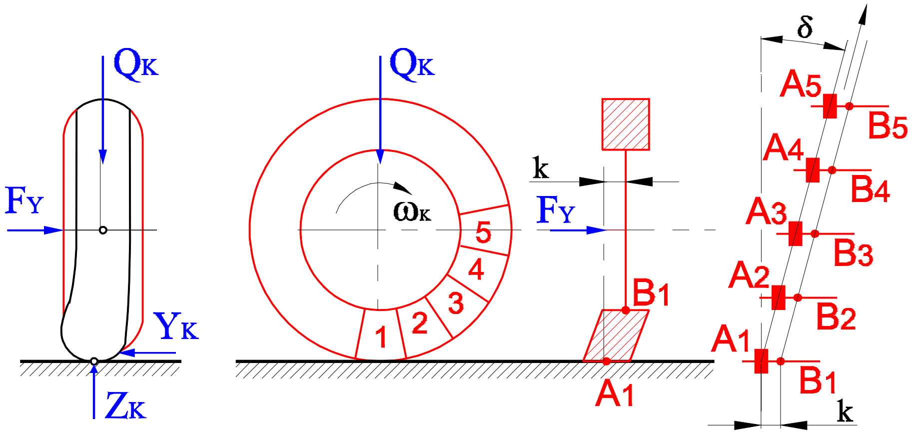

Figure 1.37 The slip angle provoked by side force FY [12]

This lateral force which is the main reason of slip angle occurrence is referred as "cornering force" when the camber angle is zero. The cornering force increases with respect to the slip angle. Moreover for small slip angles (approximately 5 degrees) the relationship can be considered as linear hence the force can be governed by *equation 1.48* [12], [15].

$$F_Y = C_\alpha \delta \tag{1.48}$$

Where:
C_α is a cornering stiffness.

δ is a slip angle.

Obviously the slip angle also has influence on the turn radius. Let's consider the turning vehicle with tires affected by slip angle presented in *figure.1.38.* As it can be seen both front (δ_1) and rear (δ_2)axle is under influence of slip angle. Each of slip angle provokes change of trajectory of both front and rear axle. This results in dislocating the centre of turn and in consequence of radius of tour which can be determined by *equation 1.49.*

$$R = \frac{L}{tg\,(\alpha - \delta_1) + tg\delta_2} \tag{1.49}$$

This equation can be farther simplify taken into consideration that slip angles are usually small. Then the equation is:

$$R \cong \frac{L}{tg\,\alpha + (\delta_2 - \delta_1)} \tag{1.50}$$

Where:

δ_1, δ_2 is the slip angle on front and rear wheels respectively.
α is the steer angle required to negotiate a turn.

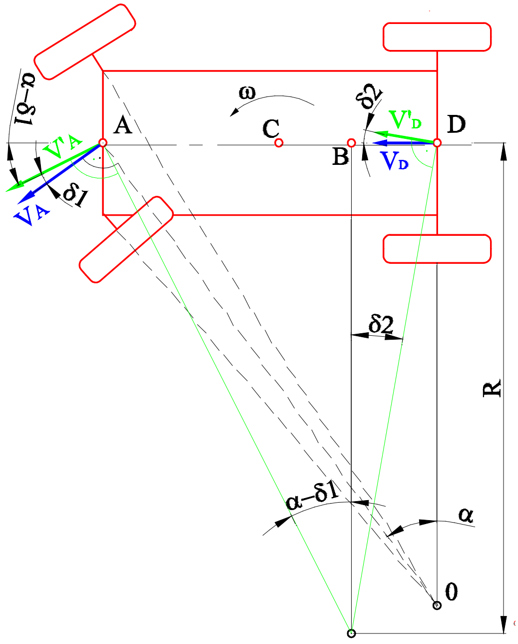

Figure 1.38 A turning vehicle with tires affected by slip angle [12]

As it was already mentioned the centrifugal force acts on centre of gravity of the vehicle (CG) however, the point of force interaction can be separated on front and rear axle in order to explained the deviations of trajectories caused by lateral force. The steer angle (α) strongly depends on the turning radius as well as the front and rear slip angles. It is also already known that the slip angles are dependent on the side forces acting on tire particular axis (front axle and rear axle) as well as the tires cornering stiffness. Hence the forces acting on front (F_{Yf}) and rear (F_{Yr}) wheels can be calculated using following equations:

$$F_{Yf} = 2W_f\frac{V^2}{gR} \quad (1.51)$$

$$F_{Yr} = 2W_r\frac{V^2}{gR} \quad (1.52)$$

Where:
W_f, W_r are normal load on each of the front and rear tires respectively under static conditions.
R is a turning radius.
G is the gravitational acceleration.

V is the speed of a vehicle.

Those forces causes the mentioned deformation of the tires, hence the slip angle of particular set of wheels can be determined by applying the forces motioned above into *equation (1.53)*[16].

$$\delta_1 = \frac{F_{Yf}}{2C_{\alpha f}} = \frac{W_f}{C_{\alpha f}}\frac{V^2}{gR} \tag{1.53}$$

$$\delta_2 = \frac{F_{Yr}}{2C_{\alpha r}} = \frac{W_r}{C_{\alpha r}}\frac{V^2}{gR} \tag{1.54}$$

Where

$C_{\alpha f}, C_{\alpha r}$ are the cornering stiffness of each of the front and rear tires, respectively.

The steer angle required to negotiate a turn is strongly dependant on wheel base, turning radius, velocity of a vehicle (lateral acceleration) and a understeer coefficient. The understeer coefficient is a function of weight distribution and tire cornering stiffens [16].

$$\alpha = \frac{L}{R} + \left(\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}\right)\frac{V^2}{gR} \tag{1.55}$$

$$\alpha = \frac{L}{R} + K_{us}\frac{V^2}{gR} \tag{1.56}$$

$$\alpha = \frac{L}{R} + K_{us}\frac{a_y}{g} \tag{1.57}$$

Where:
α is a steer angle
$K_{us}$is a understeer coefficient

a_y is a lateral acceleration

This is especially important as the handling strongly depends on the slip angles of the front and the rear wheels, hence on wheels cornering stiffness, and on the understeer coefficient. In general, the vehicle can behave in three different way during cornering or being subjected to side wind. Namely, it can oversteer, understeer or it can neutral steer.

Neutral steer $(\delta_1 = \delta_2)$, $K_{us} = 0$

The vehicle which represents this handling property is set to be “neutral steer” . The neutral steering occurs when the slip angles of both, front and rear wheels (δ_1, δ_2 respectively) are this same so the vehicle does not change its trajectory. In consequence the turn radius is not affected. Neutral steer vehicle will maintain its trajectory on turn with constant radius even during moderated speed. That means that the driver do not have to adjust correction of trajectory during driving.

Understeer $(\delta_1 > \delta_2)$, $K_{us} > 0$

The understeering occurs when the slip angle on front wheels is greater than on rear wheels. This results in displacement of centre of turn from point *(O)* to *(O')* (see *figure 1.39 a*). As it can be seen this displacement causes increase of a vehicle turn radius. In consequence, the car moves away from the desired trajectory.

The level of understeer of a car representing this kind of handling, can be quantified by parameter called characteristic speed [15]. The characteristic speed is defined as the speed at which the steer angle required to negotiate a turn equal to $2L/_R$. This speed can be determined by means of equation [16]

$$V_{char} = \sqrt{\frac{gL}{K_{us}}} \tag{1.58}$$

Oversteer $(\delta_1 < \delta_2)$, $K_{us} < 0$

The slip angle of rear wheels is greater than in the front. As it can be seen in *figure 1.39 b.* The greater slip angle on the rear wheels results in displacement of centre of turn to the point (*O'),* which creates smaller turn radius than the initial one (*O*)*.* In consequence, the car going through a turn with constant radius will deviate the trajectory towards the centre of turn. The oversteering is an undesirable handling property due to the fact that reduced turn radius is followed by increase of the centrifugal force which provokes farther deviation from the trajectory. Hence, escalates the danger of rear wheel drifting out. Another words, when the vehicle is accelerated on constant radius turn, the driver must adjust correction by decreasing steering angle. Comparing oversteer vehicle with the natural one, driving with this same speed and with this same steering angle applied, it appears that the turning radius of an oversteer vehicle is smaller than that of a neutral steer [16].

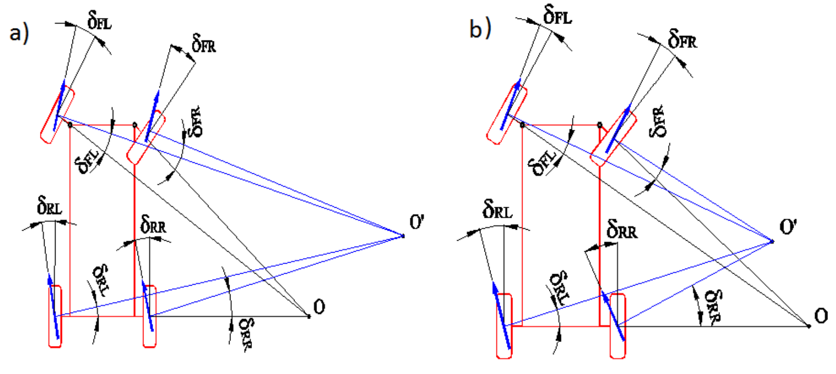

Figure 1.39 Vehicles handling characteristics. a) understeer, b) oversteer

The critical speed of the oversteer vehicle can be defined as the speed at which the steer angle required to negotiate any turn is zero. It should be emphasized that the understeer coefficient (K_{us}) has a negative value in cars representing oversteer behavior. Moreover, this speed also represents threshold value beyond which an oversteer vehicles becomes directional instable.

$$V_{Crit} = \sqrt{\frac{gL}{-K_{us}}} \tag{1.59}$$

As it can be seen the critical speed is highly dependant on a vehilce gemoetrical desig. For example long wheelbase have higher criticla speed comparing with short wheelbase vehicel.

The mass distribution of the vehicle can significantly affect, or in some cases, even alter the handling performance. For example, cars with greater mass on the front axle (front-engined) may provoke the understeer behaviour. In contrast, the cars with greater mass on the rear axle (rear-engined) may provoke oversteer behaviour. Furthermore, changing the load distribution of the vehicles (for instance by braking or accelerating during cornering) will also affect the handling performance. Accelerating on a turn will result in transfer of the longitudinal load from the front to the rear causing increase of the slip angle on front tires and decrease on the rear wheels. Accordingly, as already explained, the vehicle will exhibit understeer characteristics. During braking however, the load is transferred from the rear to the front, hence the slip angle on the rear tires increases whereas on front tires decreases. This may result in oversteer behaviour [15], [16].

As it has been already explained, the handling is strongly dependent on the cornering stiffness, thus the slip angle. Accordingly, the handling performance can be significantly strict in terms of tires and their properties. For example installing both type (radial-ply with bias-ply tires) of tires will alter original handling performance. Installing laterally stiff radial ply tires on the front and relatively flexible bias ply tires can change originally understeer vehicle into oversteer one. Moreover, the cornering stiffness strongly depends on the inflation pressure. Namely it decreases with respect to the a decrease of inflation pressure. Hence lowering the inflation pressure on the rear may result in altering the handling performance [16].

The lateral load transfer also affect the handling performance of the vehicle. The lateral load transfer from inside tire to outside one will increase the slip angle necessary to produce a given cornering force. Furthermore, the application of tractive effort turning cornering of a rear wheel drive vehicle corresponds to reducing of the cornering stiffness. Thus, this kind of vehicles mostly represents oversteering characteristics. Inversely, front wheel drive vehicle tends to exhibits lowered effective cornering stiffness. Therefore, this kind of vehicles usually are set to be understeer [16].

When the steer angle is applied, the vehicle respond with circum motion variability (i.e. lateral acceleration, yaw angle velocity, curvature). The ratio of those responses and the steer angle applied (α) is usually used for comparison of different cars in terms of their handling characteristics [16]

Lateral acceleration gain

The lateral acceleration arises when the car enters a curve and tends to push a vehicle out of its trajectory. The ratio of the steady state lateral acceleration and the steer angle is defined as the lateral acceleration gain. It is commonly used for evaluation of the steering response of a vehicle [16]. The lateral acceleration gain is given by:

$$G_{acc} = \frac{\frac{V^2}{gR}}{\alpha} = \frac{\frac{a_y}{g}}{\alpha} = \frac{V^2}{gL + K_{us}V^2} \qquad (1.60)$$

Where:

a_y is the lateral acceleration.

As it can be seen in equation above the understeer coefficient has a major influence on lateral acceleration gain. As it was already mentioned for the neutral steer vehicle the understeer coefficient is zero ($K_{us} = 0$). Hence , for neutral steer vehicle the lateral acceleration gain is dependant only on the wheel base and the velocity. For understeer vehicle the coefficient is positive ($K_{us} > 0$), hence the value of a acceleration gain will diminish. At very high velocities the first term in denominator is significantly smaller comparing with the second term. In consequence the lateral acceleration gain asymptotically approaches value of $1/K_{us}$ [16]. Therefore, the acceleration gain for understeer vehicle is always smaller then in case of neutral steer, as depicted in *figure 1.40.* Consequently, considering the negative value of understeer coefficient in an oversteer vehicle ($K_{us} < 0$) it should be notice that the lateral acceleration gain increases with respect to the increase of velocity as the denominator of *equation (1.60*) decrease with respect to speed. For a particular speed the denominator value reaches zero hence the value of lateral acceleration rises up to infinity [16].

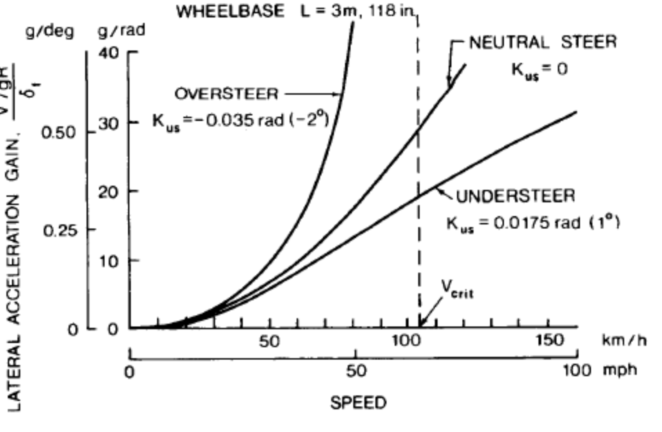

Figure 1.40 The comparison of lateral acceleration gain for different handling properties [16]

Yaw velocity gain

The yaw velocity gain (sometimes called yaw rate) is usually used for comparing different steering response of a car. The yaw velocity gain is defined as a ratio of a steady state yaw velocity and the steer angle and is express by *equation 1.61*[16]:

$$G_{yaw} = \frac{\Omega}{\alpha} = \frac{V}{L + \frac{K_{us}V^2}{g}}, \; {}^{deg}/_{s} \tag{1.61}$$

Where:

Ω is the yaw velocity during steady state cornering

In case of neutral steer the understeer coefficient is zero ($K_{us} = 0$), hence the yaw velocity gain increases linearly with increase of forwards speed. The understeer vehicle has positive understeer coefficient ($K_{us} > 0$), hence the yaw rate velocity increases with increase of forwards speed. At some particular speed however, the yaw velocity gain reaches a maximum. Usually the maximum point occurs for the characteristic speed. It is depicted in *figure 1.41.* An oversteer car has negative value of understeer coefficient ($K_{us} < 0$), hence the yaw velocity gain increases with increase of speed. At particular velocity the denominator reaches zero, and the yaw velocity approaches infinity. This threshold velocity is the critical velocity (see *figure 1.41*) [15] The oversteer vehicle tend to be more sensitive for steering impute comparing with natural steer one, whereas the neutral steer is more sensitive than the understeer one [16].

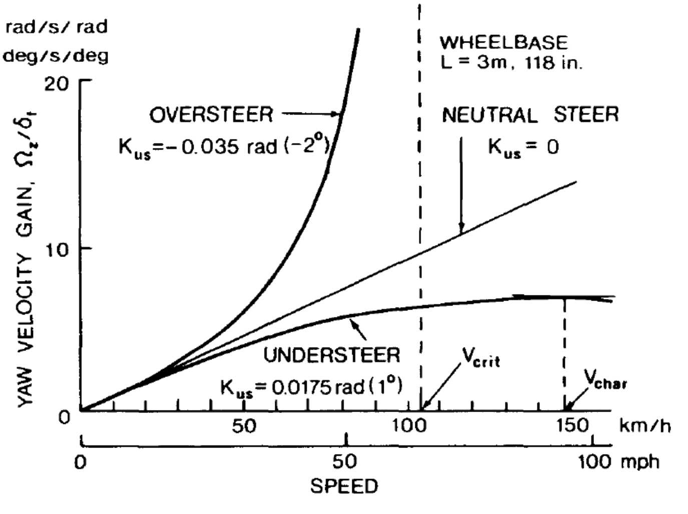

Figure 1.41 The comparison of yaw velocity gain for different handling characteristics [15]

<u>Curvature response</u>

The response characteristic of the vehicle can be quantified by means of ratio of the steady state curvature $1/_R$ and the steer angle. This ratio is express by equation below [16].

$$\frac{1/R}{\alpha} = \frac{1}{L + \frac{K_{Us}V^2}{g}} \quad (1.62)$$

The comparison of the curvature response for different handling characteristic is shown in *figure 1.42.* as it can be seen the neutral steer is expressed by a straight line. It is because the understeer coefficient for neutral steer cars is zero ($K_{us} = 0$), which makes the curvature response independent from speed. the understeer ($K_{us} > 0$), on the other hand, the curvature response decrease with respect to increase of the forward speed. The oversteer vehicles ($K_{us} < 0$), the curvature response increases with increase of speed. At particular velocity the denominator reaches zero, and the curvature response approaches infinity. This threshold velocity is the critical velocity. Another words when vehicle reach the critical velocity, the turning radius approaches zero and, in consequence the vehicle will spin without control [16].

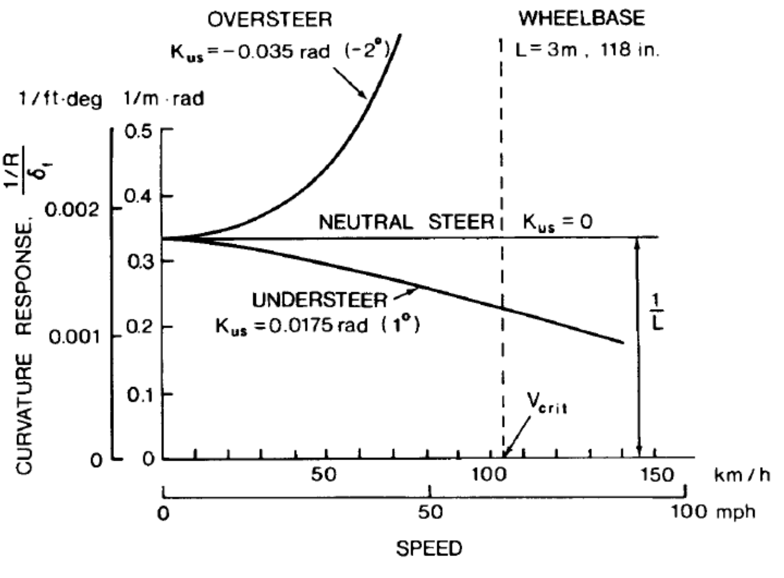

Figure 1,42 The comparison of the curvature response for different handling characteristic [16]

1.4.3. Suspension effect on cornering

As it was explained in previous subchapter, handling performance depends on the slip angle of the tires and on the mass distribution on the front and rear ends of a car. In general, if the front axle is more compliant comparing with the rear, then the lateral disturbance produces more slip angle on front, hence the vehicle is set to be understeer and it deviate outwards from the original path. Conversely, when the rear axle is more compliant then the front, the slip angle on the rear wheels is greater than on front. In consequence, the car is deviating towards centre on turn and is set to be an oversteer one [15].

Apart from the slip angle and weight distribution the suspension of the car has also significant influence on the vehicle handling. Any design factor can significantly affect vehicles response on cornering and even in some cases change the handling characteristics. generally, whether the car is under or oversteer depends on the position of the roll centre, camber angle, roll steer, lateral force compliance steer, aligning torque. All will be explained in this subchapter.

<u>Roll moment</u>

The centrifugal force caused by cornering pushes outwards the vehicle centre of gravity. This results in lifting wheels on the inner side, consequently the centre of gravity is displaced and the roll moment arises. This is depicted in *figure 1.43.* The handling performance of the vehicle depends on the roll moment distribution on front and rear axle. Greater roll moment on front results in understeer performance. Conversely, greater roll moment on the rear produces oversteer behaviour of the car. this characteristics can be altered by applying additional stabilizing device on one of the axle (e.g. stabilizer bar). Applying this on front will change the understeer vehicle into oversteer one [15]. The lateral stiffness of the suspension can be found using the *equation 1.63*

$$K_\phi = \frac{1}{2} K_s S^2 \tag{1.63}$$

Where:
K_ϕ is the roll stiffness of the suspension.
K_s vertical rate of each of the left and right springs.
S is the lateral separation between the springs.

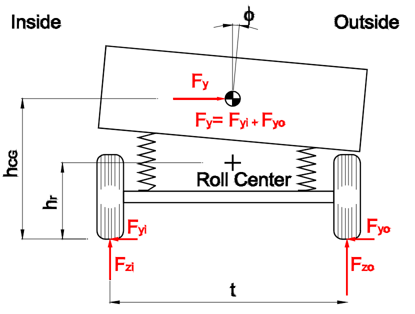

Figure 1.43 Force analysis of suspended vehicle during cornering [15]

Both, front and rear suspension must be considered in overall stability examination. The schematic representation of car and analysis of the forces to which the car is subjected is depicted in *figure 1.44.* As it can be seen the roll centres on the front and rear suspension is connected by so called roll axis. The roll centre is usually defined as the point at which the lateral forces are transferred from the axle to the sprung mass. Moreover, at this point the lateral forces produces the roll angle (ϕ). The roll moment about the roll axle can be determined by means of equation below [16]. (The explanation of the parameters of the equation are partially given on corresponding *figures 1.43 and 1.44*)

$$M_{\phi} = \left(W h_1 sin\phi + \frac{WV^2}{Rg} h_1 cos\phi \right) cos\varepsilon \tag{1.64}$$

Where:
F_{ZO} is the load on the outside wheel in the turn.
F_{Zi} is the load on the inside wheel in the turn.
F_y is the lateral force ($F_y = F_{yi} + F_{yo}$).
h_r is the roll centre height.
T is the tread (track width).
K_{ϕ} is the roll stiffness of the suspension.
ϕ is the roll angle of the body.

Furthermore, the moment about the roll axis is a sum of moments on roll centre on both, front and rear suspension. Those, in turn are expressed by a product of the roll stiffness on both suspensions. Hence:

$$M_{\phi} = M_{\phi f} + M_{\phi r} = \left(K_{\phi f} + K_{\phi r} \right) \phi \tag{1.65}$$

The equations can be rearrange in order to obtain the roll angle of the body. Hence [15]:

$$\phi = \frac{\frac{W h_1 V^2}{Rg}}{K_{\phi f} + K_{\phi r} - W h_1} \tag{1.66}$$

As it was already explained the moment about roll axis is a sum of moments on roll centre on both, front and rear suspension. The roll moment on front and rear suspension respectively can be found by [15]:

$$M_{\phi f} = K_{\phi f} \frac{\frac{W h_1 V^2}{Rg}}{K_{\phi f} + K_{\phi r} - W h_1} + \frac{W_f h_f V^2}{Rg} = \Delta F_{zf} t_f \tag{1.67}$$

$$M_{\phi r} = K_{\phi r} \frac{\frac{W h_1 V^2}{Rg}}{K_{\phi f} + K_{\phi r} - W h_1} + \frac{W_r r V^2}{Rg} = \Delta F_{zr} t_r \tag{1.68}$$

Where:

$$\Delta \mathrm{F}_{zf} = F_{zfo} - \frac{1}{2} W_f = \left(F_{zfi} - \frac{W_f}{2}\right) \tag{1.69}$$

$$\Delta \mathrm{F}_{zr} = F_{zro} - \frac{1}{2} r = \left(F_{zri} - \frac{W_r}{2}\right) \tag{1.70}$$

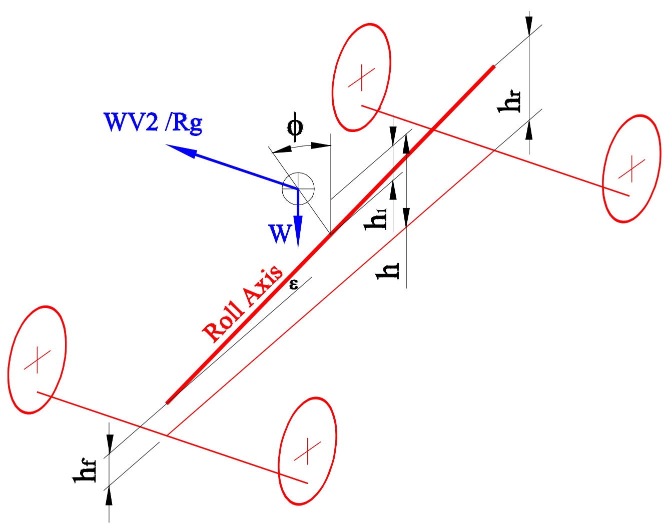

Figure 1.44 Forces analysis of the full vehicle during Steady-state cornering analysis [15]

In order to avoid oversteer behaviour, cars produced nowadays tends to have greater roll stiffness on front. Therefore, the stabilizer bars are fixed only on front or on both front and rear in this same car. Applying the stabilizer bar only on the rear can result in undesirable oversteering.

Knowing the roll moments, roll angle and the tries properties the load on inner and outside wheels respectively, can be obtained by equation:

$$F_{yf} = \left(C_{\alpha f} - 2b\Delta \mathrm{F}_{zf}^{2}\right)\delta_f = \frac{W_f V^2}{Rg} \tag{1.71}$$

$$F_{yr} = (C_{\alpha r} - 2b\Delta \mathrm{F}_{zr}^{2})\delta_r = \frac{W_r V^2}{Rg} \tag{1.72}$$

Camber angle

The wheel inclination outwards from the body is called the camber angle. The camber angle produces lateral force called camber thrust. Camber changes handling characteristics of both dependent as well as independent suspension as a result of body roll and normal camber change in jounce or rebound. Camber angle affect both dependent and independent suspension, however in case of independent suspension caber angle is in greater importance[15].

The impact of camber angle on handling performance is dependent on the tries as well, however it produces much less lateral force than slip angle. Bias-plied tires represents lower camber stiffness. *Figure 1.45* shows the schematic representation of cornering car and provoked by it camber angles and roll of the vehicle. The total camber in this case is given by [15]:

$$\gamma_g = \gamma_b + \phi \tag{1.73}$$

Where:
γ_g is the camber angle with respect to the ground.
γ_b is the camber of the wheel with respect to the body.
ϕ is the roll angle of the vehicle.

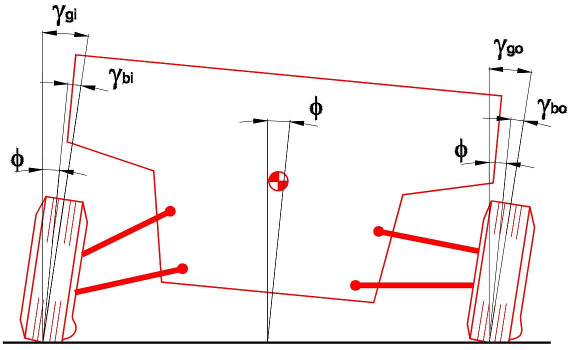

Figure 1.45 The schematic representation of cornering car and provoked by it camber angles and roll of the vehicle [15]

The camber angle, like slip angle is dependent on the wheels performance (inflation pressure, cornering stiffness) hence those two factor has to be added to each other during determination of the lateral force.

$$F_Y = C_\alpha \delta \cdot C_\gamma \gamma \tag{1.74}$$

Furthermore, the slip angles of front and rear wheels extended by the camber factor is:

$$\delta_f = \frac{W_f}{C_\alpha} a_y - \frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma f} \partial_\phi}{\partial_\phi \partial_{a_y}} a_y \tag{1.75}$$

$$\delta_r = \frac{W_r}{C_\alpha} a_y - \frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma r} \partial_\phi}{\partial_\phi \partial_{a_y}} a_y \tag{1.76}$$

Hence the steer angle is given by:

$$\alpha = \frac{L}{R} + \left[\left(\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}} \right) + \left(\frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma f}}{\partial_\phi} - \frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma r}}{\partial_\phi} \right) \frac{\partial_\phi}{\partial_{a_y}} \right] \frac{V^2}{gR} \tag{1.77}$$

Therefore, the understeer coefficient for this case is:

$$K_{cam} = \left(\frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma f}}{\partial_\phi} - \frac{C_\gamma}{C_\alpha} \frac{\partial_{\gamma r}}{\partial_\phi} \right) \frac{\partial_\phi}{\partial_{a_y}} \tag{1.78}$$

Roll steer

When the vehicle enters a turn, the suspension kinematics provokes roll of the body and in consequence the suspension roll of front and rear wheels, with total steer effects. The roll steer can be defined as the change of steer of two wheels in this same axle when the body rolls. The roll steer is mainly denoted for the rigid axles. In general, the roll steer affects handling performance due to lag of the steer input, caused by the roll of the sprung mass [15], [17].

The understeer gradient for this case can be obtained by analogues reasoning as in previously explained cases of handling.

$$K_{roll\,st} = \left(\varepsilon_f - \varepsilon_r\right)\frac{\partial \phi}{\partial_{a_y}} \tag{1.79}$$

Where:

$\varepsilon_f, \varepsilon_r$ are the roll steer on front and the rear wheels respectively.

Generally, the roll steer coefficient describes the behaviour of the car during turning. Let's consider a car entering a left turn. The inertia force causes body roll outwards, hence towards right side. In this case, the roll steer coefficient is considered as positive when the wheels steer towards left. In consequence, when the positive value of the roll coefficient is on the front wheels, the car will be understeer. Conversely, if the rear wheels reflects positive roll coefficient, the vehicle will be oversteer.

Lateral force compliance steer

The comfort of driving requires application of special materials (i.e. rubber, soft bushing) in the suspension. Indeed soft materials reduces arise of the undesirable noise vibration and harshness, but can also be a reason of lateral compliance in the suspension. This in turn can provoke unpredictable, hence undesirable handling performance [16]. In case of solid axle, the compliance steer can be represented as a rotation abut yow centre of the vehicle. This is depicted in *figure.1.46* [15]. As it can be seen the oversteer occurs for yaw centre moved forward on the rear axle (*figure 1.46 a*). It is because it the rubber material deflect under load of inertia forces and in consequence changes the initial trajectory towards outside of the turn. accordingly, when the yaw centre on the rear axle is located rearwards the vehicle will understeer (*figure 1.46* b). The situation changes diametrically when the front axle is under consideration. When the yaw centre is located rearwards, the car will oversteer whereas understeer for yaw centre located forward [15].

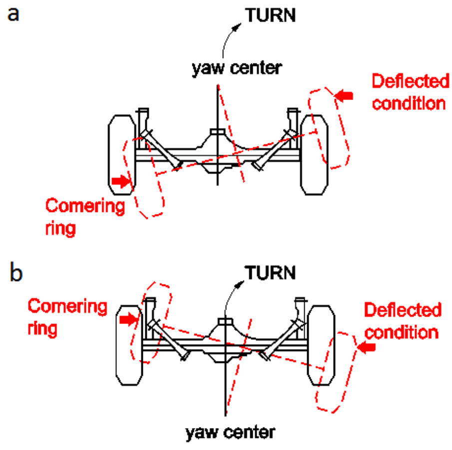

Figure 1.46 Handling change due to lateral compliance in the suspension. a)oversteer, b)understeer [15]

The understeer coefficient for the lateral compliance is dependent on the steer angles produced on the front and the rear wheels and of course on the load of the axles and lateral acceleration. Hence the coefficient is given by:

$$K_{com} = \frac{\alpha_{cf}}{F_y} W_f - \frac{\alpha_{cr}}{F_y} W_r \tag{1.80}$$

Align torque

The align torque arises due to the fact that the lateral force produced by a tire is not developed on centre of tire but after some distance behind it. This distance is called *pneumatic trail* and is usually denoted by (*P*). Since the align torque depends on the tire characteristics (i.e. their cornering stiffness), the wheel base and the vehicle load, the understeer coefficient for this case is given by:

$$K_{at} = W \frac{P}{L} \frac{C_{\alpha f} + C_{\alpha r}}{C_{\alpha f} C_{\alpha r}} \tag{1.81}$$

As it can be seen in the equation above the value of the understeer coefficient for align torque can only be positive, since the cornering stiffness cannot be negative. In consequence , only understeer behaviour is possible in this case [15].

1.4.4. Experimental measurement of understeer gradient

Very comprehensive deliberation concerning measurement of understeer gradient is published in publication referred under [16]. The Society of Automotive Engineering in the publication referred under [18] defines understeer gradient as "the quantity obtained by subtracting the Ackermann steer angle gradient from the ratio of the steering wheel angle gradient to the overall steering ratio". This handling characteristics is experimentally measured by simulation of a car entering a turn on a skid pad. The steer angle, velocity of the tested vehicle and yaw velocity are measured during the tests. Generally, the skid pad is a flat, paved area. The measurement contains three types of tests: the constant radius test, the constant forward speed test, the constant steer angle test. [16]

Comparing all those tests, the constant speed test is most correspondent to the real road behavior as the driver mostly tries to maintain constant speed during turning. The constant radius test, however does not require as many measuring equipment as it is in case of other tests due to the fact that experimental measurement of lateral acceleration or yaw velocity is not necessary. The constant steer angle test is easiest to perform [16].

<u>Constant radius test</u>

During this test the vehicle drives through a curve with known, constant radius but with various speeds. The steering angle and corresponding lateral acceleration is observed. The lateral acceleration of turning vehicle can be deduced in two ways. It can be determined either by means of accelerometer installed on the vehicle or, knowing the velocity of the car and the radius of turn the lateral acceleration can be calculated using the following equation:

$$a_y = \frac{V^2}{Rg} \tag{1.82}$$

The results obtained during tests are plotted on diagram $\left(\alpha = f(a_y)\right)$. An example of such diagram is presented in *figure 1.47.* as it can be seen the handling behavior of the vehicle dependents on the slope of the curve. Another words, the slope of the curve corresponds to the value of understeer coefficient. The slop of the curve can by find by [16]:

$$K_{us} = \frac{d\alpha}{d\left(\frac{a_y}{g}\right)} \tag{1.83}$$

Accordingly, if the steer angle required to maintain the vehicle on desired trajectory is this same for all forwards speeds, then the vehicle represents neutral steer characteristic. On *figure 1.47* the slope representing neutral steer characteristics is zero ($K_{us} = 0$).

The understeer arises when the slope of the curve presented in *figure 1.47*, and thus understeer coefficient is positive ($K_{us} > 0$). Consequently, the vehicle is oversteer when the slope, hence also the understeer coefficient is negative ($K_{us} > 0$).

Farther investigation of *figure 1.47* can lead to conclusion that the from practical point of view the vehicle will exhibit oversteer characteristics for high lateral acceleration, whereas at low lateral accelerations the vehicle will understeer. This is because the value of understeer coefficient varies with operating conditions (e.g. load transfer, tractive effort, nonlinearity of tires behaviour, etc.).

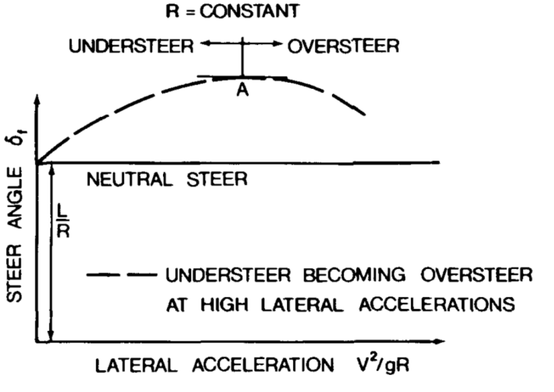

Figure 1.47 Handling characteristics of test with constant radius [16]

Constant speed test

During this test the vehicle is driven with constant forward speed but with various radius. Besides the steer angle and the velocity, the radius must be determined for each conditions as well. The radius can be found either by measuring the lateral acceleration or yaw rate [15]:

$$R = \frac{V^2}{a_y} = \frac{V}{r} \tag{1.84}$$

Where:
V is the forward speed
a_y lateral acceleration

r is the yaw rate

The results obtained during the tests are plotted on a diagram as shown in *figure 1.48.* the steering behavior can be estimated based on the slope of the steer angle-lateral acceleration curve. The slope of the curve, can be found by:

$$\frac{d\alpha}{d\left(\frac{a_y}{g}\right)} = \frac{gL}{V^2} + K_{us} \tag{1.85}$$

The handling characteristics of a vehicle can be estimated based upon diagram presented in *figure 1.48*. The vehicle will be natural steer when the understeer coefficient is zero ($K_{us} = 0$). Therefore, the line will be constantly increasing and straight.

For understeer vehicle the understeer coefficient is positive, therefore the slope of the steer angle-lateral acceleration curve is greater comparing with the line representing neutral steer (see *figure 1.48*). The vehicle is oversteer when the understeer coefficient is less than zero ($K_{us} > 0$). Thus the slope of the curve must be less than that for the neutral steer response.

When the slop of the curve is zero, then the oversteer vehicle is operating at critical speed. In consequence it is on verge of directional instability [16].

$$\frac{gL}{V^2} + K_{us} = 0 \tag{1.86}$$

$$V^2 = \frac{gL}{(-K_{us})} = V^2_{critical} \tag{1.87}$$

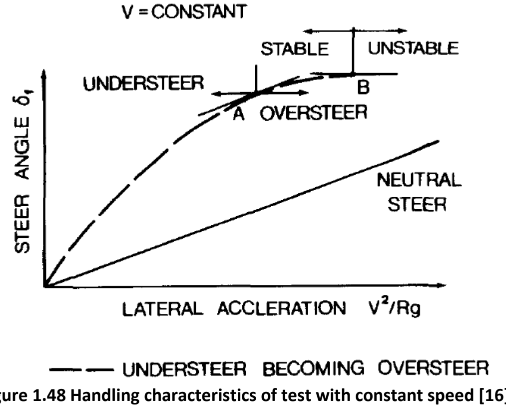

Figure 1.48 Handling characteristics of test with constant speed [16]

Constant steer angle test

This test consist on measuring the lateral acceleration of a vehicle driven with fixed steering wheel angle at various forward speeds. Knowing the forward speed and the lateral acceleration it is very easy to determine the curvature of the turn ($1/R$). The curvature shell be plotted on a chart against the lateral acceleration as depicted in *figure 1.49.* Similarly to the test explained earlier the handling performance of the vehicle can be estimated based on the slope of the curvature-lateral acceleration curve. Accordingly, the slope of the curve can be developed [16] . Hence:

$$\frac{d\left(\frac{1}{R}\right)}{d\left(\frac{a_y}{g}\right)} = -\frac{K_{us}}{L} \qquad (1.88)$$

As it was already pointed out the understeer coefficient for the neutral steer vehicle is zero ($K_{us} = 0$). Therefore the slope of the curvature-lateral acceleration curve is also zero. Thus, the neutral steer vehicle is expressed by straight horizontal line. For the understeer car the understeer coefficient is negative ($K_{us} < 0$) which provokes the slope of the curvature-lateral acceleration curve to be positive as indicated in *figure 1.49.* Conversely, an oversteer vehicle is characterised by positive understeer coefficient ($K_{us} > 0$). Therefore, slope of the curvature-lateral acceleration curve is also positive [16].

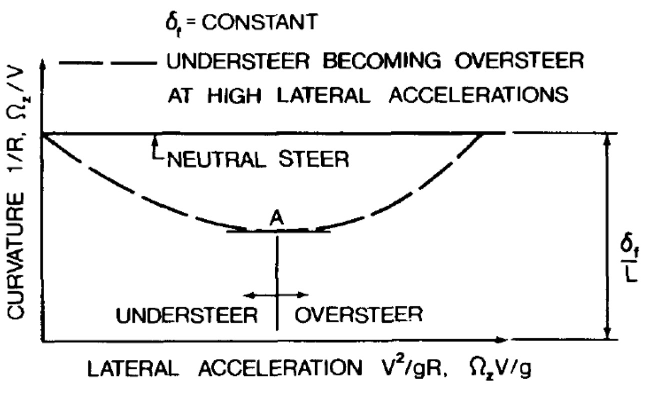

Figure 1.49 Handling characteristics of test with constant steer angle [7]

1.5 Braking

Braking is possible due friction between tyre-road and brake disc-brake pad (for disc-brakes). Longitudinal weight transfer which happens during braking results in increase in front wheel load and reduction in rear wheel load. When the load (force) is multiplied by the friction coefficient it can be seen that front wheels can transfer a lot higher braking force. This is why brake systems brake-force distribution in not equal, but usually 60%-70% front to 40-30% rear.

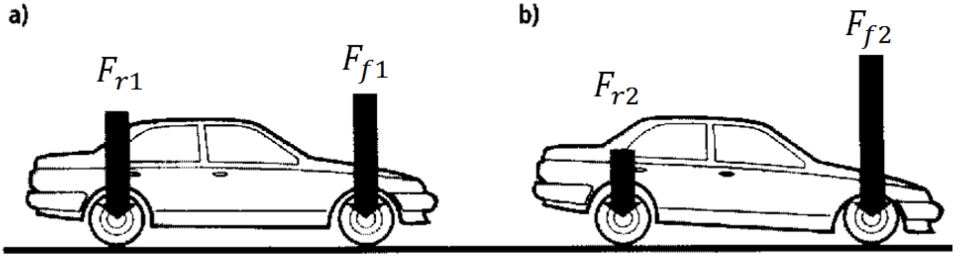

Figure 1.50 Load transfer during braking

During braking the tyre slips. Friction coefficient is slip depended and this relation can be seen on Fig.1.51

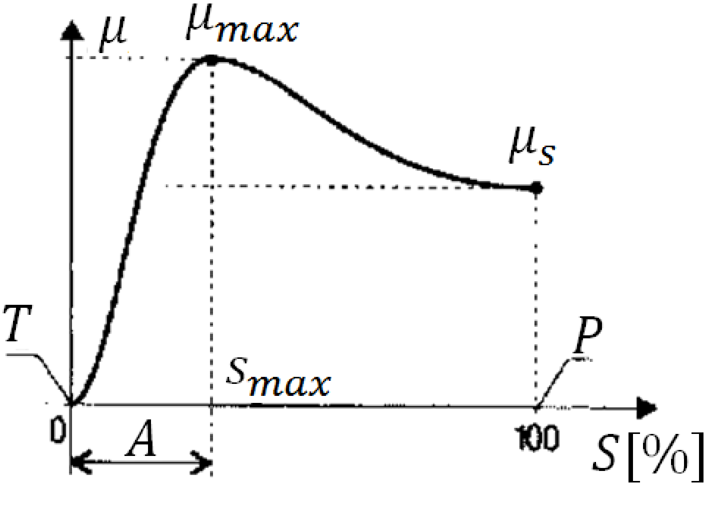

Figure 1.51 Friction coefficient in relation to tyre slip

μ_{max} – maximum friction coefficient

μ_s – friction coefficient of a locked wheel (100% slip)

A – slip value at which max friction coefficient is reached – usually about 20%

Braking process is defined as all action from the moment of noticing an obstacle by the driver to a standstill position of the car (fig. 1.52).

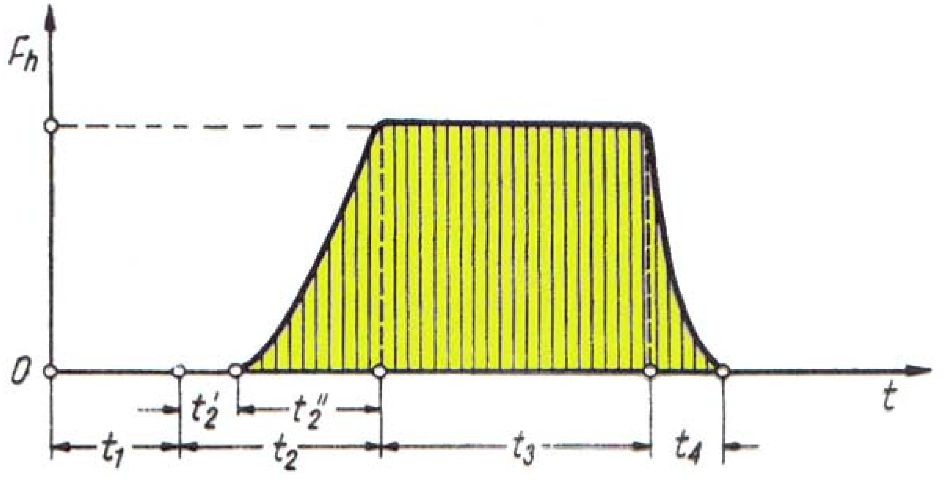

Figure 1.52 Braking process in time

t_1 – drivers reaction time

t_2 – time from pushing the brake pedal to moment when full brake power is applied on the disc/drums. It can be divided into first period – when all kinds of slack in the system is eliminated, and second period – when the pressure is building up to the maximum value.

t_3 – main braking time

t_4 – brake release time

Braking forces

Transferable force

Brake system should be suited to a specific vehicle taking into consideration weight, longitudinal weight transfer, tires used, etc. Therefore a maximum transferable brake force should be calculated to design a system capable of using all that traction and giving the driver enough “feel”.

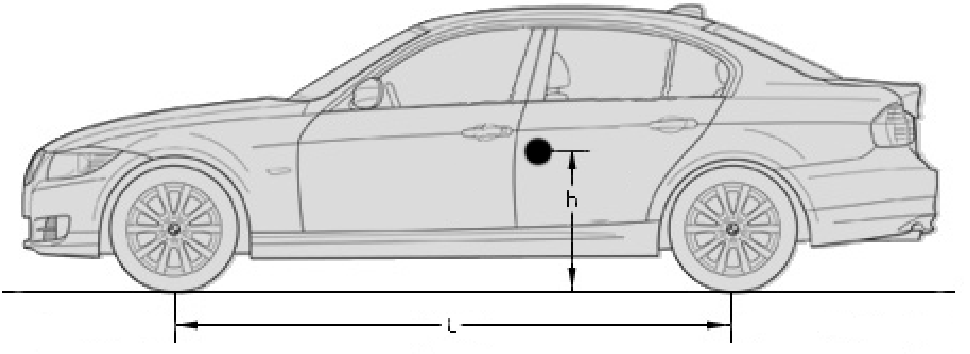

Figure 1.53 Centre of gravity vertical position [17]

First step is to calculate dynamic weight transfer during braking.

$$W_t = W \cdot \frac{a}{g} \cdot \frac{h}{l} \tag{1.89}$$

Next the dynamic loads on front and rear axis can be calculated.

$$W_{F\,braked} = W_{F\,static} + W_t \tag{1.90}$$

$$W_{R\,braked} = W_{F\,static} - W_t \tag{1.91}$$

The maximum transferable force by the tire is calculated by multiplying force acting on the wheels by the friction coefficient. Typical value of the coefficient found in real life friction between tire and tarmac is around 0.7-0.8.

$$F_{max\,F} = W_{F\,braked} \cdot \mu \tag{1.92}$$

$$F_{max\,R} = W_{R\,braked} \cdot \mu \tag{1.93}$$

Brake ratio can be calculated by:

$$\alpha = \frac{F_{max\,F}}{F_{max\,R}} \tag{1.94}$$

To prevent wheels locking during braking the following ruled need to be met:

$$F_f \leq F_{max\,F} \tag{1.95}$$

$$F_r \leq F_{max\,R} \tag{1.96}$$

$$\frac{\alpha}{1+\alpha} \cdot W \cdot \frac{a}{g} \leq \mu \cdot \frac{W}{l} \cdot \left(W_{F\,static} + h \cdot \frac{a}{g}\right) \tag{1.97}$$

$$\frac{a}{g} = \gamma_F \leq \frac{\mu \cdot W_{F\,static}}{\frac{\alpha}{1+\alpha} \cdot l - \mu \cdot W_{F\,static}} \tag{1.98}$$

$$\frac{\alpha}{1+\alpha} \cdot W \cdot \frac{a}{g} \leq \mu \cdot \frac{W}{l} \cdot \left(W_{R\,static} + h \cdot \frac{a}{g}\right) \tag{1.99}$$

$$\frac{a}{g} = \gamma_R \leq \frac{\mu \cdot W_{R\,static}}{\frac{\alpha}{1+\alpha} \cdot l - \mu \cdot W_{R\,static}} \tag{1.100}$$

γ_F – front wheels traction utilization coefficient

γ_R – rear wheels traction utilization coefficient

$$\zeta = \frac{\mu \cdot W_{F\,braked}}{\mu \cdot W_{R\,braked}} = \frac{W_{F\,braked}}{W_{R\,braked}} \tag{1.101}$$

$$\zeta_0 = \frac{W_{F\,static}}{W_{R\,static}} \tag{1.102}$$

ζ – front to rear traction ratio

ζ_0 - front to rear traction ratio for no acceleration

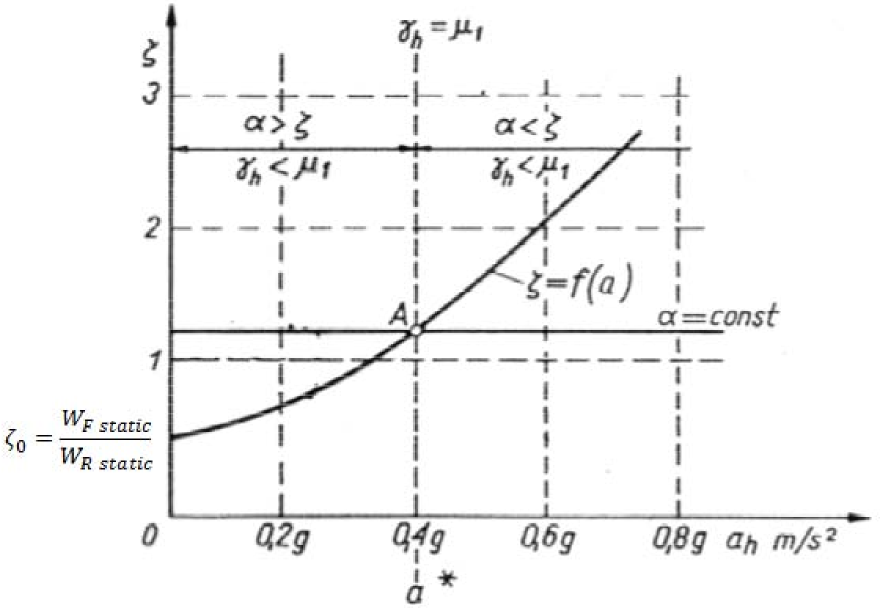

Figure 1.54 Front rear load ration in relation to acceleration (deceleration)

Pressure in the circuit

To calculate the required pressure in the system several values are required:

- Maximum transferable brake force
- Tyre diameter
- Brake disc diameter
- Caliper piston area

$$p = \frac{F_{max} \cdot \frac{R}{r}}{A \cdot \mu_{bp}} \tag{1.103}$$

p – circuit pressure

$\frac{R}{r}$ – wheel diameter to disc brake centre diameter ratio

A – brake pads area

μ_{bp} – friction coefficient between brake pad and brake disc

Master cylinder bore designation

Driver during braking pushes the brake pedal with the force of 500 N. Pedal box design defines brake pedal ratio. Brake pedal acts like a leverage and actuates master cylinder with higher force. This leverage is called brake pedal ration or brake system mechanical ratio.

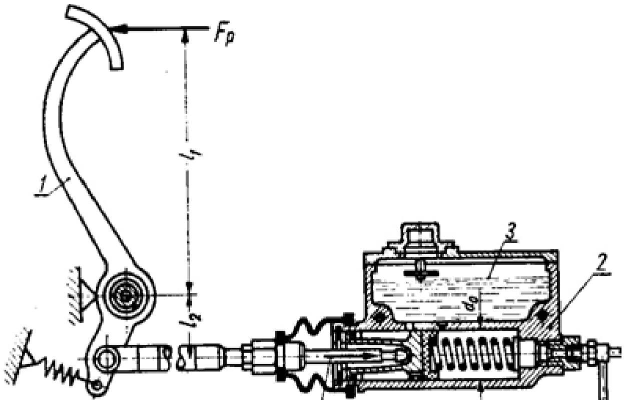

Figure 1.55 Brake pedal geometry

It can be calculated by:

$$i_m = \frac{l_1}{l_2} \tag{1.104}$$

Therefore the force acting on the master cylinder is equal to

$$F_{mc} = F_p \cdot i_m \tag{1.105}$$

Knowing the pressure in the circuit and force on the master cylinder, bore size can be calculated.

$$p = \frac{F_{mc}}{A_{mc}} = \frac{F_{mc}}{\frac{\pi \cdot d_{mc}{}^2}{4}} \tag{1.106}$$

$$d_{mc} = \sqrt{\frac{4 \cdot F_{mc}}{\pi \cdot p}} \tag{1.107}$$

Brake system ratios

There are two main ration describing brake system. First is the mechanical ratio determined by brake pedal design, mentioned above. Second is the hydraulic ratio – which is determined by master cylinder and caliper pistons area. It can be calculated as the area of all caliper or drum pistons in the circuit to master cylinder bore area.

$$i_h = \frac{A_p}{A_{mc}} \tag{1.108}$$

To acquire ratio of the whole system – both ratios need to be multiplied.

$$i_0 = i_m \cdot i_h \tag{1.109}$$

1.6 Rollover

Rollover accident is characterized by great severity and serious injuries of the occupants of the vehicle. Rollover may occur on both flat and leveled surface. This kind of accident can arise due to exceeding the threshold level of the lateral acceleration or due to impacting or driving onto obstacle. However, the last scenarios will not be discussed in this text.

When a vehicle negotiates a turn, driving with speed which provokes exciding of a maximum threshold lateral acceleration, the inner wheels will be lifted up while the outer wheels will be subjected to greater load. This begins the rollover process [19]. Furthermore, the rollover can be escalated by occurrence of side wind as well as by the suspension effect.

1.6.1 Quasi- static rollover of a rigid vehicle

When the vehicle enters a curve the lateral acceleration arises acting on the centre of gravity. In response the lateral forces act in the ground plane to counterbalance the lateral acceleration. In consequence those forces crates moment on the vehicle which rolls the car outside a turn.

In order to analyse the rollover problem during cornering consider a vehicle presented in *figure 1.56.* What is shown there is the rigid vehicle driving through a turn. The rigid vehicle means that the deflection of the suspension and tires will not be considered during the analysis. this kind of mechanism is most rudimentary in terms of balance of forces acting on a vehicle during cornering. Accordingly, the moments about the point *A* in the *figure 1.56* yields*:*

$$F_Q h_s - F_{in}\frac{t}{2} + F_{out}\frac{t}{2} = 0 \tag{1.110}$$

Note that F_{in} and F_{out} represents a sum of forces acting of both inner and outer wheels respectively.

The rollover is considered to be initiated when both inner wheels loses contact with the road $(F_{in} = 0)$. Therefore the outer wheels are subjected to the total weight of the vehicle $(F_{out} = Q)$. Hence:

$$F_Q h_s - Q\frac{t}{2} = 0 \tag{1.111}$$

Where Q is the weight of the vehicle (Q=mg)

Bearing in mind that the centrifugal force (F_Q) is represented by product of the mass of the vehicle and the lateral acceleration it is possible to estimate the relationship of radius of a turn which a car is negotiating as well as the velocity and the constructional aspect of a given vehicle.

$$F_Q = ma_y = m\frac{V^2}{R} \tag{1.112}$$

Consequently, the maximum safe velocity also called a threshold velocity, for a given car negotiating a particular turn can be found by:

$$V_{max} = \sqrt{\frac{Rg\frac{t}{2}}{h_s}} \tag{1.113}$$

As it can be seen the rollover of the vehicle is mainly dependent on its construction (i.e. the wheel base and the height of the centre of gravity). For example a sports car with wide wheel base and low of CoG height will rollover after reaching 1,2-1,7g. Conversely, a heavy truck will rollover at 0,4-0,6g of lateral acceleration.

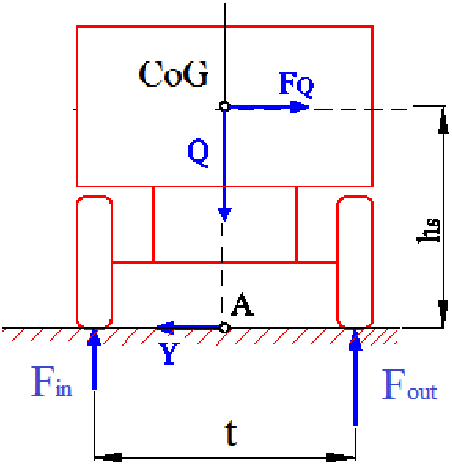

Figure 1.56 Forces acting on a rigid car while cornering [12]

1.6.2 Quasi-static rollover of a suspended vehicle

Considering rollover of a suspended vehicle compliances in the tires and suspensions can no longer be neglected due to the fact that this overestimates the rollover threshold of a vehicle. The lateral forces provokes displacement of the centre of gravity towards outside of the turn. This displacement reduces the moment arm on which is a base for gravity forces resisting the rollover [15].

The schematic representation of a suspended vehicle driving through a curve is shown in *figure 1.57.* Taking the moments about the pointer roll centre (*R*) gives (after reference [12]):

$$F_Q h_{dis} + Qs_{\varphi} = M_{sp} \tag{1.114}$$

Where the spring moment in passenger car can be found by:

$$M_{sp} = \left(2k_1 \frac{t_{s1}^2}{4} + 2k_2 \frac{t_{s2}^2}{4} + k_{s1} + k_{s2}\right)\varphi = K_{\varphi}\varphi \tag{1.115}$$

Bearing in mind that the lateral shift of the roll centre can be expressed by an roll angle and the arm of the centrifugal force (F_Q) with aid of the equations above it is easy to estimate the roll angle:

$$\varphi \approx \frac{F_Q\, h_{dis}}{K_\varphi - Q h_{dis}} \tag{1.116}$$

As it is in previous case, the rollover is considered to be initiated when the inner wheels lose contact with the surface $(F_{in} = 0)$ as a consequence of the lateral the acceleration. Accordingly, considering the sum of moments which attempts to roll the vehicle outside the turn:

$$F_Q - Q\left(\frac{t}{2} - s_\varphi\right) = 0 \tag{1.117}$$

The equation above can be further developed in order to determine the threshold velocity of a suspended car:

$$\frac{Q}{g}\frac{V^2}{R} h_s - Q\frac{t}{2} + Q h_{dis} \frac{\frac{QV^2}{g\,R} h_{dis}}{K_\varphi - Q h_{dis}} = 0 \tag{1.118}$$

Hence the threshold velocity is:

$$V_{max} = \sqrt{tRg \frac{K_\varphi - Q h_{dis}}{2 h_s (K_\varphi - Q h_{dis}) + 2 Q h_{dis}}} \tag{1.119}$$

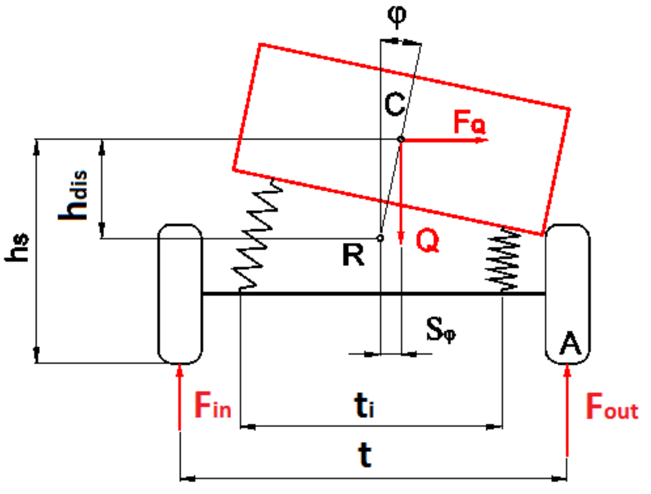

Figure 1.57 Roll reaction on suspended vehicle [12]

Comparing the suspended and rigid vehicle one can notice that the threshold velocity of a rigid car is higher than in case of suspended car. Consequently the maximum safe lateral acceleration of the suspended vehicle is reduced for approximately 5%. Of course, this is greatly dependant on stiffness of the suspension and height of the centre of gravity. The sports cars with low centre of gravity will endure greater lateral acceleration then the higher vehicles without rollover. Furthermore, the solid axles also reduces the lateral shift due to the fact that this kind of suspension tends to have a high roll centre. Conversely the independent suspension reflects greater lateral shift than the solid axle. It is due to reduced distance from centre of gravity to the roll centre [15], [12].

1.6.3 Transient rollover

The transient rollover means that the vehicle roll varies with time. Another words, the transient rollover includes the rapidly changing lateral acceleration. The estimation of ability to rollover consist on performing simulation of different driving conditions. The essential properties of the vehicle in terms of maneuvering response, can be found with aid of a simple roll model. This model consist investigates the vehicle response to suddenly applied lateral acceleration in the nature of step input. Moreover, this method is used as a representative of a transient occurring when a vehicle drives through a turn with applied brakes and, when the brakes are released, is subjected to the sudden return of cornering forces. It cal also simulate the effect of sliding friction [15].

In order to estimate more precisely response of vehicle roll behaviour it is necessary to develop more comprehensive vehicle models which simulates both yaw and roll response. This method is therefore called yaw-roll models. The yaw motion of a vehicle (e.g. during slalom driving) produces the lateral acceleration which in consequence provokes the roll motion. The roll motion alters the yaw responds due to the fact that it affect the tire cornering forces. The effect of the phase interval enables the car to yaw and change the direction when the lateral acceleration is moderated. The lateral acceleration is obtained by spreading the acceleration over a longer time period [15]. The passengers cars therefore, will exhibit the lack of responsiveness in transient cornering. In contrast large cars will be less influenced in terms of responsiveness comparing with small cars. It is because the time lag increases with increase of the wheelbase. Four-wheel-steer cars exhibits increased responsiveness in transient cornering. It is because this kind of steering invariably steer the rear wheels in this same direction as the front wheels. This in consequence, eliminates the phase lag [15].

The most frequently occurring type of rollover [20], and therefore requiring special modelling, arises when the skidding vehicle hits an object (e.g. a curb or soft ground). In consequence the vehicle trips into rollover. As it was already explained, the height of the centre of gravity (CoG) has significant influence on the rollover. Cars will high CoG are in greater rollover danger comparing with cars with low CoG. The impact of the vehicle against the obstacle (a curb) produce the rotation of the vehicle which in turn provokes the kinetic energy to arise. This energy is proportional to half of the moment of the inertia forces of the sprung and unsprung mass about their rotation points, multiplied by their re times their respective rotational velocities squared. At this same time the lifting of the CoG produces additional energy which is proportional to the mass times the increase in CoG height. If the sum of those energy exceeds the threshold value, the rollover will occur [15]. It should be remembered however, that this reasoning neglects additional energy input or dissipation from wheel contact with the ground. The energy storage or dissipation in the tires and suspension also is not taken into consideration. Therefore, considering relationships mentioned above it is assumed that that all of the kinetic energy is transformed to potential energy when lifting the CoG to the rollover point. It appears that the weight of the vehicle as well as the suspension stiffness and damping properties is rather in secondary importance in this type of transient rollover[15] .

1.7 Tyre-road collaboration

The wheel is the last but very important component of the power train system in a vehicle. Its parameters have crucial influence on the movement conditions. A pneumatic-tired wheel consists of a rim and an inflated tire. The parameters of these two components decide on driving conditions.

The movement of the vehicle is possible thanks to the very important phenomena of adhesion between the tire and the road. The wheel is subjected to a complex state of loading. It is required to transfer the vertical, longitudinal and lateral forces as well as rotational moments between the road and the axle. Additionally the wheel rim is desired to ensure good carrying away of the heat from the brakes and safe connection between the wheel and the hub.

The wheels differ in sizes and shapes, allowable loading and maximum speed, kind of tire used. However the phenomena connected with the movement of the wheel are similar and common for all kinds of on road vehicles.

1.7.1 Wheel reference system

To easy describe the interactions between the tire and the road it is convenient to attach a Cartesian coordinate system to the wheel as it is shown in the Figure 1.30. There are some reference elements that should be introduced to help with the analysis.

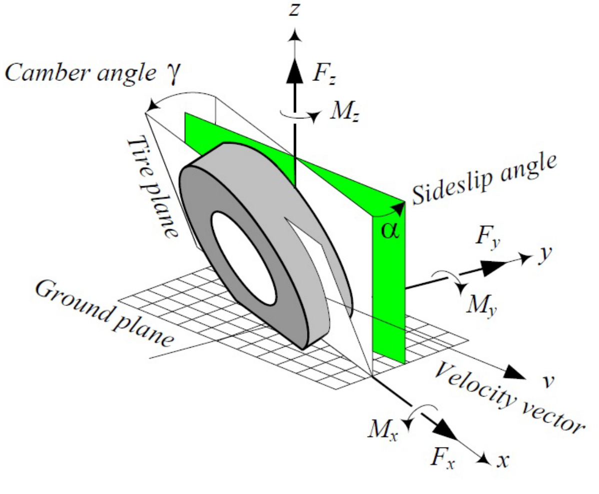

Figure 1.58 Wheel reference system

- Tire plane – a plane that would be the effect of narrowing the wheel to a flat disc, it is the symmetry plane of the rim
- Ground plane – a plane coincident with the flat road surface
- Velocity vector – points in the direction of movement forward
- Sideslip angle α – the angle between the velocity vector v and the x-axis measured about the z-axis.
- Camber angle γ – the angle between the tire plane and the vertical plane measured about the x-axis.
- Forces and moments acting on the wheel can be decomposed along x, y and z axes.
- Longitudinal force $\boldsymbol{F_x}$ (also called *forward force*) - it is force acting along the x-axis. It is positive while the car is accelerating and negative during braking.
- Normal force $\boldsymbol{F_z}$ (also called *vertical force* or *wheel load*) – it is a vertical force, perpendicular to the ground plane. The resultant normal force is pointed upwards.
- Lateral force $\boldsymbol{F_y}$ – it is tangent to the ground and perpendicular to both F_x and F_z. It is considered positive if pointing towards the vehicle.
- Roll moment $\boldsymbol{M_x}$ (also called *bank moment*, *tilting torque* or *overturning moment*) – it is a moment about the x-axis and tends to turn the tire about the x-axis.
- Pitch moment $\boldsymbol{M_y}$ (also called *rolling resistance torque*) – it is a lateral moment about the y-axis which occurs when positive tends to turn the tire about the y-axis and move forward.
- Yaw moment $\boldsymbol{M_z}$ (also called *aligning moment*, *self aligning moment* or *bore torque*) – it is an upward moment about the z-axis which exist when positive tends to turn the tire about the z-axis.

1.7.2 Tire

The tire is an element of the wheel which is crucial in transmission of the forces between the road and the vehicle. Its characteristics strongly influence the area of contact, the longitudinal, lateral and vertical forces. It should also meet certain requirements which, in case of passenger vehicles and light commercial vehicles, can be divided into following groups in order of importance.

a) **Driving safety.** As tire failure by high speed driving is extremely dangerous and can lead to fatal accidents this is the most important requirement. To ensure safety the tire must fit tight to the rim flanges so it will not slip during movement on various road surfaces and be hermetically sealed in order to prevent the air escape. Very important features are also low sensitivity to overloading and puncture resistance.

b) **Handling.** The tire should provide proper driving conditions by ensuring, among others, high coefficient of friction in various operating environments, good cornering stability and quick response to steering movements.

c) **Riding comfort.** This is very desirable feature from the point of view of the customer. It includes low noise, good damping properties and easy steering during parking or driving.

d) **Service life.** Durability in aspect of long-term use and high-speed stability.

e) **Economy.** It includes the influence of the tire characteristics on the costs connected with the operation of the vehicle, for example purchase price or fuel consumption.

f) **Environmental compatibility.** It includes the influences of the tire production, operation and disposal on the environment – tire noise, energy consumption of the manufacturing process, possibility of recycling.

Tire types

There are two main types of tyres.

Diagonal ply tyres. As concerning the passenger vehicles in industrialized countries this type of tyres has been mostly replaced by the radial ply tyres due to their advantages. However diagonal ply tyres (or *cross ply tires*) are still used as spare tyres (low requirements concerning durability), in motor cycles (beneficial inclination of the wheels), racing cars (lower moment of inertia) and agricultural vehicles.

Radial ply tyres. They consist of circumferential belt of cords, which provides necessary stiffness; radial cords (or *flanks*) and two bead cores (reinforcement cables made of steel or plastics). Radial ply tyres are now used in passenger cars because of the following advantages:

- Significantly higher mileage
- Greater load capacity at lower weight
- Lower rolling resistance
- Better aquaplaning properties
- Better wet-braking behaviour
- Increased ride comfort at high speeds

Almost all of the tyres used in passenger cars are tubeless. This is caused by the easier and faster mounting and the ability of the inner lining to self-seal small gashes in the tyre.

Tread structure

Tyre tread structure is presented on fig. 1.31. Let's mention main elements of this structure.

a) **Sipes** – narrow slits forming gaps with 0.3 to 1.5 mm wide in the tread block. They improve traction on wet roads and snow and participate in the removing of the water from under the tyre.

b) **Tread block** – elements forming the tread. Their task is to ensure the good traction properties of the tyre.

c) **Tread ribs** – form the crown of the tyre, often provided with crosswise grooves.

d) **Groove** – concave part of the tyre tread. Pattern of grooves, their shape and size play a fundamental role in the quality of the tyre. Grooves improve braking effectiveness and steering properties of the tyre. The depth and shape of grooves determine the level of noise emitted by the tyre when vehicle is in motion.

e) **Dimples** – improve the cooling of the tyre.

f) **Tread grooving** – create the space necessary to remove water from the tires on wet surfaces.

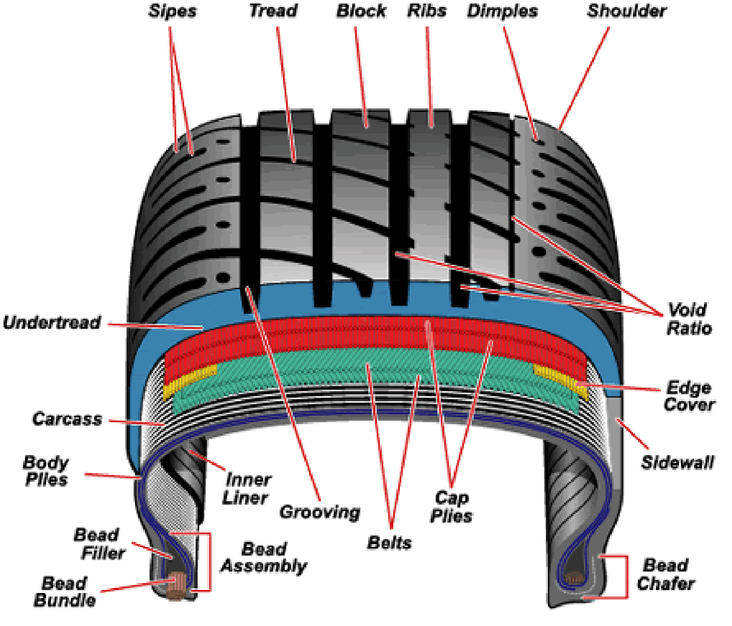

Figure 1.59 Tyre tread structure

Tire stiffness

The deformation behaviour of tires to the applied forces and moments are the first important tire characteristics in tire dynamics. They are described by stiffness coefficients. Calculating the tire stiffness is generally based on experiment and therefore, they are dependent on the tire's mechanical properties. There are three principal stiffness coefficients used in the description of the behaviour of the tire.

Longitudinal stiffness – describes the deflection of the tire due to the acting of the longitudinal force

$$C_x = \frac{dF_x}{dx} \tag{1.120}$$

Deflection shape is shown below (fig. 1.60):

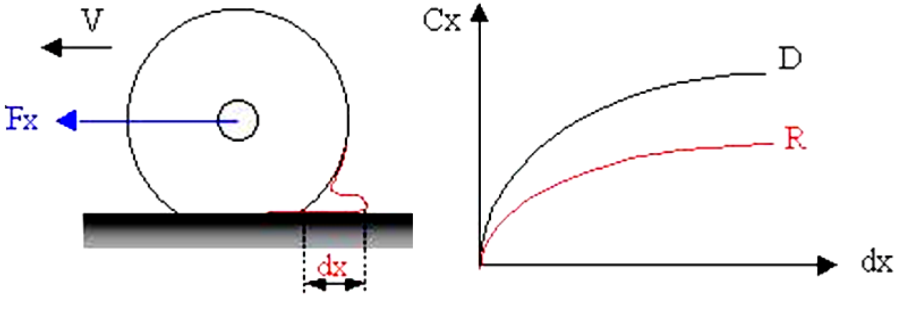

Figure 1.60 Longitudinal stiffness

Lateral stiffness – describes the lateral deformation of the tire

$$C_y = \frac{dF_y}{dy} \tag{1.121}$$

Deformed shape is shown below (fig. 1.61):

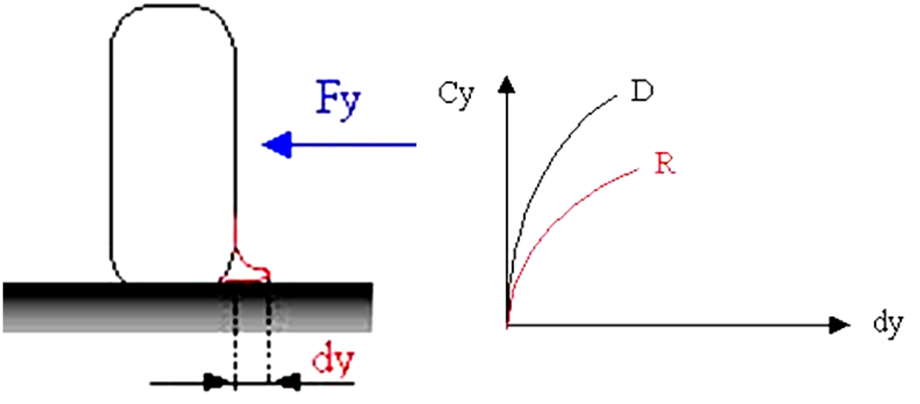

Figure 1.61 Lateral stiffness

Torsional stiffness – describes the angular deflection of the tire subjected to the yaw moment (M_z)

$$C_\alpha = \frac{dM_z}{d\alpha} \tag{1.122}$$

Deflection shape for torsion is shown on fig. 1.62:

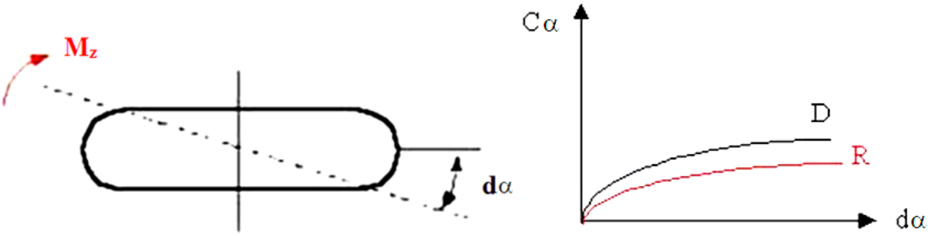

Figure 1.62 Torsional stiffness

1.7.3 Slip of the wheel

The loads on the wheel and the circumferential elasticity of the tire cause the slip of the wheel. It appears all time during the movement of the vehicle and in a consequence the path of the wheel centre is different from the multiplication of its circumference and number of rotations.

When a moment acts on the wheel it causes the compression of the part of the tire coming into the area of contact and simultaneously the stretching of the part leaving the area of contact.

It results in the deformation of the running tread and as a consequence the linear displacement of the wheel centre is smaller than the displacement in free rolling.

The slip increases with the increase of the moment and circumferential forces acting on the wheel. The relative speed on the circumference of the wheel v_W is therefore larger than the linear speed of the axle, which is equal to the speed of the vehicle v_V. The slip of the wheel is defined as:

$$s = \frac{v_W - v_V}{v_W} \qquad \text{when } v_W > 0 \tag{1.123}$$

The equation is however different in the case of deceleration of the vehicle:

$$s = \frac{v_V - v_W}{v_V} \qquad \text{when } v_V > 0 \tag{1.124}$$

The slip of the wheel can change from 0 to 1. In the ordinary movement conditions it is several percent and causes abrasion of the tire tread and the surface of the road.

1.7.4 Forces and moments loading the wheel

When a moving vehicle is concerned there are three basic states of loading of its wheels. If a wheel is pulled by a horizontal force acting in its axis is called *a free rolling wheel*. The movement of a *drive wheel* is caused by a rotational moment about its axis, transferred from the engine. If there is a moment which tends to slow the motion the wheel is *braking*.

1.7.5 Free rolling wheel

The distribution of forces acting on the rolling wheel is presented in the fig. 1.63. The surface of the road is considered to be rigid which means that its deformation can be neglected during the analysis. The system of equations of equilibrium for this state of loading is following.

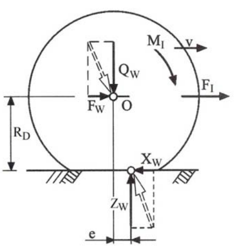

Figure 1.63 Loading acting on a free rolling wheel

- The sum of forces along the horizontal direction

$$X_W = F_W + F_I \tag{1.125}$$

- The sum of forces along the vertical direction

$$Z_W = Q_W \tag{1.126}$$

- The sum of moments about the wheel centre

$$X_W R_D - Z_W e + M_I = 0 \tag{1.127}$$

where:

M_I, F_I – moment and force of inertia of the wheel with mass m_K

$$M_I = -I_W \frac{d\omega_W}{dt} \tag{1.128}$$

$$F_I = -m_W \frac{dv}{dt} \tag{1.129}$$

I_K – mass moment of inertia of the wheel in respect to its axis of rotation

The reactions – tangent circumferential X_W and normal Z_W, are components of the force acting from the road surface on the wheel which is marked with the dashed line. The force F_W needed for the wheel to roll comes from the car forcing the movement. The sum of the moments leads to an equation:

$$X_W = \frac{e}{R_D} Z_W - \frac{M_I}{R_D} = \frac{e}{R_D} Z_W + \frac{I_W}{R_D} \frac{d\omega_W}{dt} \tag{1.130}$$

For the movement with the constant velocity:

$$X_W = \frac{e}{R_D} Z_W = F_W \tag{1.131}$$

The last equation allows to determine the lowest force needed to roll the wheel.

$$F_W = \frac{e}{R_D} Q_W \tag{1.132}$$

1.7.6 Drive wheel

The torque from the engine is transferred by the power train components to the drive wheel. Therefore there is a rotational moment acting on the wheel that tends to move it forwards. The force F_W has the opposite sense than in the previous situation. It is the reaction force from the vehicle being pulled by the drive wheel. The distribution of forces acting on the drive wheel is presented in the Fig 1.64 The system of equations of equilibrium is following.

$$X_W = F_W - F_I \tag{1.133}$$

$$Z_W = Q_W \tag{1.134}$$

$$M_W = X_W R_D + Z_W e - M_I \tag{1.135}$$

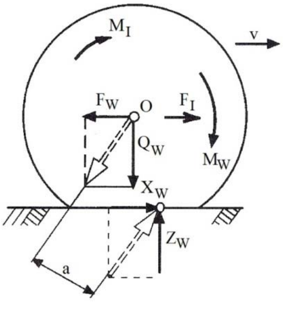

Figure 1.64 Loading acting on a drive wheel

The pair of forces on the arm a marked with the dashed line gives the moment balancing the action form the driving moment M_W and the moment of inertia M_I. It is convenient to introduce the driving force F_D.

$$F_D = \frac{M_W}{R_D} \tag{1.136}$$

The system of equations leads to:

$$F_D R_D = X_W R_D + Z_W e - M_I \tag{1.137}$$

The driving force in general situation can be expressed by:

$$F_D = X_W + \frac{e}{R_D} Q_W + \frac{I_W}{R_D} \cdot \frac{d\omega_W}{dt} \tag{1.138}$$

In the case of steady motion ($\dot{v} = 0$):

$$F_D = X_W + \frac{e}{R_D} Q_W \tag{1.139}$$

From the equation of moments it is possible to calculate the maximum values of driving moment and driving force of the wheel.

$$M_{Wmax} = X_{Wmax} R_D + Q_W e \tag{1.140}$$

$$F_{Dmax} = X_{Wmax} + \frac{e}{R_D} Z_W \tag{1.141}$$

These equations lead to an important statement. The moment M_W transferred by the wheel and the driving speed F_D are limited not only by the power of the drive system but also by the value of the tangent circumferential reaction X_W. This force depends on the properties of the tire and the road surface.

1.7.7 Braking wheel

During the process of braking the velocity decreases rapidly so the influence of the inertia force and moment is significant. The presence of the braking moment M_B leads to the force distribution shown on the Fig. 1.65.

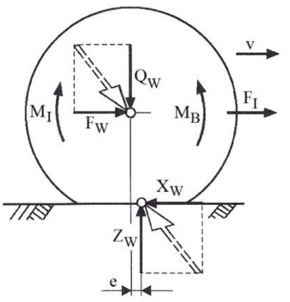

Figure 1.65 Loading acting on a braking wheel

The system of equations of equilibrium is following.

$$Z_W = Q_W \tag{1.142}$$

$$X_W = F_W + F_I \tag{1.143}$$

$$M_B - M_I = X_W R_D - Q_W e \tag{1.144}$$

Braking moment divided by the wheel radius gives the braking force F_B.

$$F_B = \frac{M_B}{R_D} \tag{1.145}$$

From the system of equations it can be calculated.

$$F_B = X_W - \frac{e}{R_D} Q_W - \frac{I_W}{R_D} \frac{d\omega_W}{dt} \tag{1.146}$$

Using the equations it is possible to calculate maximum values of the braking moment and force that can be obtained on the wheel.

$$M_{Bmax} = X_{Wmax} R_D - Z_W e + M_I = X_{Wmax} R_D - Q_W e - I_W \frac{d\omega_W}{dt} \tag{1.147}$$

$$F_{Bmax} = X_{Wmax} - \frac{e}{R_D} Q_W - \frac{I_W}{R_D} \frac{d\omega_W}{R_D} \tag{1.148}$$

1.7.8 Wheel on compliant soil

The previously described situations referred to the movement on the solid road. The loading conditions are different when the deformation of the surface of the road is visible in the form of ruts. In this case it is possible to neglect the tire deformation as much smaller.

For a free rolling wheel in such conditions and movement with constant speed, forces are shown on the Fig. 1.66:

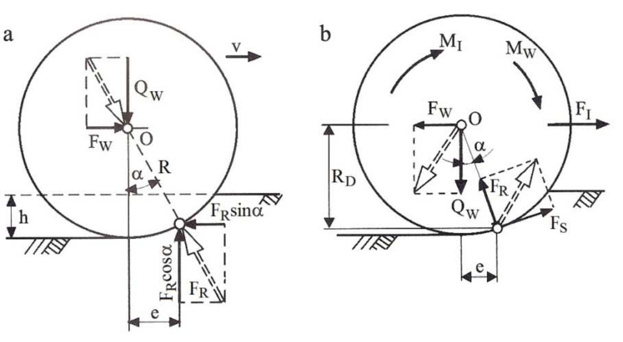

Figure 1.66 Loading acting on a wheel moving on a compliant soil

This phenomena can be described with the following system of equations of equilibrium:

$$F_R sin\alpha = F_W \tag{1.149}$$

$$F_R cos\alpha = Q_W \tag{1.150}$$

$$eF_R cos\alpha - RF_R sin\alpha = 0 \tag{1.151}$$

From the above the force needed to move the wheel can be obtained.

$$F_W = \frac{e}{R} Q_W \tag{1.152}$$

The force distribution on the drive wheel is very important. In this case the equations of equilibrium are defined for movement with variable speed.

$$M_W + M_I = F_S R_D \tag{1.153}$$

$$F_S cos\alpha - F_R sin\alpha = F_K - F_B \tag{1.154}$$

$$F_S sin\alpha + F_R cos\alpha = Q_K \tag{1.155}$$

The driving moment then equals:

$$M_W = F_S R_D - M_I \tag{1.156}$$

And for constant speed:

$$M_W = F_S R_D \tag{1.157}$$

This relationship shows that the wheel on the compliant soil can transfer the moment with a value limited by the tangential circumferential reaction F_S. This force is connected with the resistance to shearing of the soil and the force of internal friction between its particles. The larger the size of the wheel the higher the moment it can transfer.

1.7.9 Sideslip of the wheel

When a lateral force, from for example curvilinear movement or a crosswind, acts on the wheel the phenomena of sideslip appears. The lateral force *F* cause a reaction in the area of contact (*cornering force Y*) and a deflection of the tire. Subsequent parts of the tire getting into contact with the road surface deform and move in the lateral direction. As a consequence the direction of the translational motion is tilted in the respect of the plane of rotation of the wheel by the angle α, called a *sideslip angle*.

As a result of the tire deformation the reaction force is shifted by a distance *e* from the axis of the wheel and makes an aligning moment *M* which equals:

$$M = Ye \tag{1.158}$$

This moment turns the wheel in the real path of the movements and so tends to decrease the sideslip angle. The value of the angle increases with the increase of the lateral force and velocity.

1.7.10 Understeering and oversteering

In standard terminology defined by the Society of Automotive Engineers (SAE) J670 and the International Organization for Standardization (ISO) 8855, understeering and oversteering are based on differences in steady-state conditions where the vehicle is following a constant-radius path at a constant speed with a constant steering wheel angle, on a flat and levelled surface. If the speed is increased slightly for the same radius path and, after settling into steady state, the same steering is measured, then the vehicle is said to have neutral steer. If more steering is needed at the higher speed to maintain the same radius of curvature, then the vehicle is said to have understeering. If less steering is needed at the higher speed, then the vehicle is said to have oversteering.

Understeering and oversteering are defined by an understeering gradient U that is the difference between a reference steer angle gradient and the Ackerman steer angle gradient. The reference steer angle (δ_R) is the average steer of the front axle wheels minus the average steer of the rear axle wheels. The Ackerman steer angle (δ_A) is defined for a given radius of turn as the reference steer angle that would be used at a very low speed. For a four-wheeled vehicle with steering only at the front wheels, the Ackerman angle δ_A (at the wheels) is the arctangent of the wheelbase divided by the turn radius (at the center of the rear axle).

Understeering and oversteering are formally defined using the gradient U: if U is positive, the vehicle is understeering; if U is negative, the vehicle is oversteering; if U is zero, the vehicle is neutral.

1.7.11 Area of contact

Because of the deflection of the tire there is no one point of contact between the wheel and the road as in case of a rigid wheel. The larger area of contact results in periodically variable peripheral velocity of the tread connected with the longitudinal deformation. The velocity of the points approaching the contact zone is decreasing and a circumferential compression appears. The speed in the contact zone is almost none because only limited sliding is present. Points leaving the area of contact accelerate as the deformation disappears and the tread length returns to initial. As a consequence the rotational speed of a pneumatic-tired wheel is lower than that of a rigid wheel with the same dynamic radius. The instantaneous centre of rotation is than located a little under the road surface.

Pressure in the contact area is very important. In a stationary vehicle the wheel is loaded by a force Q_W resulting from the car weight which distributes on the whole area of contact. Pressure distribution q_P shown in the fig. 1.67a [5] is even and the resultant force Z_W, normal to the road, is collinear with the Q_W. The pressure distribution changes when the vehicle is moving. In this situation there appears additional horizontal force F_W or rotational moment M_W acting on the wheel (fig. 1.67b and fig. 1.67c). The higher pressure is in the front zone, in the direction pointed by the velocity vector. The resultant force is therefore moved to the front by a distance *e* in respect to the wheel centre.

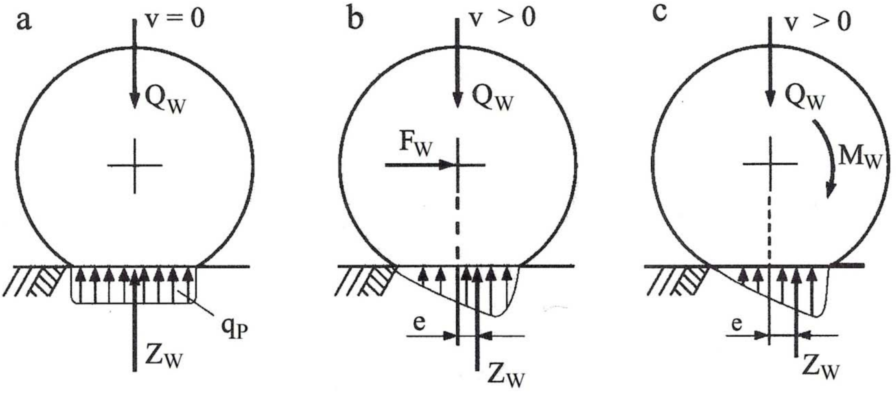

Figure 1.67 Pressure in the contact area

1.7.12 Energy losses

Because of the structure of the tire and its deformation there are energy losses connected with the movement of the wheel. They can be divides as following.

- **Loss of hysteresis**. Due to the viscoelastic properties of the rubber the stiffness of the tire is different during loading and unloading of the wheel. This phenomenon is called hysteresis and the area within the loop reflects the energy dissipated in one cycle of loading and unloading. During the movement the tire experiences repeatedly such cycles and the energy is lost. The size of hysteresis and thus the dissipated energy depends on the tire characteristics. A high hysteresis shortens the stopping distance of the vehicle but causes rapid wear. This phenomenon is the main source of energy losses in the wheel.
- **Sliding friction loss**. It is caused by the relative movement of the tire tread in respect to the road surface. The energy is dissipated as heat producing by friction.
- **Adhesion loss**. It is caused by the deformation of the tread pattern in the area of contact and the adhesion between the contacting surfaces.
- Energy loss from the friction in the bearings
- Energy loss connected with the air turbulences in the zone around the wheel.

The energy loses should be reduced in every places using scientific knowledge coming from laboratory investigation and computer simulations.

1.7.13 Resistances of movement connected with the wheels

- Rolling resistance
- Wheel resistance on wet pavement
- Bearing friction
- Toe-in resistance
- Turning resistance
- Resistance caused by rough pavement

Let's consider shortly all of them.

1.7.14 Tyre rolling resistance

During the movement of the wheel parts of the tire continuously enter the area of contact and are deformed, springing back to the initial shape when leaving the area. Producing the deformation requires energy that is not completely regained after ending the contact. Due to the internal damping of the elastic material the dissipation of energy occurs.

The previously calculated force F_W expresses the resistance of the wheel during the movement. It is called the rolling resistance F_R and can be expressed by the equation.

$$F_R = F_W = \frac{e}{R_D} Q_W \tag{1.158}$$

The force expresses the sum of energy losses which appear during the motion. It is proportional to the normal load acting on the wheel. The displacement of the reaction force that can be observed in the movement on the solid soil is the base for determining the coefficient of rolling resistance.

$$f = \frac{e}{R_D} \tag{1.159}$$

And the rolling resistance in then defined as:

$$F_R = f Q_W \tag{1.160}$$

The coefficient of rolling resistance contains not only the coefficient of rolling friction but also energy losses connected with the tire deformation and interaction between the tread and the road surface. There are many factors that influence its value.

Effect of the speed. The rolling resistance generally increases with the speed of the vehicle. The changes of the rolling resistance coefficient are shown in the Fig. 1.68. At the speed called the critical speed the curve becomes very steep. It is caused by the vibrations in the tire at high speeds when standing circumferential waves appear and the rolling resistance increases. At the critical speed the tire stops functioning normally and it should never be approached or exceeded during the normal use of the vehicle. It is very important factor in the choice of the proper tires.

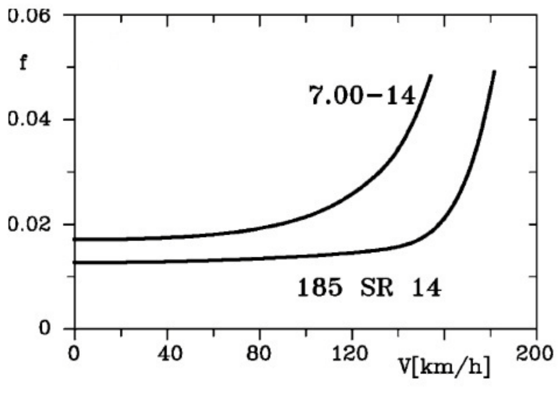

Figure 1.68 Diagram of rolling resistance f as a function of speed V

Effect of tire type and material structure. The radial ply tires show about 20% lower value of the coefficient of rolling resistance than the bias ply tires and a higher critical speed. As concerning the material different rubber compositions reveal different internal damping characteristics and its dependence upon the loading frequency. In general natural rubbers are characterized by lower damping than the synthetic. It means lower resistance but also lower critical speed.

Effect of tread wear. The rolling resistance is lower for the low profile tread. It is because the deformation and hysteresis phenomena occur mainly in the tread band. With the wearing of the tread the rolling resistance decreases.

Effect of operating temperature. With the increase of temperature the internal dumping of the tire decreases which leads to lower rolling resistance. The operating temperature tends to stabilize at some point because lower rolling resistance means less energy dissipated as heat.

Effect of inflation pressure and vertical load. The rolling resistance increases with the tire deflection. So the higher inflation pressure and a lower normal force both decrease the rolling resistance coefficient and increase the critical speed. However the pressure in the tires is determined by the producer as a function of the loading force Q_W. In the vehicle operation it shouldn't be increase in order to reduce the rolling resistance.

Effect of tire size. The geometrical parameters of the tire have influence on the rolling resistance. Tires with the larger radius and lower aspect ratio H/S are characterized by lower coefficient of rolling resistance and higher critical speed. Favourable is the decrease of the aspect ratio. The stiffness of the sidewalls is then higher and the deflection under load is smaller which leads to lower hysteresis losses. The ratio used on fast modern cars now is as low as 0.4.

Effect of road. The effect of the road surface is taken into account by choosing the proper fundamental rolling friction coefficient f_0 for different road types. The differences can be very significant.

Effect of the wheel sideslip angle. The larger sideslip angle α caused by a lateral force or a toe angle produces strong increase of rolling resistance as the tire path is not collinear with the wheel plane.

Effect of longitudinal force. The forces existing during braking or accelerating also affects the rolling resistance. An interesting fact is that minimum rolling resistance occurs for low driving force (it can be as low as 75-85% of resistance in free rolling conditions) but it increases sharply when the force is higher. It favours four wheel drive arrangements, in which there are moderate driving forces on all wheels, over the two drive wheel layouts where two wheels work with high driving loads and the others are idle.

The analysis of the rolling resistance is different for the movement on compliant soil. In this situation the main energy losses are connected with the creation of ruts and pressing, moving and shearing ground, sand, mud or snow. The attempts to reduce the rolling resistance focus on lowering the depth of the ruts by decreasing the pressure acting on the ground. It can be achieved by increasing the size and width of tires, lowering the inflation pressure or applying multi-axial drive systems. On the compliant soil the rolling resistance of the drive wheels is lower than of the free rolled wheels which produce a heap of soil in front of them. This favours all wheel drives.

1.7.15 Hydroplaning

Hydroplaning is sliding of a tire on a film of water. Hydroplaning can occur when a car drives through standing water and the water cannot totally escape out from under the tire. This causes the tire to lift off the ground and slide on the water. The hydroplaning tire will have little traction and therefore, the car will not obey the driver's command.

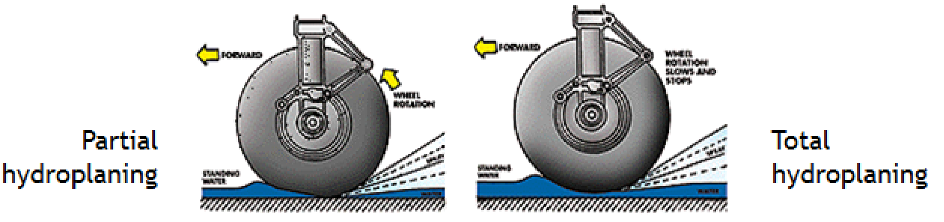

Figure 1.69 Hydroplaning

Deep grooves running from the centre front edge of the tyreprint to the corners of the back edges, along with a wide central channel help water to escape from under the tire.

There are three types of hydroplaning: dynamic, viscous, and rubber hydroplaning.

Dynamic hydroplaning occurs when standing water on a wet road is not displaced from under the tires fast enough to allow the tire to make pavement contact over the total tireprint. The tire rides on a wedge of water and loses its contact with the road. The speed at which hydroplaning happens is called hydroplaning speed.

Viscous hydroplaning occurs when the wet road is covered with a layer of oil, grease, or dust. Viscous hydroplaning happens with less water depth and at a lower speed than dynamic hydroplaning.

Rubber hydroplaning is generated by superheated steam at high pressure in the tireprint, which is caused by the friction-generated heat in a hard braking.

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2 POWER TRAIN

2.1 Introduction

In chapter 2 power train system is briefly described. The main elements of the system are mentioned. Fundamental information about friction clutches as well as torque converters are presented. Different types of gearboxes from classical up to continuous variable are described. Short sub-chapter presents propeller shafts and produced contemporary universal joints. Different types of differential ends the chapter.

2.2 Power train elements

Power train can be classified as all of devices and systems necessary to transfer the power provided by engine to the wheels in order to obtained movement. Essentially, the power train includes such main elements as:

1. The engine,
2. The clutch,
3. Gearbox (transmission),
4. The propeller shafts and universal joints (e.g. Cardan joint),
5. The differential,
6. The drive axle and wheels assemblies.

The cars manufactured nowadays however, are very amply - equipped with electronics (e.g. systems for emissions regulation, fuel economy, and performance, electronic controls, automatic transmission control, electronic control of clutches in a differential, etc.), therefore it must also be considered as a port of a power train[1].

An example of power train key elements is shown in *figure 2.1*. The torque provided by the engine is further delivered to the drive train via the clutch and gearbox. The clutch contacts two independent shaft whereas the transmission amplifies the torque by the gear ratio. Furthermore the torque delivered via the driveshaft and differential to the axle shaft is amplified by the final drive ratio. It should be remembered that all inertia forces of rotating elements interferes the overall performance of power drive elements. Therefore, the inertial forces should be taken into account in overall deliberation[2].

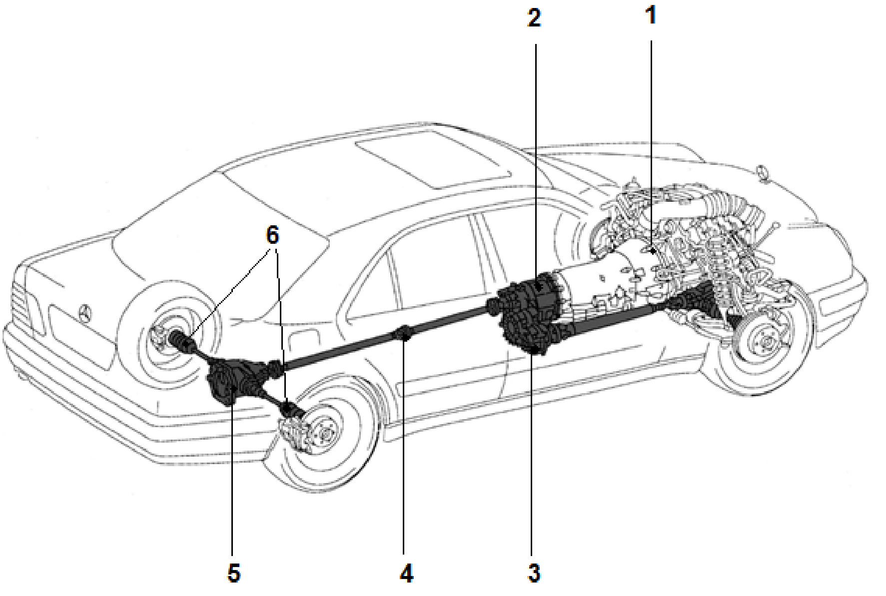

Figure 2.1 The key elements of power drive. 1) engine, 2) clutch, 3) gearbox, 4) Cardan joint, 5) differential, 6) drive axle [2]

2.3 Clutch

Essentially, the cultch is a device responsible for connecting and disconnecting two rotational shafts. The clutch is said to be "*engaged*" or "*in*" when the shafts are connected and "*disengaged*" or "*out*" when the shafts are released. Furthermore, mechanical clutches can be separated on two categories. Namely, *positive engagement* and *progressive engagement.* The positively engaged means that both drive and driven shafts are connected and rotate with this same speed. when the system is positively disengaged, there is no torque transfer from drive shaft into the driven one. However this type of clutch cannot be used for connecting an engine with shaft as the connection consist on applying splines, keys or dogs which make the connection permanent. This type of clutch is used rather in gear boxes. The progressive type is gradually engaged. That means that the speed of the driving shaft decreases and at this same time the rotational speed of driven shaft increases until both of the shafts reaches this same rotational speed. Usually the friction clutch of a progressive engagement type is used for engine-gearbox connection [3].

In case of automatic gearbox, mostly the torque converter is used instead of the clutch. The torque converter not only deliverer the torque from the engine into the gearbox but also increases the torque at the beginning of car movement. This can only happened however, when the engine is turning much faster than the transmission.

2.3.1 Friction clutches

As the name of this clutch suggests, the friction clutch operates based on friction phenomenon. Naturally, the friction is inextricably linked with the temperature. The temperature generated during the clutch operation has an influence on the static/dynamic friction coefficient, and therefore especially imported due to the fact that it affects the dynamic engagement characteristics [4].

In order to satisfy all kind of vehicle and in consequence different working condition various design of clutches are introduced. For example in high performance vehicles a multi plane clutch with high efficiency are used. Wet clutches, that is the discs immersed in lubricant reduces the temperature generated by friction and in consequence have longer operation life. This kind of clutch however, is less efficient due to slipping of the discs but this can be compensated by introducing multi clutch discs to the system. Furthermore, clutches can have safety limitation provided by their design. For example centrifugal clutch will disengage itself when the rotational speed of the shafts breaches the threshold value.

Fundamentals

Generally in the case of the simplest manual gearbox the clutch is assembled with two plates connecting drive and driven shafts. The more substantial driven plate is installed on the engine flywheel. Usually lighter disc, called the pressure plate is fitted to the output shaft. The flywheel is bolted to a flange on the end of the crankshaft. The clutch is originally engaged, which means that pressure plate presses against the engine flywheel. In consequence, the torque is delivered into the gearbox. This is obtained by a spring or set of springs. The disengagement of the clutch is fully depended upon the driver decision. Pushing the clutch pedal releases the pressure between the discs and there is no torque delivery between the shafts and then the engine is idling or arises the possibility of shifting the gear [5].

In order to ensure better performance of the clutch third much lighter disc called the *central plate* or *friction plate (number 3 in figure2.2.* [6]*)* is employed coaxially between the driving discs.

Of course this disc has to be caver with high friction material on both sides. this device double

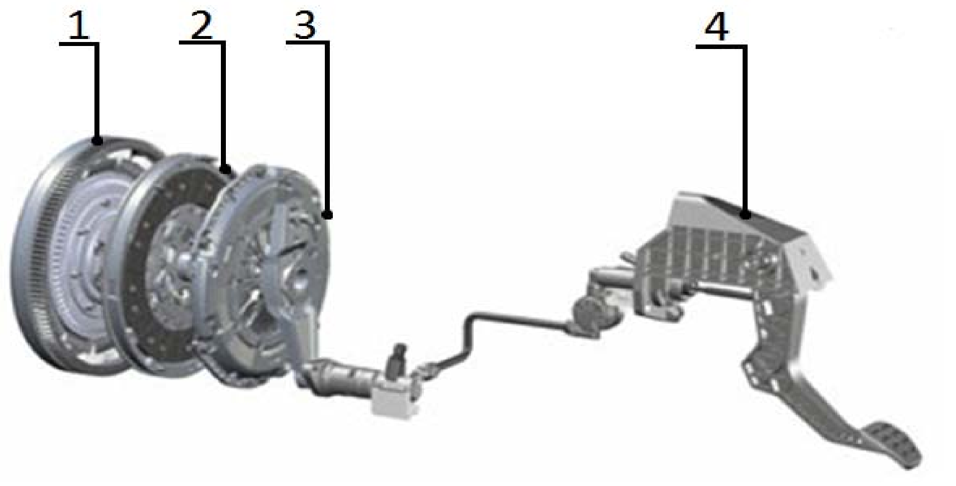

Figure 2.2 The clutch system. 1) flywheel, 2) central disc, 3) pressure plate, 4) clutch pedal[6]

torque capacity of the clutch, moreover reduces to the half the speed of rubbing during progressive engagement and in consequence increasing the life of the friction material [3]

The moment of friction (*T*) produced on the plates should be sufficiently great to transfer maximum torque from the engine(T_{max}). Another word the friction produce on the plates should be great enough to prevent slipping. In order to estimate the computational relationship the coefficient of clutch margin (β) was introduced.

$$\beta = \frac{T}{T_{max}} > 1 \tag{2.1}$$

During the clutch operating the friction material will wear and in consequence the length of the springs will be increase whereas the pressure force will decrease. Therefore, the coefficient should be sufficient enough to transfer total torque delivered by the engine during its life cycle.

Approximate values of coefficient β are:

— Passenger cars 1,4÷1,8

— Trucks 1,5÷2,0

— Buses 1,3÷1,7

Kipping in mind the coefficient β given in equation 1 the moment of friction produced by the clutch can by found by:

$$T = \beta T_{max} = \frac{\mu i F_p r_m}{1000}, Nm \tag{2.2}$$

Where:
i is the number of friction surfaces,
μ is the friction coefficient, usually 0,25 ÷ 0,35.
r_m is the means radius of linings. With reasonable approximation the mean radius yields $r_m = \frac{r_1 + r_2}{2}$
F_p is the pressure force to plates.

The number of friction surfaces are mainly responsible for transferring the torque from the engine to the gearbox and mostly it is the number of surfaces equal to the double number of driving elements. For example for single plate clutch number of friction surfaces is i=2, whereas for twin clutch this number rises to i=4 [7].

The pressure force of the plates fixed on the radius of r_m can be found by:

$$F_p = \pi(r_2^2 - r_1^2)p \tag{2.3}$$

Where:
p is pressure per unit area, expressed in $N/_{mm^2}$.

The pressure per unit area describes, in some extend, the level of plate wear. In order to ensure most appropriate working condition this factor must be comprised between a permissible, boundary value different for various cars. The permissible pressure per unit area is dependent upon the ratio of torque produced by the engine proper to the mass of the vehicle.

Having all those data it is possible to compute desired outer radius (r_2) of the friction disc for given vehicle.

$$r_2 = \sqrt[3]{\frac{3000\beta T_{max}}{2\pi i p(1-\delta^3)}} \tag{2.4}$$

Where δ is a coeffienct which characterizes the dimensions of friction lining and represents a ratio of inner radius to outer radius $(\delta = {}^{r_1}/_{r_2})$. The inner radius cannot be changes proportionally to the outer radius, therefore for small clutches greater value of coefficient δ is acceptable (e.g. in passenger cars), whereas lower values of the coefficient are common for bigger clutches (e.g. large trucks). The value of δ is normalized, hence can be found is specialist literature. It varies between 0,52÷0,72 [7].

Knowing the rotational speed of the shaft (*n*), the power transmitted by the clutch can be found by:

$$P = \frac{2\pi T n}{1000}, kW \tag{2.5}$$

Clutch friction materials

A clutch transfers torque from the engine into the gearbox thanks to the friction occurring between clutch plates, therefore the material used for clutch lining should reflect great friction coefficient, low wear susceptibility and sufficient durability. Naturally, tow material rubbing against each other will produce large amount of heat $(500 \div 700°C)$ so it is desired that the friction ceffictent of the lining is not sensitive on the temperature. Furthermore the material should be insensitive to the water or oil spillage.

Generally, in case of dry clutch the friction material is very similar to this used in brakes [4]. Nowadays most common are three type of materials: organic, sintered iron, and ceramic.

Organic friction material

This material is most widely used. It mainly consist of asbestos (or more recently other mineral fibers) yarn woven with copper or brass wire [4]. The woven material can be farther separated on two types:

- Solid woven. The cloth is woven to the desired thickness of the lining before impregnation.
- Laminated. Several layers of the cloth is assembled and then impregnated [3].

The organic friction material is capable of withstanding maximum operation pressure of 2 bar and relatively low maximum temperature 230℃. Operating at temperature approaching to this value will cause rapid wear of the material. Moreover, this kind of material represents the friction coefficient ranging 0.2 ÷0.45. This material however, is less durable comparing with sintered iron, and ceramic but due to its simplicity is used in many low cost application [4]

Sintered iron, and ceramic friction material

Sintered iron, and ceramic material is able to operate at much greater pressures and temperatures than organic material. Furthermore, ceramic material is relatively light, hence clutches with this kind of lining has low moment of inertia. The light weight, temperature and pressure resistance are reasons of employment of those clutches in high performance vehicles. Unfortunately, all those advantageous properties is followed by large cost of the clutch.

In general, linings are usually riveted onto the plates usually by copper or aluminum rivets. This material will not damage the plates or cones of the clutch when they rubs against the rivets. Moreover the rivet heads are located in counter bores in the friction material.

Obviously wet clutches has different friction material than dry clutch due to different operation conditions. First kind of friction material of wet clutch is so called “paper” type. Later this material was improved by adding graphite which lead to establish another type of friction material called “graphite’ type.

Paper type of friction material

The miss-leading name of this type material origins from the fact that are manufactured by using the paper making machines. In fact the paper type friction material is produced by mixing a water slurry of asbestos, cellulose and fillers, which are afterwards shaped and dried. In order to obtain required rigidity the paper is saturated with oil or liquid resin. The paper type of friction material is cheap and represent relatively high dynamic friction coefficient [4].

Graphite type of friction material

Graphite material is an improved of paper type. This material is enriched with additional graphite which makes the material more resistance to heat due to graphite heat conductivity. In consequence it allows to transfer greater torque, hence are widely used for high – energy transmission [4].

Polymeric friction material

Due to high resistance of polymer on resilience and durability this material represents the highest power absorbing properties among any other clutch friction material. Essentially this material is made of such compound as elastomer, ceramic and organic fibers, lubricant and a curing system.

The properties of wet clutch friction materials are highly dependent on the properties of oil in which they are immersed. Essentially, the oil is an highly compounded hydrocarbon lubricant. The friction properties of the clutch lining can be moderated by adding viscosity-index improvers, antifoam agents, antioxidants, and extreme-pressure lubricants [4].

Diaphragm clutch

Essentially, this type of clutch is used to decrease clutch pedal pressure. Therefore it is informally called “soft” or “velvet touch” clutch. Moreover, such design enables higher torque capacity and increased durability. Therefore it is used in almost every car. The operation of this kind of clutch is based on the larvae, hence the load on the pedal is reduced and , simultaneously grater pressures can be obtained.

Generally the clutch comprises a cast iron pressure plate, a diaphragm spring (or diaphragm disc) made of steel, and a low carbon steel cover pressing. The diagram spring resembles a dished disc divided into fingers located helically around a circumference. Those fingers applies pressure when the clutch is engaged. The principle of operation can be easily explained with aid of *figure 2.3.* As it can be seen when the clutch is engaged and connected with the flywheel clutch cover , presses the diaphragm spring into the pressure plate. In consequence the pressure plate presses against the flywheel and the torque is delivered. When the clutch is disengaged (the clutch pedal is pushed down and there is no torque delivered to the transmission) the diaphragm spring is disturbed and resembles dished disc. In consequence the pressure disc is pulled backwards from the clutch disc and the engine is idling.

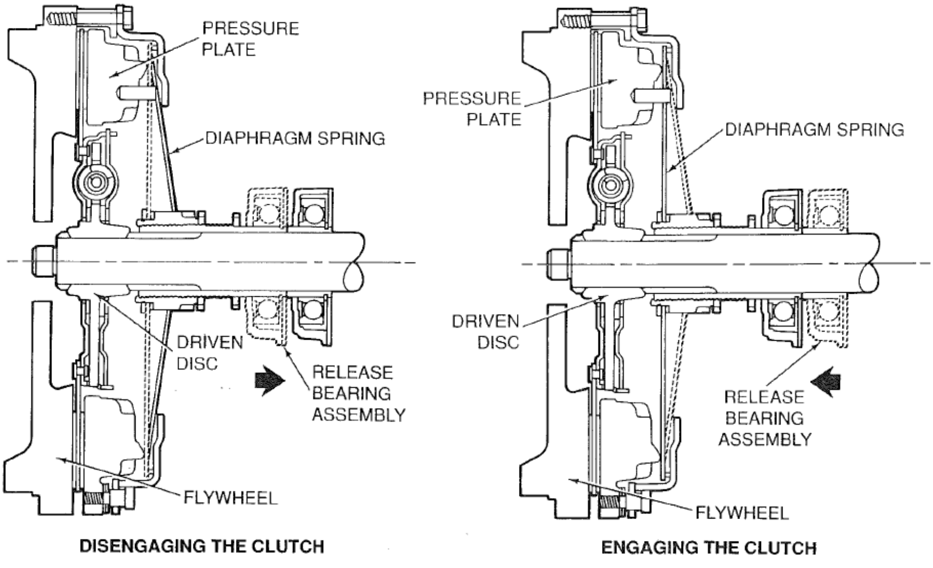

Figure 2.3 Diaphragm clutch engaged and disengaged.

Multiplate diaphragm

The multiplate diaphragm clutch is presented in *figure 2.4*. as it can be seen such a mechanism consist mainly of drive and driven plates. The dire plates are blocked with lugs and slots so they cannot spin. The driven plates however, are fixed to corresponding spines formed on the gearbox spigot shaft. The diaphragm resembles the construction of the plate used in regular diaphragm clutches. It is a dished disc located on the input shaft with a shouldered pivot post. The pivot post in turn, is riveted to the cover pressing. When the clutch is engaged the diaphragm spring presses the outer ring. This in consequence provokes clamping of all elements. Then the torque is transferred. Conversely, disengaging the clutch will result in pressing the inner ring. Then the friction materials loses contact and there is no torque transfer.

Besides of ability to transfer greater torque an another advantage of this type of clutch is that it will not suffer efficiency loss when the friction linings begins to wear as the diaphragm spring will deform and became more dished. Consequently will exert larger axial clumping load [8].

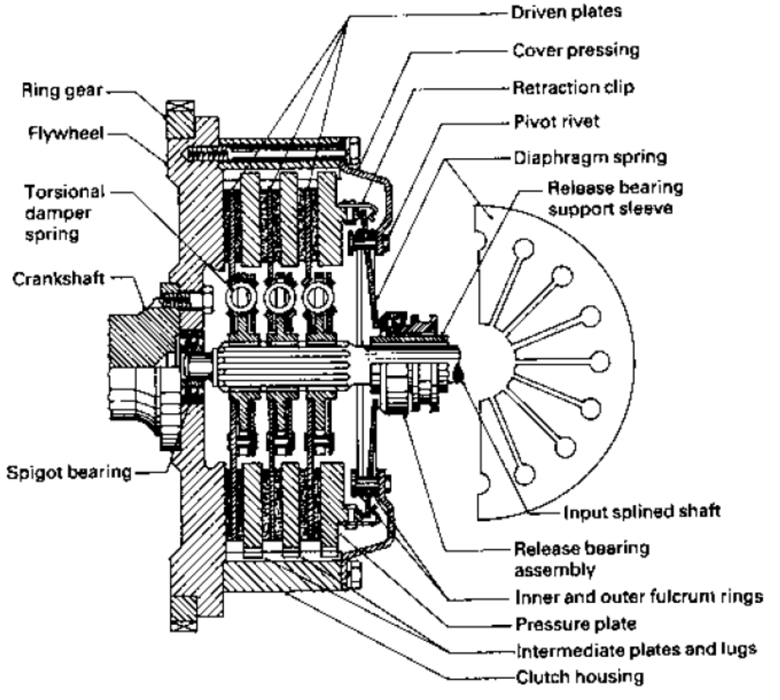

Figure 2.4. The multiplate diaphragm clutch [8]

Actuating mechanics

Essentially , there are three main mechanics involving the clutch operation control. All of them are explained in this subchapter.

- Cable linkage control system,
- Hydraulic control system,
- Servo assisted clutch air/hydraulic.

<u>**Cable linkage control system**</u>

The linked control system is widely used in present applications. The working principia of this kind of mechanics is very simple and is visualized by *figure 2.5.*

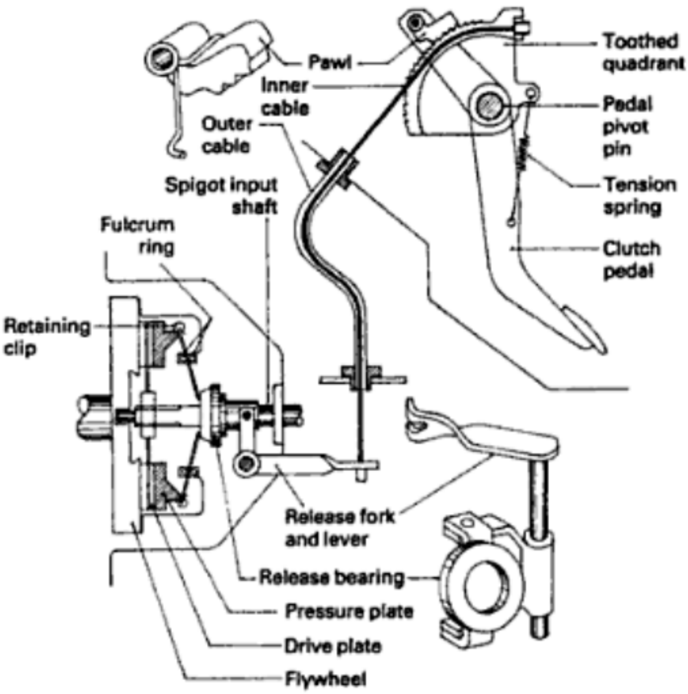

Figure 2.5 Clutch cable linkage control system [8]

In general, the inner cable connects the clutch pedal with the release forge and leaver. The release bearing in turn is in constant contact with the diaphragm spring by means of the pedal self-adjustments mechanism. This mechanism ensures that the adjustment pawl do not contact the teeth not the pedal quadrant when the pedal is released [8]. In order to provide correct operation of the system the inner cable must be tensioned. This is usually obtained by the tension spring. Accordingly when the pedal is pressed the cable is pulled influencing on the leaver and released bearing. In consequence the clutch mechanism is disengaged.

Hydraulic control system

The hydraulic control system is mostly used in heavy vehicles. This systems is depicted in *figure 2.6*. when the pedal is pressed the piston of the master cylinder starts to force the working fluid to the slave cylinder which, in turn, presses the fingers of the diaphragm springs. In consequence the clutch is disengaged. Unfortunately, the resistance force provided by the spring is insufficient to engage back the spring and move back the pedal when released. Usually the pedal movement for clutch engagement and disengagement is supplemented by the over-centre spring. When the pedal is pressed the over-centre spring changes its position as indicated in *figure 2.6.* This position helps to keep the pedal pressed. Conversely, when released the spring changes its position and the tension of the spring holds the pedal up.

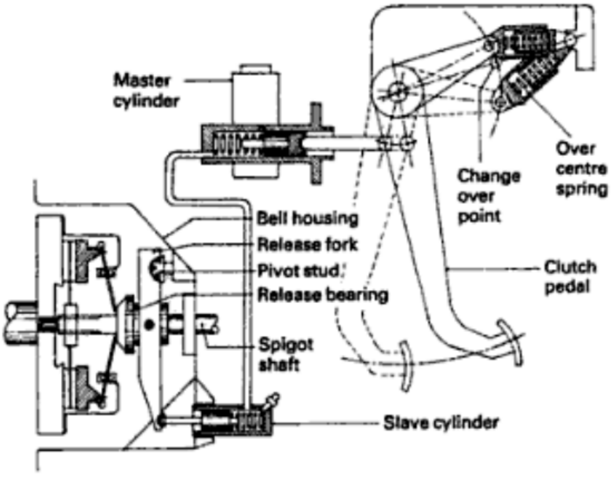

Figure 2.6 Clutch hydraulic control system with over-centre spring [8]

Servo assisted clutch air/hydraulic.

Essentially, servomechanism is used as a supplement for the hydraulic system as the high torque capacity clutches provokes unacceptably heavy loads on the pedal clutch. This is obtained by so called an air-hydraulic clutch servo unit. This unit uses compressed air for assist the hydraulic pressure transmitted to the clutch slave cylinder[9].

A servo assisted clutch air/hydraulic is presented in *figure 2.7*. As it can be seen the system contains two coupled cylinders from which the pneumatic cylinder represents larger diameter comparing with hydraulic cylinder. There is a slave piston within each of the cylinders. The slave piston in turn is connected with a push rod which operates the clutch withdraw mechanism. At this point two pressurized media starts to cooperate together to move the slave piston. Furthermore, the system is equipped with typically vertical control valve unit. Is usually bolted to the smaller hydraulic cylinder. The control valve cylinder is separated on two chambers, upper of which contains a reaction hydraulic plunger. The lower in turn contains a stepped air control piston. Understandably, the hydraulic and pneumatic media must be separated, therefore a diaphragm seal is employed for this purpose. This solution allows to operate in unison without mixing. The control valve unit is farther equipped in the exhaust outlet located above the air inlet valve working under regime of an air control piston. Both pistons and valves are fixed on springs provoking them to return upwards when the pedal is released [9].

In as much the construction of the system may appear more complicated comparing with the previous solution, its operating principia is quite easy. When the clutch pedal is pressed the hydraulic master cylinder moves forcing the fluid to flow through the upper chamber of the control valve unit into the hydraulic cylinder and in consequence influence the slave piston. Of course, there is some force applied but in order to facilitate the process the air pressure must be involved. When the fluid flow through the upper chamber of the control valve, it “pushes down” the reaction plunger which in turn pushes down the air control piston. Then the supply of compressed air is released and directed to the pneumatic cylinder. The increasing air pressure within the cylinder moves the piston, in consequence providing an assistance for the slave piston. Furthermore, the air acting on the underside of the control piston creates a reaction force which attends to lift the control piston upwards. Hence when this upwards force is greater than the downwards force given by the plunger, the control piston will be lifted just enough for the inlet valve to close and shuts off the air assistance in the servo mechanism. This will happen for example when the clutch pedal is partially pressed. When the pedal is gradually realest the reaction pressure on the plunger decreases. In consequence the control piston will be lifted mainly by means of springs. Furthermore the control piston will no longer be in contact with already closed inlet valve. Then the “used” air from the pneumatic cylinder can be led outside system via the central passage of the air control piston [9].

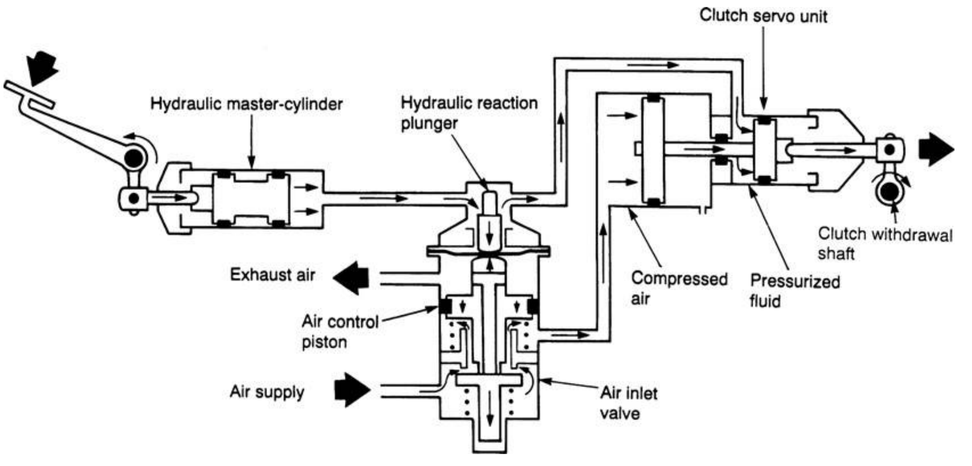

Figure 2.7 A servo-assisted clutch [9]

2.3.2 Torque converters

An automatic transmission does not use clutches for torque transfer from engine into the gearbox. Instead the torque converter is applied. The torque converter combines both converter and coupling operations [1]. Another words the torque converter is a type of fluid coupling which allows, in some extend, the engine to rotate independently from the transmission. One of the great advantage of the torque converter is that it can even double or triple the torque by the engine when the car is starting to move. This however is possible only when the engine is spinning with much greater speed than the transmission. Eventually, at high speed the transmission equalize it speed whit the engine and both of them rotates with almost the same speed. Practically, both sides of the system very rarely gain the same speed. This difference provokes the waste of energy, hence the old cars with automatic transmission are, in nature, more fuel consuming then the cars with manual transmission. In order to counteract this phenomenon modern torque converters are equipped with so called lockup clutch. When a threshold rotational speed is breached this clutch will lock both sides of the torque converter together preventing them to slip and in consequent improve efficiency.

Hydrokinetic fluid couplings

Hydrokinetic fluid coupling (also called a fluid flywheel) like a torque converter transfer torque form the engine into the gear box. As the name may suggest the operation of this device consist on the change of the kinetic energy of a circulating fluid. Constructionally, the torque converter and fluid couplings are quite similar, however in oppose to a fluid coupling, a torque converter is capable of multiplying the engine torque. It is because fluid is returning from the turbine flow in the direction which prevents the fluid to assist rotation of the impeller.

It is common to compare the contraction of the fluid coupling system to two halves of grapefruit, facing each other with removed pulp and intact dividers [9]. In reality, the fluid couplings is assembled from two saucer-shaped discs, an input impeller (pump), and a turbine (runner). The turbine is divided by a flat radial vanes (blades) so they creates a curved cells as indicated in *figure 2.8.* [8]. The another function of these blade is to support hollow semicircular cores for guide rings (also called torus). Those rings in turn reduces turbulences in the coupling and are located with some offset from each other within their cavities in order to equalize flow areas in the cells an assist the fluid flow circulation at the earliest moment of the rotation of both elements [8], [9]. Furthermore the two halves are put together so the turbine and impeller are facing each other. Understandably, in order to provide correct operation there must be a gap left between those two elements. In order to reduce noise in the coupling the blades are unequally spaced or different numbers of blades are installed on each member of the coupling. The impeller is connected to the engine flywheel via a tours cover. The turbine is enclosed by the fly wheel and the tours cover and is connected with a gearbox.

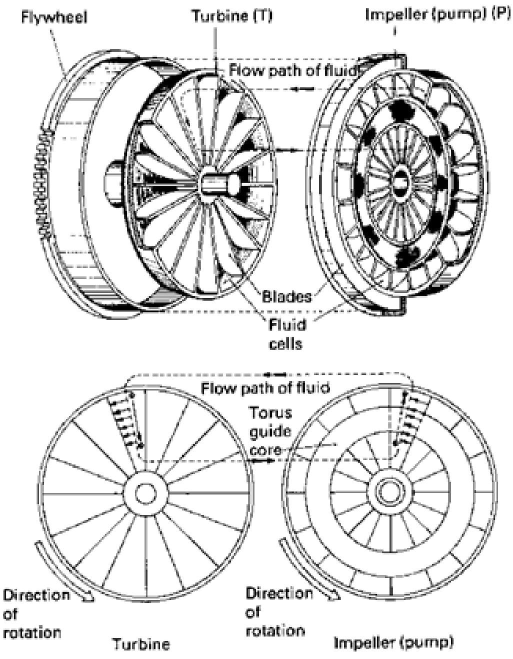

Figure 2.8 Fluid coupling construction [8]

The fluid coupling members are made of aluminum die castings or, more recently, of steel pressings. The fluid used in this device must represent low viscosity and high density. It is because the operation of the coupling is based on the kinetic energy of the fluid. The motion of impeller and turbine is rot a result of a viscous drag. The high viscous fluid would increase slip [9]

Generally there are two types of flow within the fluid coupling (see *figure 2.8.*). Namely, the *rotary flow* and *vortex flow*. As the name suggest the rotary flow arises in the direction of impeller rotation. The vortex flow is slightly more complicated. Generally this kind of flow arises when the impeller rotates with much greater speed than the turbine. It is because the centrifugal forces of the fluid in smaller moving member of the system will be smaller (turbine, when the car is accelerated) than the centrifugal force of the fluid inside the faster moving part (in this case the impeller). Due to the fact that the pressure of fluid on the outer wall of the impeller will be the same as the pressure at the outer wall of the turbine, a cross flow of the fluid must occur between both members of the couplings. In consequence the slower mowing turbine will be constantly filled with the fluid. This fluid constant movement will in consequence create a helical flow around the coupling.

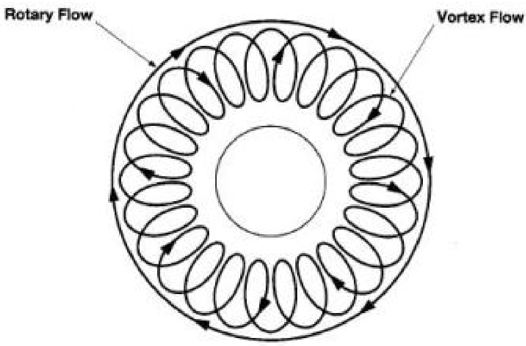

Figure 2.9 Fluid flow inside the coupling

In order to understand operation principia of the fluid coupling let's consider firstly a situation where the engine is idling so the flywheel and thus the impeller is rotating with very low speed. Of course, this provokes some movement of the fluid and consequently "attends" to move the turbine but the inertia of the car or slightly applied brake by the driver will be great enough to resist this motion. Hence the car does not moves. In this case the vortex flow arises in the coupling. When the driver presses on the acceleration pedal the engine produce more torque, hence the flywheel and impeller rotates faster extorting greater kinetic energy of the fluid. The fluid therefore, more sufficiently interacts with the turbine and at some point the kinetic energy of the fluid will be great enough to overcome all resisting forces and eventually move the turbine. When the car is driven with constant speed, eventually the turbine will reach approximately the same speed as the impeller. Then the torque capacity is sustained only by the rotary flow. This situation is known as the coupling phase of the coupling [9]. Unfortunately the slippage cannot be eliminated and at best it can reach about 2-4% [1]. Furthermore the slip (S) can be found with aid of *equation 2.6.*

$$S = \frac{\omega_{IM} - \omega_{TU}}{\omega_{IM}} = 1 - \frac{\omega_{TU}}{\omega_{IM}} = 1 - v \qquad (2.6)$$

Where:

ω_{IM}, ω_{TU} are the rotational speed of the impeller and turbine, respectively,

In case of emergency when the wheels will be rapidly stopped with engine running with high rotational speed the system will not be damaged. Let's now assume that the driver decided to slow down and released the acceleration pedal. Then the flywheel reduces its speed but the turbine for some time stays at the same speed. hence the turbine rotates faster than the impeller. Engine compression resist this motion and in consequence the vehicle slows down.

The efficiency of the Fluid couplings is highly dependent upon torque of particular member and the slippage that occur. Usually the coupling reaches maximum efficiency between 93% ad 95%. The efficiency of the fluid coupling is given by:

$$\eta = \frac{T_{TU}\omega_{TU}}{T_{IM}\omega_{IM}} = \frac{T_{TU}\,\omega_{TU}}{(T_{TU}+T_f)\omega_{IM}} = \frac{T_{TU}}{(T_{TU}+T_f)}(1-S) \tag{2.7}$$

Where :
T_{TU},T_{IM} are the torque of a turbine and impeller respectively.
T_f is the external air friction.

The particular torque of the each member depends upon the density of the fluid (ρ), the flow rate (Q) and the twist difference between the baled input and output $\Delta(rc_u)$. This difference is dependent upon a radius (r) and the circumferential component of absolute speed (c_u). Hence the torque can be found using Euler's turbine equation [10]:

$$T = Q\rho\Delta(rc_u) \tag{2.8}$$

Torque converter performance technology

As it was already explained the torque converter is a type of fluid coupling which is capable of increasing torque given by engine while reducing the speed. Hence, in general it converts not only rotational speed (as a clutch), but both speed and torque (as a transmission) [10].

Constructionally, a torque converter is very similar to previously explained fluid coupling. The only difference is that the torque converter besides an impeller and turbine is additionally equipped in third element called reactor or stator. Stator is a bladed wheel placed in-between the impeller and turbine. Usually the stator is equipped with an overrun clutch which connects it to the propeller shaft. This clutch ensures the possibility to rotate in both directions.

The operation of a torque converter is based on this same phenomenon as it is in case of fluid couplings. The impeller of the torque converter is also bolded to flywheel of the engine, yet again when the engine operates with low rotational speed the impeller "throw" the fluid against the vanes of the turbine forcing it to rotate. When the impellers revolutions are too low to overcome all resisting forces the engine torque will not be passed to the gearbox. When the impellers spins faster the hydrokinetic forces are great enough to move the turbine, and thus transfer the torque from the engine into the transmission. in case of fluid coupling however, the returning flow fluid can slow the rotation down whereas in case of torque converter the stator leads the oil towards the centre of impeller and in consequence provided an additional thrust. The fluid enters the passages shaped by the impeller entrance [8]. The flow direction is presented in *figure 2.10*. This increases the torque given by the engine. Therefore the torque balance now is:

$$T_{engine} + T_{reaction} = T_{output\,turbine} \qquad (2.9)$$

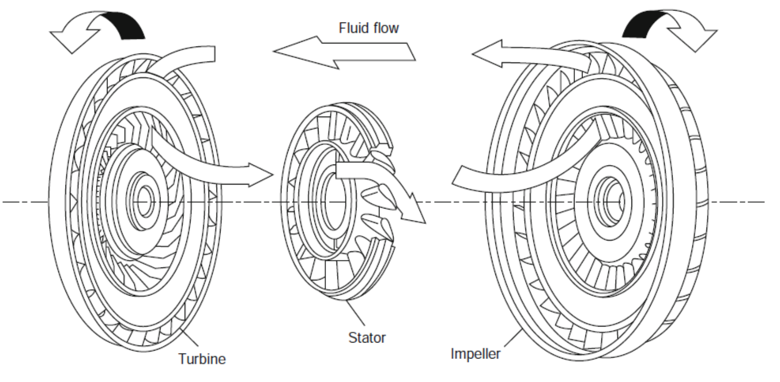

Figure 2.10 Fluid flow within a torque converter [1]

As it was already pointed out the lockup clutch will be engaged and lock both impeller and turbine together when they reach similar speed.

The performance of torque converter are dependent upon the speed ratio (C_{sr}) (*equation 2.12)*, torque ratio(C_T) (*equation 2.12*), efficiently (η) (*equation 2.12*) and what is known as a torque converter capacity factor or sometimes size factor(K_{TC}) (*equation 2.12*). This factor essentially describes the ability of the converter to absorb or transmit the torque given by the engine [11].

$$C_{sr} = \frac{\omega_{TU}}{\omega_{IM}} \qquad (2.10)$$

$$C_T = \frac{T_{TU}}{T_{IM}} \tag{2.11}$$

$$K_{TC} = \frac{\omega}{\sqrt{T}} \tag{2.12}$$

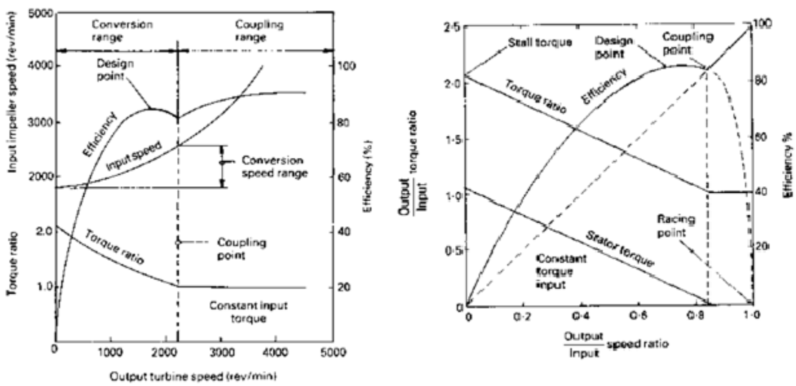

Figure 2.11 Performance characteristics of a torque converter [11]

A charts representing the performance of the torque converter is shown in *figure 2.11.* as the figure indicates the maximum torque multiplication arises for the stall condition of the engine. This means that engine, and hence impeller, slowly rotates whereas the turbine does not move. Then the torque ration reaches 2:1. At this point the vortex flow of the fluid will have maximum as well. Since one member of the converter does not move the speed ratio is zero. The speed ration increases with decrease of the torque ratio. When the hydrokinetic forces is great enough the turbine begins to rotate. Then the efficiency rises and the velocity of the vortex flow gradually decreases. Eventually the vortex flow velocity will be neglected and only the rotary flow will arise. This will only happened when the turbine and impeller rotates with approximately (the slip cannot be forgotten) equals speed. In consequence the output to input torque ratio will decrease as well while the efficiency reaches maximum. Now the reaction torque is equal to zero and the stator is freewheeled (this condition is frequently termed as the coupling point). This means that the stator can freely rotate in both directions. In consequence the fluid does not have so much resistance to circulate. Moreover, this improvers the coupling efficiency.

In order to determine the actual operating conditions of a torque converter is necessary to establish a factor describing the operating of an engine which gives a torque to torque converter . This factor is known as an engine capacity factor (K_{en}) and is given by [11]:

$$K_{en} = \frac{\omega_{en}}{\sqrt{T_{en}}} \tag{2.13}$$

Where: ω_{en}, T_{en} are the engine speed and torque, respectively.

In order to obtain appropriate engine and torque converter matching both of them should reflect similar values of capacities factor $(K_{TC} = K_{en})$. The values of those factors can be found by *equations 2.13* and *2.13.* Knowing values of the factors the output torque of a converter is:

$$T_{TC} = T_{en} C_{sr} \tag{2.14}$$

And the output speed of a converter is:

$$\omega_{TC} = \omega_{en} C_{sr} \tag{2.15}$$

Where n_{TC}, T_{TC} are the output torque and output speed of the converter, respectively.

Knowing the reduction ratios of a gearbox drive axle it is possible to calculate the tractive force:

$$F = \frac{T_{TC} \xi_0 \eta}{r} = \frac{T_{en} C_{sr} \eta}{r} \tag{2.16}$$

and speed of a vehicle:

$$V = \frac{\omega_{TC} r}{\xi_0}(1 - i) = \frac{\omega_{en} C_{sr} r}{\xi_0}(1 - i) \tag{2.17}$$

An example of tractive force corresponding to vehicle speed of a car with torque converter and three-speed gearbox is shown in *figure 2.12* [11]

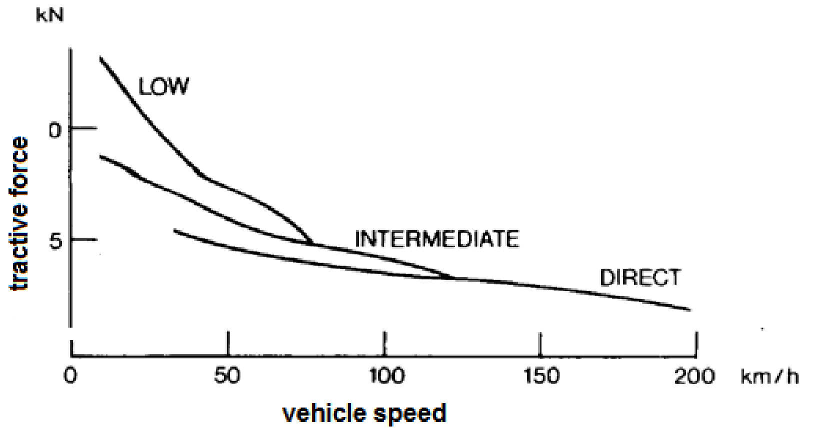

Figure 2.12 Tractive force with respect to the speed of a passenger car with torque converter and three speed gearbox [11]

2.4 Gearbox

The gearbox is a mechanism of a propelling system, which is used for change of torque produced by engine in order to provide the wheels with force necessary to overcome all driving resistances.

- The conventional gearbox used in piston engine enables
  - Torque change
  - Obtaining rotational speed of drive shaft sufficient for required velocity of a vehicle
  - Reverse movement, obtained by inverse direction of the driving shaft or drive half axle
  - Disconnection of the wheels from the engine – by setting neutral gear
- The gearbox should:
  - Be easy to control
  - Be economical in terms of controlling fuel consumption with respect to assumed engine characteristic
  - Be silent-running

There are several possible divisions of gearbox types:

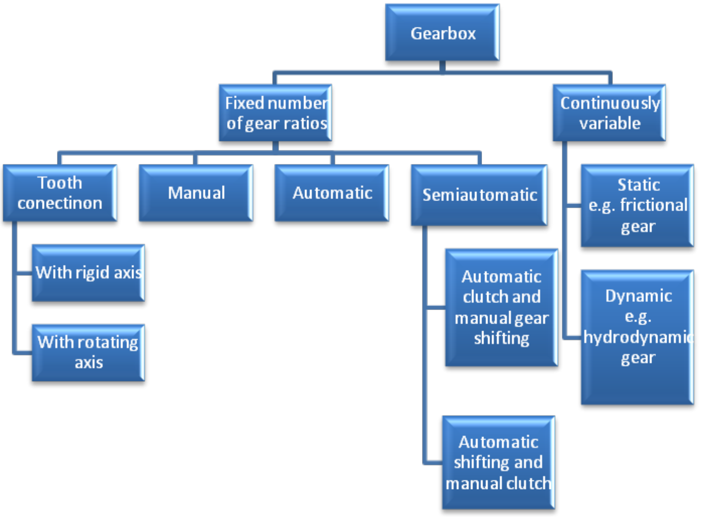

Figure 2.13 Gearboxes division

Gearboxes with fixed number of gear ratios are the most common in the automotive industry. The main principle of gears ratio in explained on the figure below:

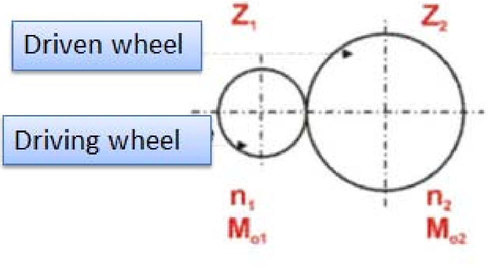

Figure 2.14 Basic ratio relation

The formula describing this relation is:

$$i = \frac{z_2}{z_1} = \frac{n_1}{n_2} \tag{2.18}$$

In a gearbox not only two gears are working together but usually it is a chain of gear pairs.

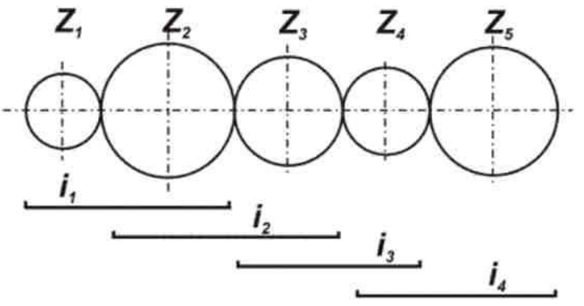

Figure 2.15 Chain of gear pairs

For a chain of gears - the equations describing the total ration would be the product of all gear pairs and look like:

$$i_c = i_1 \cdot i_2 \cdot i_3 \cdot i_4 \tag{2.19}$$

2.4.1 Gear ratio calculation

Gearbox role is to change gear ratio between the engine and wheels – therefore change the momentum available on the wheels to accelerate the vehicle. Because of high static resistance – the first gear need to have a high ratio to provide high enough momentum on the wheels. Knowing the construction data of a gearbox – gear ratios on specific gear can be calculated.

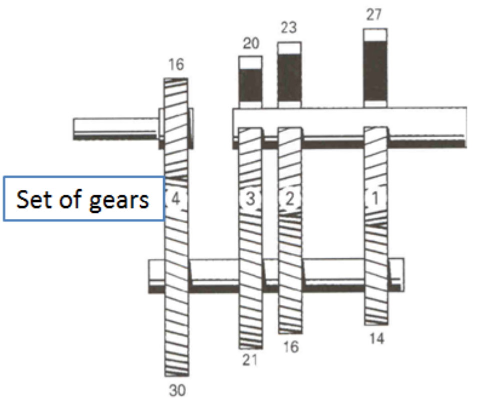

Figure 2.16 4-speed gearbox scheme

The equations for specific gear can be calculated by:

$$i_{1st} = i_4 \cdot i_1 = \frac{30 \cdot 27}{16 \cdot 14} = 3{,}62 \tag{2.20}$$

$$i_{2nd} = i_4 \cdot i_2 = \frac{30 \cdot 22}{16 \cdot 16} = 2{,}659 \tag{2.21}$$

$$i_{3rd} = i_4 \cdot i_3 = \frac{30 \cdot 20}{16 \cdot 21} = 1{,}79 \tag{2.22}$$

By connecting both shafts 1,00 gear ratio of the 4-th gear can be reached.

It has to be noted that these are gear ratios of the gearbox alone – the ratio of the first, second and third gear in relation between engine and wheels will be affected by the final drive ratio.

2.4.2 Synchro-mesh

Synchro-mesh mechanism equalise the different rotational speed of gear that are about to be meshed to enable smooth gear change.

Considering the working principle the synchro-mesh mechanisms can be separated on :

- Simple synchro- mesch
- Blocking synchro- mesch gear
- Progressive synchro- mesch gear

2.4.3 Classical gearbox

Classical gearbox is referred as a manual, fixed number of ratios gearbox. Basic elements of a manual gearbox are the following:

- Input shaft
- Output shaft
- Gears – placed on shafts
- Synchronizer – present in all modern cars

Gears are being changed by meshing different pairs of gears, as it is shown on the figure below

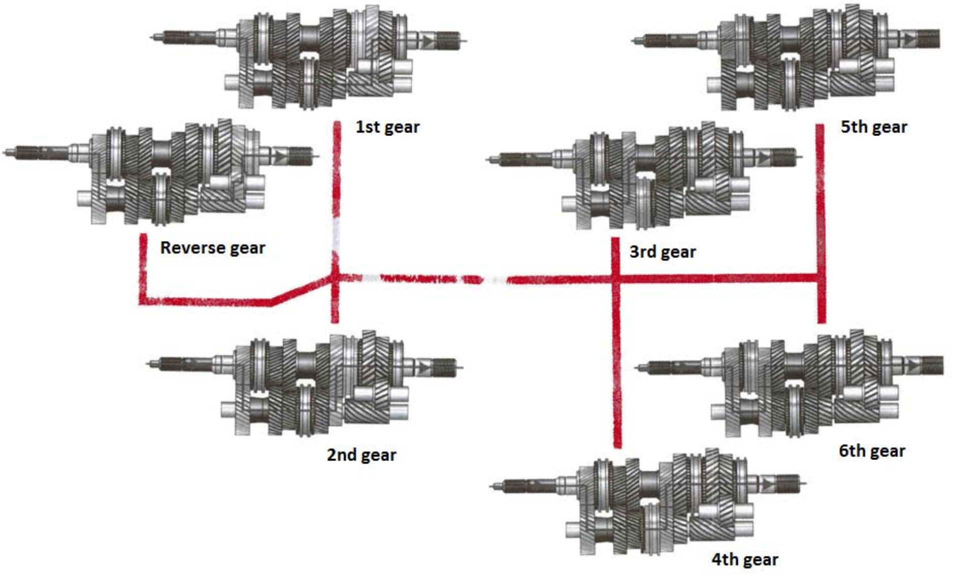

Figure 2.17 Gear change

2.4.4 Automatic gearbox

Automatic gearbox is built in a different way than the manual. First of all it uses planetary gear mechanism. By stopping or releasing specific gears in the planetary system different ratios are achieved.

The main elements of a hydraulic automatic transmission:

- Planetary gearset
- Torque converter
- Clutches and band
- Pump
- Valve body
- Hydraulic and lubricating oil

A cross view through the automatic gearbox can be seen on figure 2.18

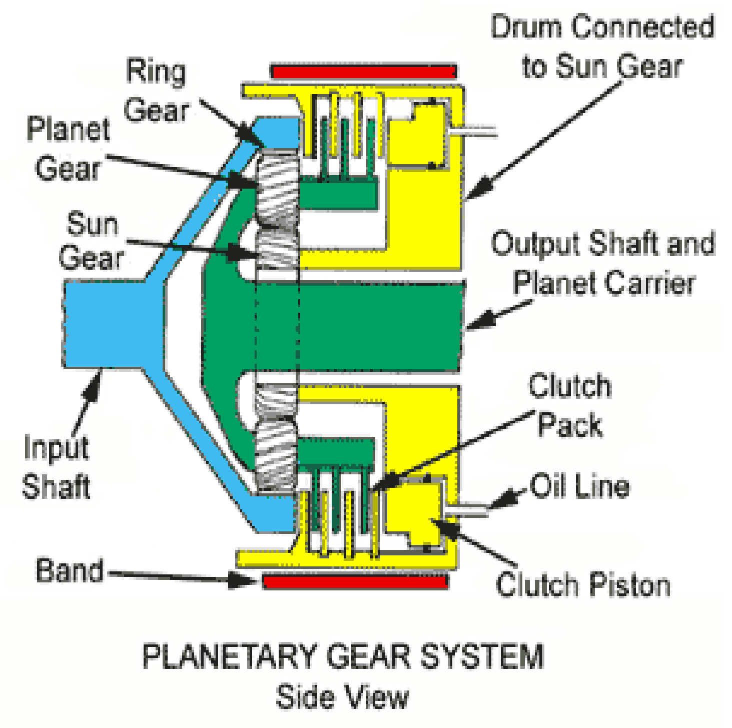

Figure 2.18 Main elements of a automatic gearbox with planetary gearset

2.4.5 Semi-automatic gearbox

Semi-automatic gearbox is basically a classical manual gearbox with some functions automated. There are two main approaches: automatic clutch actuation and manual gear change or manual clutch and automatic gear change. The first approach is present in the majority of the semi-automatic gearboxes available on the market.

Dual-clutch gearbox

Volkswagen introduced DGS gearbox is semi-automatic – the clutch is automated and the gear can be changed manually of automatically – thanks to electronic controller. What is interesting about the DSG gearbox is the fact that it uses two clutches – one for even gears and second for odd gears. Thanks to that the gear change time can be reduced significantly, as during the change one clutch is disconnected and the second connected – has the next gear prepared.

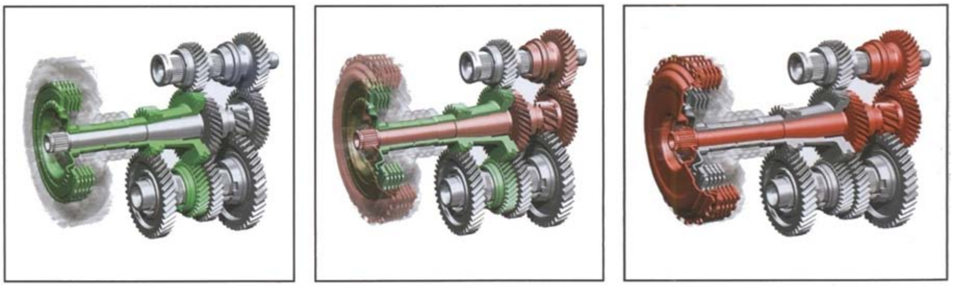

Figure 2.19 Gear change in DSG gearbox

Specific gears are connected to specific clutch – in can be seen on the following figure:

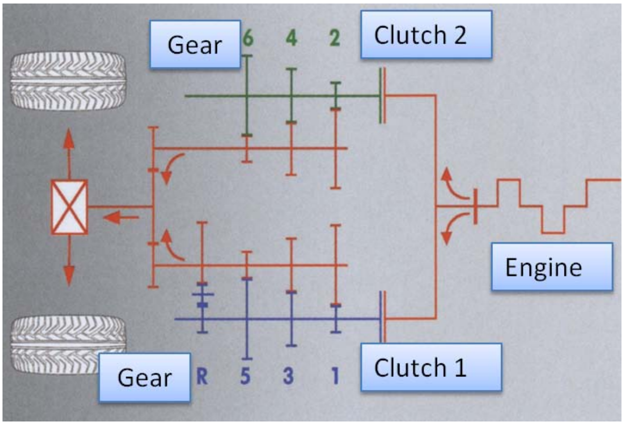

Figure 2.20 DSG layout

2.4.6 Continuous variable

Because of the need for speed change of the vehicle – gearboxes are required. Combustion engines used in most cars has a narrow work revolutions range. It can be seen from the engine map that high momentum is reached in very narrow rpm range – therefore to keep the engine in the most efficient rpm range frequent gear changes are required. Unfortunately even 6 or 7 gears still results in operating only near the maximum efficient engine rpm. To keep the efficient engine speed at all-time Continuous Variable gearboxes were introduced. The working principle is based on two pulleys with variable width. The basic principle can be seen on the figure below:

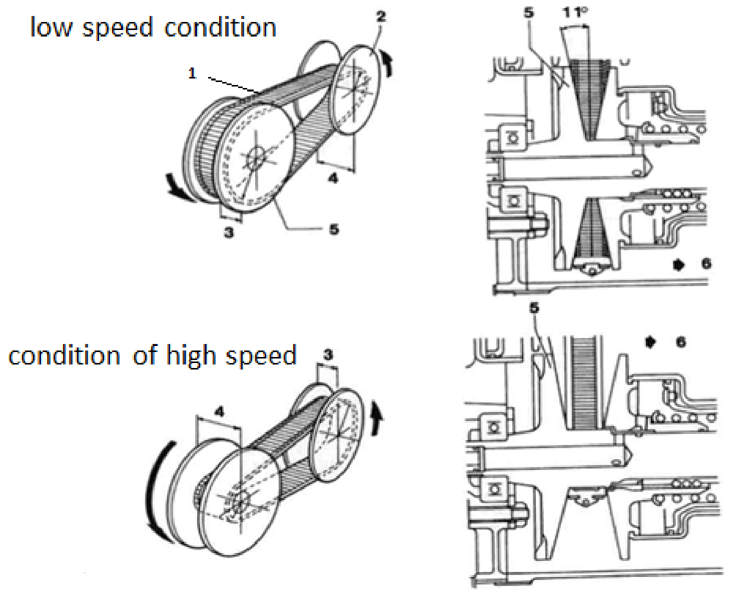

Figure 2.21 Continuous variable gearbox

1) Belt
2) Main belt pulley
3) Narrow grove
4) Wide grove
5) Secondary belt pulley
6) Housing/movable housing

2.5 Propeller shafts and universal joints

Propeller shafts are used to transfer torque between driving elements and wheels. There are two main shafts used in powertrain system: longitudinal shafts and drive half shafts connected to driven wheels. As the shafts rotate at very high angular speeds – for each one of them the maximum rotational speed should be calculated. It can be done with the following equations:

$$\omega_{\max} = 2 \cdot 10^5 \frac{\sqrt{d_o^2 + d_i^2}}{l^2} \tag{2.23}$$

d_0 - outside diameter of the driveshaft tube

d_i - inside diameter of the driveshaft tube

l - driveshaft length

Joints

As in most cases elements which are connected with a driveshaft are not aligned or change their relative position during wheel motion – joints allowing miss-alignments are required. There are two main types of joints:

- Asynchronous – cardan universal joint;
- Synchronous – constant velocity joints.

In asynchronous joint speed variation occurs – when the joint is angled, during rotation the speed on the output shaft is rising and falling every 90 degrees of rotation.

In the powertrain system two main constant velocity joint are being used:

- Rzeppa constant velocity (CV) joint;
- Birfield CV joint – improved Rzeppa construction.

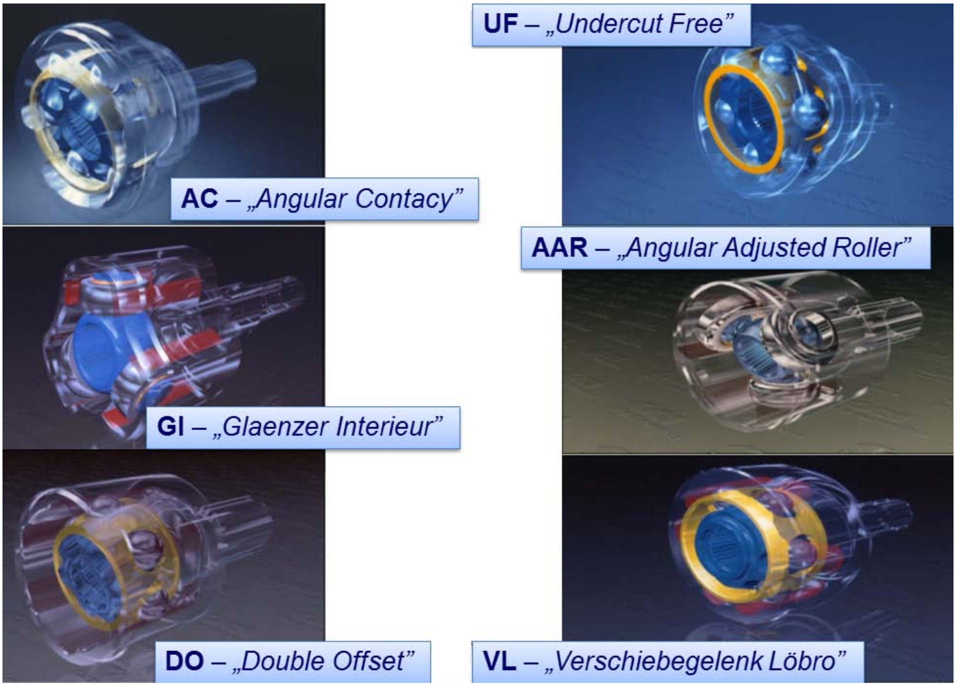

Figure 2.22 Types of constant velocity joints

2.6 Differential

Differential is a device which divides the power on two half shafts and enables rotation of two axle wheels with different speed.

The standard differential mechanism assembly contains:

- Two crown wheels – large gears providing power-dive for half axle
- Four (or more) planet gears – small
- Two slip-rings
- Cross
- Cage

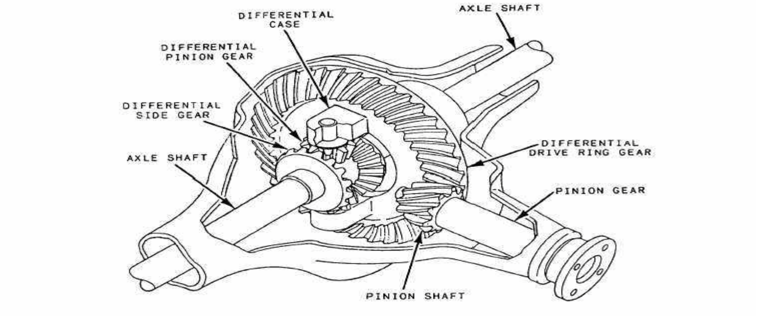

Figure 2.23 Differential elements

Rotating input shaft (pinion shaft on fig. 1.16) is driving the ring gear, thus rotate the differential case. The pinion gears drive side gears. When the pinion gear is not rotating the rotational speed of each side gear and therefore of each wheel is the same – the differential is not working. This only happens in straight line acceleration with no slip/equal slip in the tyres. When the side gear is rotating – wheel speed on each side is different.

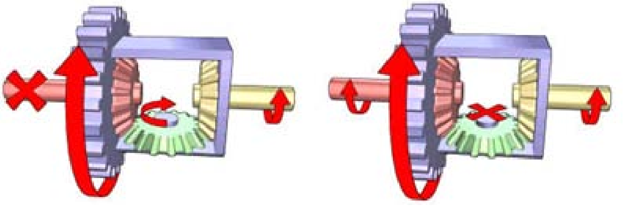

Figure 2.24 Differential working principle

2.6.1 Basic powertrain calculations

The schematic representation of a differential is shown on Fig. 2.25.

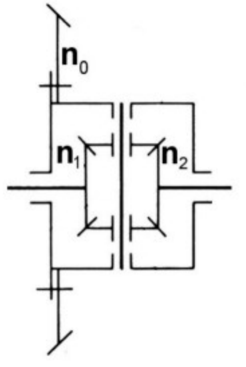

Figure 2.25 Differential schematic

The relation between the gear and the differential is described by the following equations.

- Relation between wheel speeds

$$n_1 = n_0 + \Delta n$$
$$n_2 = n_0 - \Delta n \tag{2.24}$$

Relation between wheel speed and cage rotational speed

$$n_1 + n_2 = 2n_0 \tag{2.25}$$

The differential

The torque in the differential is calculated by finding the lower value from two possible equations:

$$M_{o2} = M_{eMax} \cdot i_1 \cdot i_a \cdot i_f \tag{2.26}$$

M_{eMax} – maximum engine torque

i_1 – gearbox 1st gear ratio

i_a – additional gearbox ratio (if used)

i_f –final drive ratio

The resistance torque resulting from the tyre traction can be calculated with the following formula:

$$M_\mu = \mu \cdot r_d \cdot W_d \cdot \frac{1}{i_z} \tag{2.27}$$

Where:

μ – friction coefficient

r_d – dynamic wheel radius

W_d – driven wheels load

i_z – wheel reduction gear

Initial calculation of outer reference diameter of the half axis crown wheel:

$$d_2 = (0{,}57 \div 0{,}657)\sqrt[3]{M_{o2}} \quad [cm] \tag{2.28}$$

Moments on both half-axles can be calculated from the following equation:

$$\begin{aligned} M_1 &= 0{,}5M_0 - M_t \\ M_2 &= 0{,}5M_0 + M_t \end{aligned} \tag{2.29}$$

M_1, M_2 - half-axles torque

M_0 - torque on the differential cage

M_t - frictional torque of the differential

The relation between wheel moment and frictional moment of the differential:

$$M_1 - M_2 = 2M_T \tag{2.30}$$

2.6.2 Torsen differential

Torsen differential was designed to improve vehicle traction. Thanks to its construction when one wheel is slipping the torque is being transferred to the other wheels which has traction. The schematics can be seen on figure 2.26.

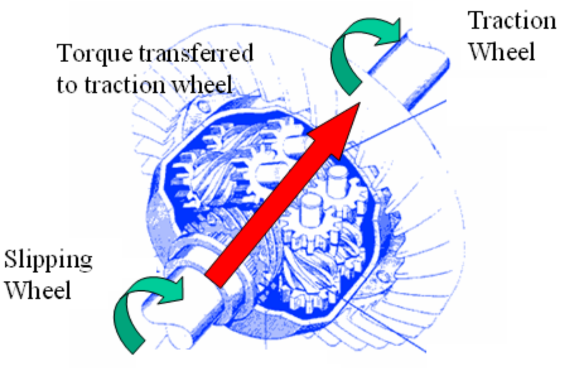

Figure 2.26 Torsen differential

2.6.3 Limited slip differential (LSD)

The main purpose of LSD is to prevent wheel slip. As in a typical conical differential when one wheel starts to slip – all the torque is being transferred to it – therefore there is no driving torque. This is why locking of the differential was introduced. When one wheel stars slipping one of clutch starts to overcome this by means of its frictional force.

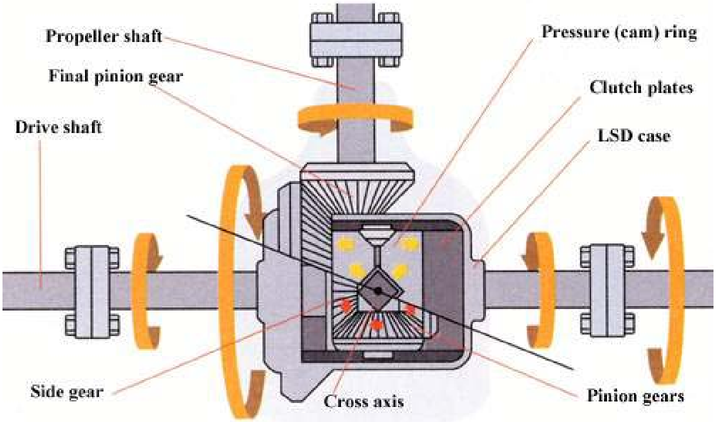

Figure 2.27 Limited slip differential (LSD)

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3 STEERING SYSTEM

Steering system role is to control vehicle movement in all conditions. It is thereby very important that the driver receives accurate information about the force relations between the wheels of the car and the road. It has been shown that the driver reacts much faster on haptics than on optics. Therefore a distinct assignment of the angle of the steering wheel to the position of the wheels and only smallest slack in the steering chain between steering wheel and wheels is very undesirable.

Main functions of the steering system are:

1) High mechanical efficiently for the force transmission from the steering wheel into the wheels
2) Stabilization properties, which enables wheels to return to their initial position after cornering
3) Reduced to minimum impact force transfer from the wheels to the steering wheel caused by road irregularity
4) Correct vehicle kinematics while cornering
5) Maintain the relationship between force applied to the steering wheel and the driven wheels turning moment

3.1 Forces on the wheel

There are several forces acting on the wheel which were mentioned before. Let’s consider some details.

3.1.1 Vertical force

Vertical force is an effect of vehicles weight and dynamic load change during motion.

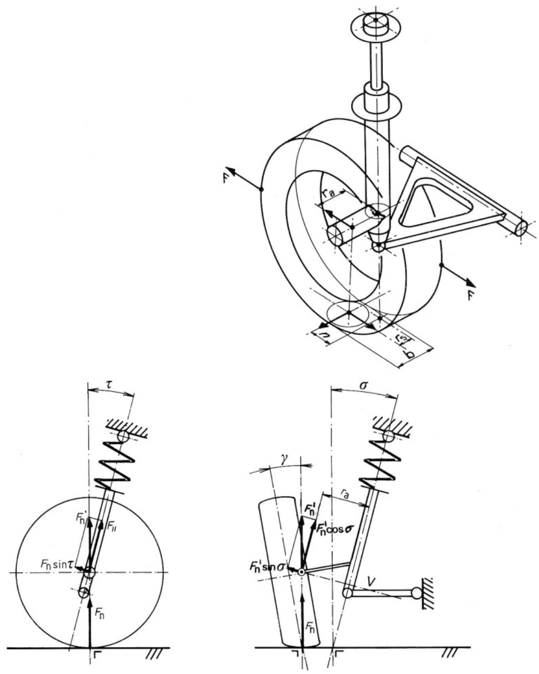

Figure 3.1 Vertical force

For the type of suspension shown on fig. 3.1 - force F_n can be divided into force acting on the spring:

$$F_s = F_n \cdot cos\sigma \tag{3.1}$$

and force actuated on the wishbone:

$$F_w = F_n \cdot sin\sigma \tag{3.2}$$

3.1.2 Lateral force

Lateral force as an effect of cornering is acting with an leverage on the steering rod, over wheel turning axis.

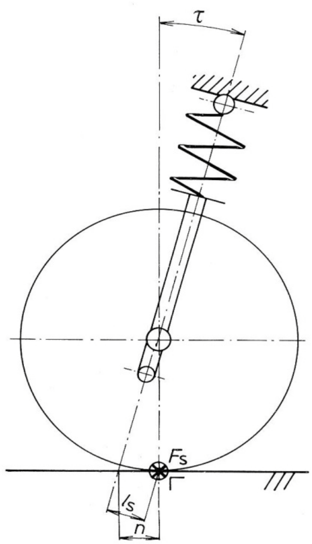

Figure 3.2 Lateral force

The momentum can be calculated with the following expression:

$$M = l_s \cdot F_s \tag{3.3}$$

Caster and wheel trail is connected with such relation:

$$n = l_s \cdot cos\tau \tag{3.4}$$

n – wheel trail

τ – caster angle

In most production cars caster angle is very low therefore the equation can be simplified to:

$$M = F_s \cdot n \tag{3.5}$$

3.1.3 Longitudinal forces

Longitudinal forces are caused by vehicles acceleration and braking. Those forces are causing the wheel to toe-in or toe-out – depending on the scrub radius.

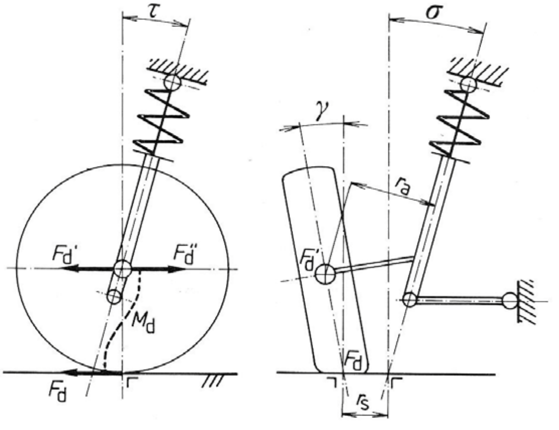

Figure 3.3 Longitudinal forces

Drive moment can be calculated using:

$$M_d = F_d \cdot r_{dyn} \tag{3.6}$$

The momentum that is forcing the wheel to turn can be calculated using one of the following equations depending on what data is available:

$$M_{Id} = F_d' \cdot r_a \tag{3.7}$$

$$M_{Ib} = F_r \cdot r_s \tag{3.8}$$

It can be seen that when the wheel scrub radius is equal to zero – there will be no momentum on the steering rods during braking. Usually there is a little positive scrub radius present to provide more “feel” of the vehicle.

Longitudinal force can be also caused by road irregularity – a hump.

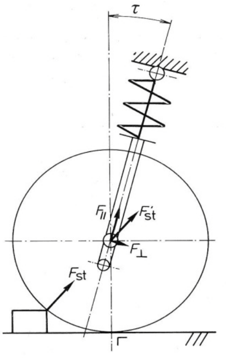

Figure 3.4 Longitudinal forces caused by road irregularities

Force resulting from this situation is equal to:

$$M = F_{\perp} \cdot r_a \tag{3.9}$$

3.2 Steering system layouts

Depending on the what kind of kinematics is desired in the steering system and how much space is available for mounting the elements – different layouts are being used (fig. 3.5).

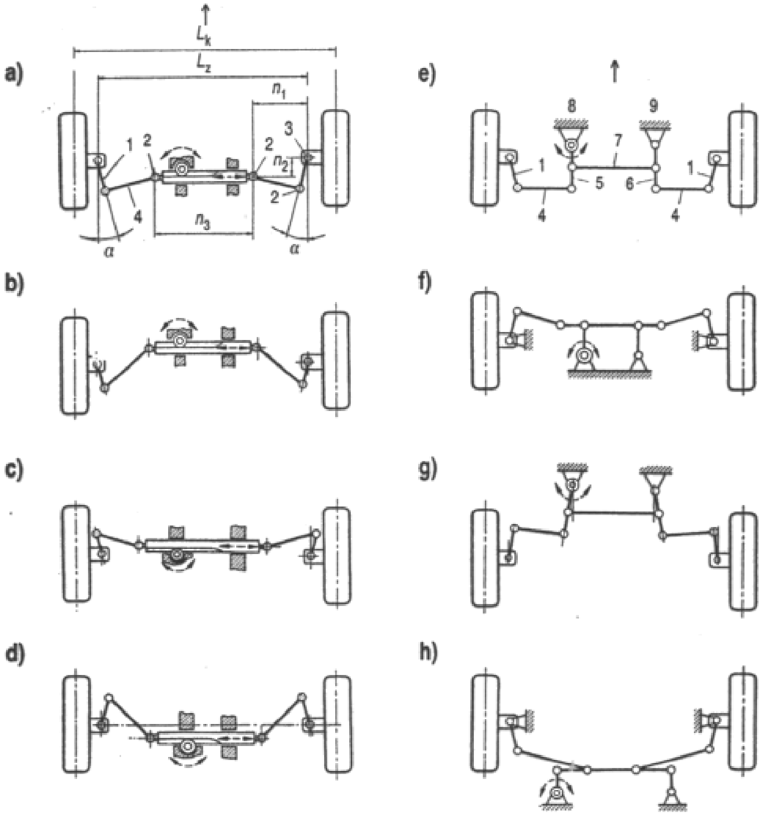

Figure 3.5 Typical steering system layouts

3.3 Steering system elements

Steering system consist of several elements:

- Steering wheel
- Steering shaft
- Steering column
- Steering gear
- Tie-rods

Let's consider some of them.

Steering gears

There are various solutions for steering mechanisms available on the market. Typical classical, mechanical solutions are shown in fig. 3.6 and 3.7

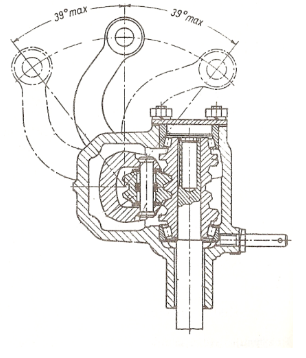

Figure 3.6 Worm gear

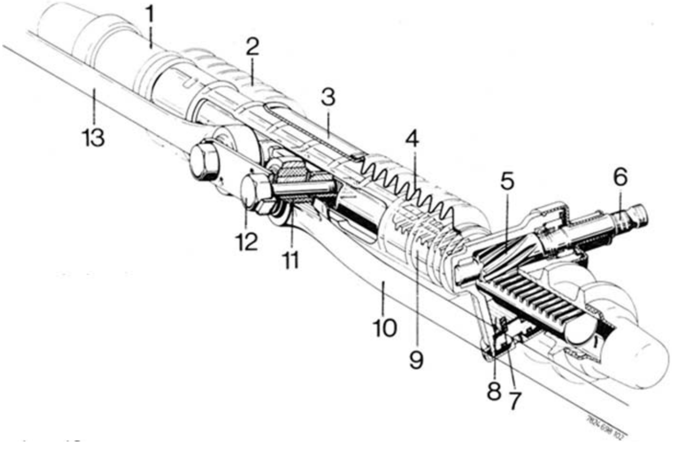

Figure 3.7 Rack and pinion: 1 - housing, 2 - dust cap, 3 - steel tube, 4 – dust cap, 5 – pinion, 6 – connection with steering column, 7 – spring, 8 – rack support, 9 – drag rod, 10 – silent bloc, 11 – bolt, 12 – drag rod.

3.4 Power steering

In the recent past decades comfort demands have increased and hence the mechanical steering was replaced by hydraulic servo-power steering. Two types are nowadays accepted: The hydraulic rack-power-steering which is the cheapest power steering (fig. 3.8).

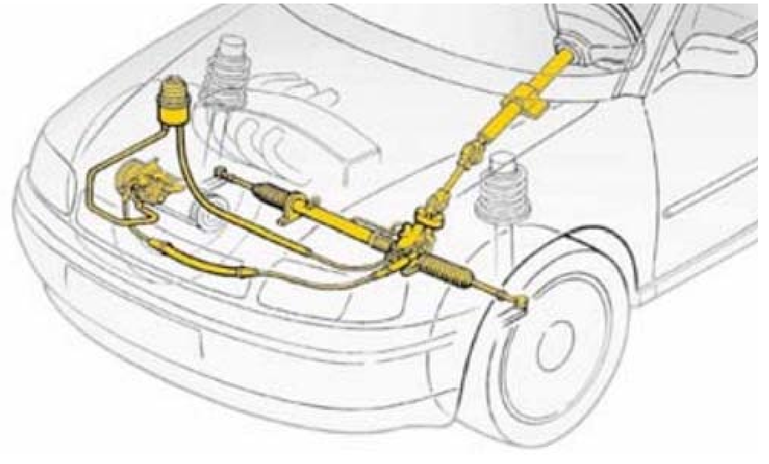

Figure 3.8 Steering system in passanger car

And the ball-nut hydro-power steering. Latter is nowadays only used in some SUV's. In heavy conditions where robustness and high chassis clearance is required ball-nut hydro-power steering will be the choice.

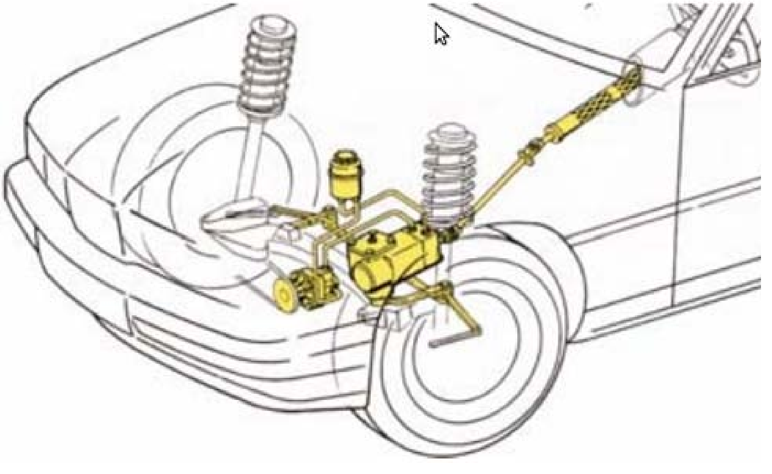

Figure 3.9 Power steering system

The steering servo pump, usually a vane pump, is driven with a v-belt from the crank shaft of the combustion engine. Hence the oil flow is linearly dependent on the rpm of the engine.

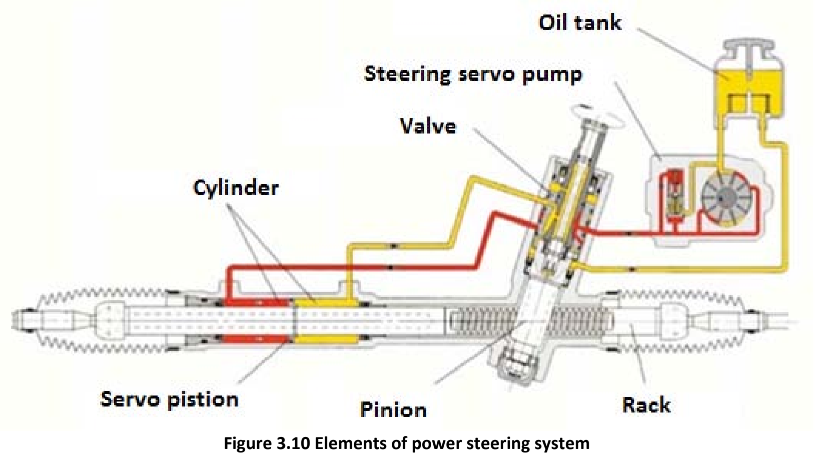

Figure 3.10 Elements of power steering system

The advantage of this system is its robustness and the fact that there is no need to maintain it during the whole life of the vehicle. On the other hand the big disadvantage is its inefficiency. During high engine rpm's, mainly at high speed on the highway, the steering servo pump deliverers the highest oil flow even though there is no need for steering work. Due to that higher fuel consumption results, which is not favourable.

To solve this problem, at least partly, the electro-hydraulic steering system was developed (fig. 3.11):

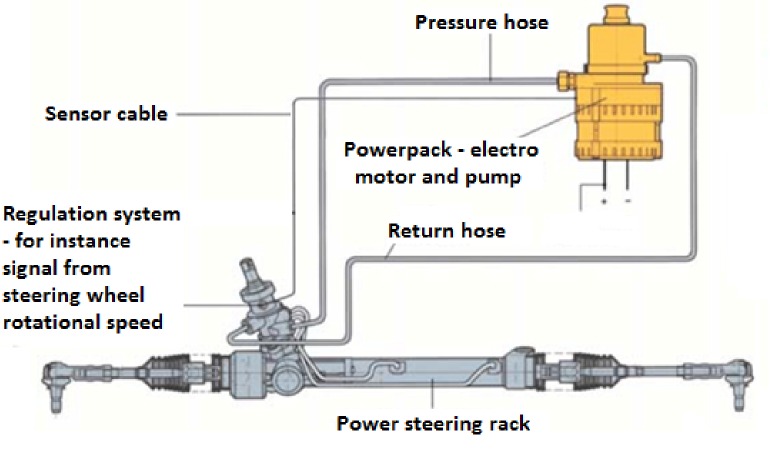

Figure 3.11 Electro-hydraulic steering system

With this system the efficiency, mentioned in the last paragraph, is increased already considerably. Instead of the direct drive of the pump from the crank shaft of the combustion engine so called Powerpack is used. The vane pump is driven by a electro motor. This unit (Powerpack: combination electro motor and pump) can be installed independently from the combustion engine, which is a huge advantage. The oil pressure and oil flow is thereby independent from the engine rpm. No steering work is required because the oil pump delivers only a little stand-by flow. Hence a reduction of the fuel consumption of about 0.3L/100km is possible.

However the electro-hydraulic power steering is confined to lower middle class cars like for instance Opel Astra. For higher requirements on the drive dynamics quite sophisticated regulation strategies of the pump control are necessary. In higher middle class cars there is also an approach to use a variable displacement pump in order to save more fuel. Standard steering is usually using fix steering characteristics which is a compromise between low steering moments on the steering wheel for parking manoeuvres and high moments for high speeds on the highways. For luxury cars with high steering axle loads the Servotronic- or Parametersteering was developed. This system allows separate steering characteristics for every drive condition. Namely for parking low moment on the steering wheel is required and high moment at high speeds. Hence the steering moment is a function of the speed. This provides a secure drive feeling. Everything was received thanks to mechatronics.

Figure 3.12 Pump and speedometer connection

As it can be seen in figure 3.12 the pump is controlled with the parameter ”speed” directly from the speedometer.

3.4.1 Variable steering transmission

The construction of the steering gearing should be able to bear high forces on one hand and at the same time it should guarantee low clearance, elasticity and as less friction as possible. The handling of a vehicle is in a great deal dependent on the transmission ratio between the pinion and the rack. In practice for every vehicle an individual transmission ratio for the steering gearing is defined. Depending on the type of vehicle two to three revolutions of the steering wheel from max left to max right position of the wheels is considered as comfortable.

The geometrical design of the gearing pinion-rack is extremely sophisticated. In case of failure of the power steering support the gearing has to bear very high mechanical forces. To achieve only one to 1.5 revolutions of the steering wheel from the neutral position to one of the maximums the pinion may only have 6 to 10 teeth whereas involute gearing should be chosen. To get a good comfort feeling high efficiency of the gearing to steer into a curve and a low efficiency of the gearing for turning the steering wheel back into neutral position to damp unevenness's of the road. Therefore very complicated calculation methods for the dimensioning of the gearing are applied.

To increase the handiness of sports cars in the city and on windy roads a special involute gearing for the rack was developed. The transmission in the middle area is rather a bit indirect. Towards the maximum to the right and left the transmission ratio is getting much more direct. Hence at high speeds a not to quick guidance behaviour from the neutral position is achieved.

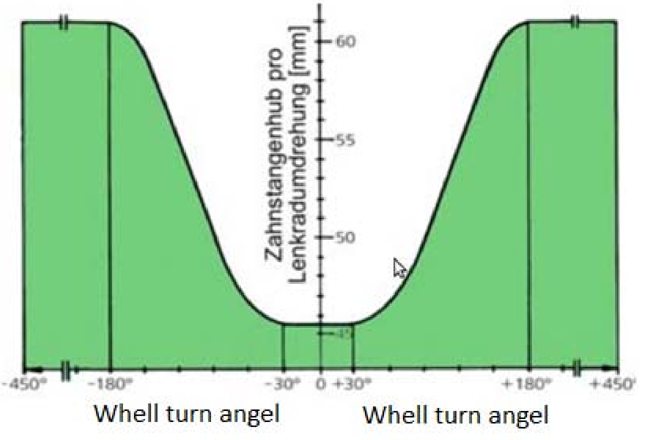

Figure 3.13 Transmission ratio

The pinion has a standard involute gearing and the rack a special variable involute gearing. Through moving of the pitch circle of the pinion from dedendum at neutral position to addendum at the maximum turning angle of the wheels is the stroke of the rack in the area of its end position bigger than in the area of its neutral position.

3.4.2 Dosing of the power steering support

The dosing of the power steering support is controlled by the driver over the rotary slider. The steering moment is defined by of the rotary bar which torsion angle or also torsion moment is linearly dependent on the steering resistance of the wheels. Accordingly to the diameter of the rotary bar and the geometry of the grooves on the rotary slider and control bush the valve characteristics can be adapted to the vehicle (see figures 3.14 and 3.15).

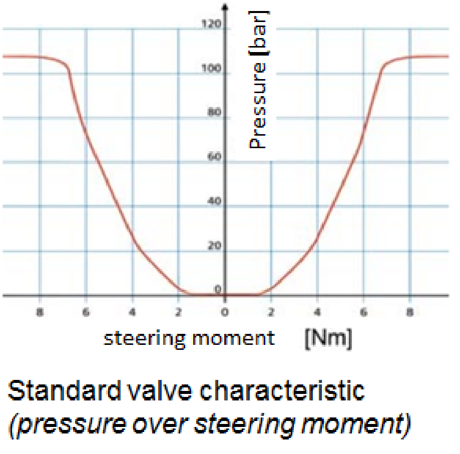

Figure 3.14 Pressure and moment relation

The steering system servotronic as well bases upon the rotary slider valve. According to the speed a different steering characteristic is provided in this case – see following figure:

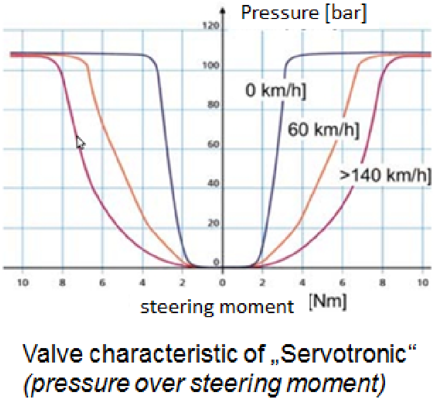

Figure 3.15 Pressure-moment relation at different speeds

While parking the car, at very low speeds, already a little moment on the steering wheel produced a high oil pressure for the power steering.

3.4.3 Work principle of a rotary slide valve

When turning the steering wheel in order to turn the wheels the driver is supported by the powersteering. The steering gearbox of the rack-and-pinion-steering consists mainly of a rack and a pinion with bearing elements, piston and a rotary slide valve (see figure 3.16).

Figure 3.16 Contemporary steering system

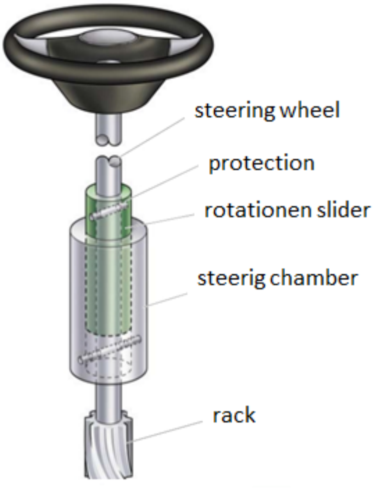

Figure 3.17 Steering wheel and joined elements

As it can be seen in fig: 3.17 the rotary bar is fix connected with a pin on it is upper end with the rotary slider. On it is lower end the rotary bar is connected with the pinion and the control bush with a pin. Rotation on the steering wheel initiated by the driver causes a moment on the rotary bar. Latter is twisted due to that. The relative position of the grooves and the crossing channels in the rotary slider and the control bush changes.

3.4.4 Electro mechanic steering

In an electro mechanic steering the servo effect is carried out with an electrical instead of a hydraulic system. It also uses the rack-bar principle like the hydraulic system. Hence lots of proven mechanic parts can be used here as well. A torque sensor detects the torque applied on the steering wheel and a controller tells the electrical motor to support. Latter either applies torque on the steering collar or a force on the rack, depending upon the system.

The energy density of the electric system is lower than of a hydraulic system. Therefore the package is more critical. Depending upon the type of car 5 different design possibilities were developed and applied.

Drive on the Steering column (fig. 3.18): This power steering concept is devoted only for parking support. At higher speeds the electric motor will be turned off . Because the system is mounted directly behind the steering wheel, inside the passenger cell, it has to bear less temperature gradients which makes lots of parts cheaper. A drawback of this concept is that the support torque acts on a high position in the steering column where up to now only little moments were applied. Therefore parts like for example the rotary bar need to be dimensioned thicker, heavier and more expensive.

Figure 3.18 Drive on steering column

Servo drive on the pinion (fig. 3.19): In this case the steering column can be produced cheaply because only load comes from the steering moment from the driver. Only the pinion should be dimensioned stronger. This version cogitates a drive on the steering column in points like steering properties and production costs. Figure 3.19 on the left side shows the servo drive on the pinion.

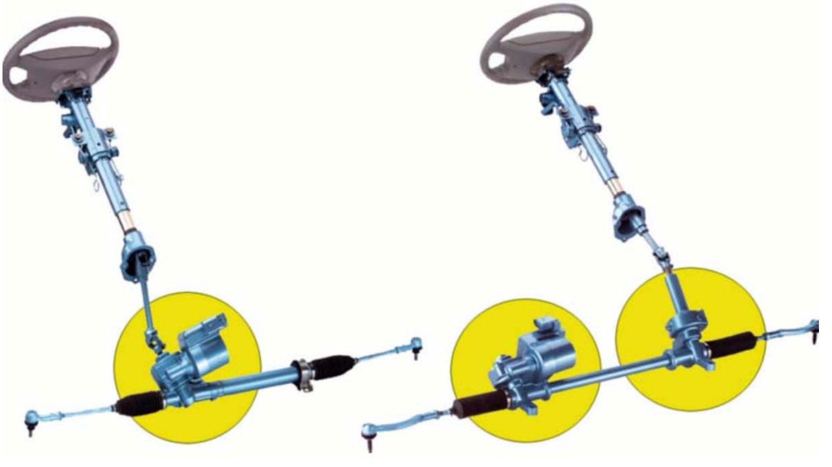

Figure 3.19 Drive on the pinion

The drawback in this version is that the electric motor is placed where the pedals (clutch, brake, accelerator) are placed. Furthermore it also produces enormous risk of injuries in case of a crash. Therefore till now only a few cars were supplied with this version.

Double pinion drive: To solve the problem of the injury risk and the stronger pinion the double pinion drive, right side - fig. 3.19) was developed. The electric motor is installed on the right side of the rack. This system from ZF-Lenksysteme was built into the 2003 Golf-platform and the 2005 Passat. But if there are still higher servo moments are required then a pinion drive is overstrained.

4. Axle-parallel drive: The electric motor is mounted on the opposite side. The servo moment from the electric motor drives a ball screw mechanism: Around the rack is a helix groove. The force is initiated over the ball screw, which is driven over a belt or gearing by the electric motor, over the hardened steel balls to the grooves in the rack, which is moved either to the left or the right side. The ball screw is able to transfer very big servo moments (fig 3.20).

Figure 3.20 Axle - parallel drive

Pipe-shaped electric motor around the rack: For special applications a pipe shaped electric motor around the rack can be used instead of the axle-parallel drive. The moment is applied to the ball screw via planetary gearing. The pipe shape of the electric motor decreases the dynamics and the power density (fig. 3.21).

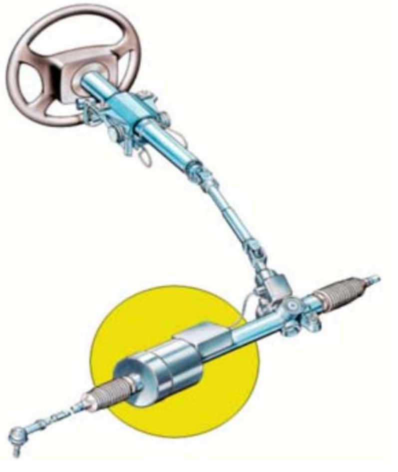

Figure 3.21 Electric motor mounted around the rack

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4 BRAKE SYSTEMS

4.1 Types of systems

According to the regulations there are three types of brakes required in car: main brake, parking brake and emergency brake. Usually parking brake is also used as emergency brake while driving – therefore the number of systems is reduced to two.

Brake systems may also be divided into different circuits configuration (fig.4.1). There are 5 systems used in the car industry: X, II, HI, LL, HH. Although X and II are the most commonly used. One of the reasons is that if one wheel brake fails (for example due to damaged wheel, caliper, etc), only one circuit will be out of order and the driver can continue braking with the second circuit.

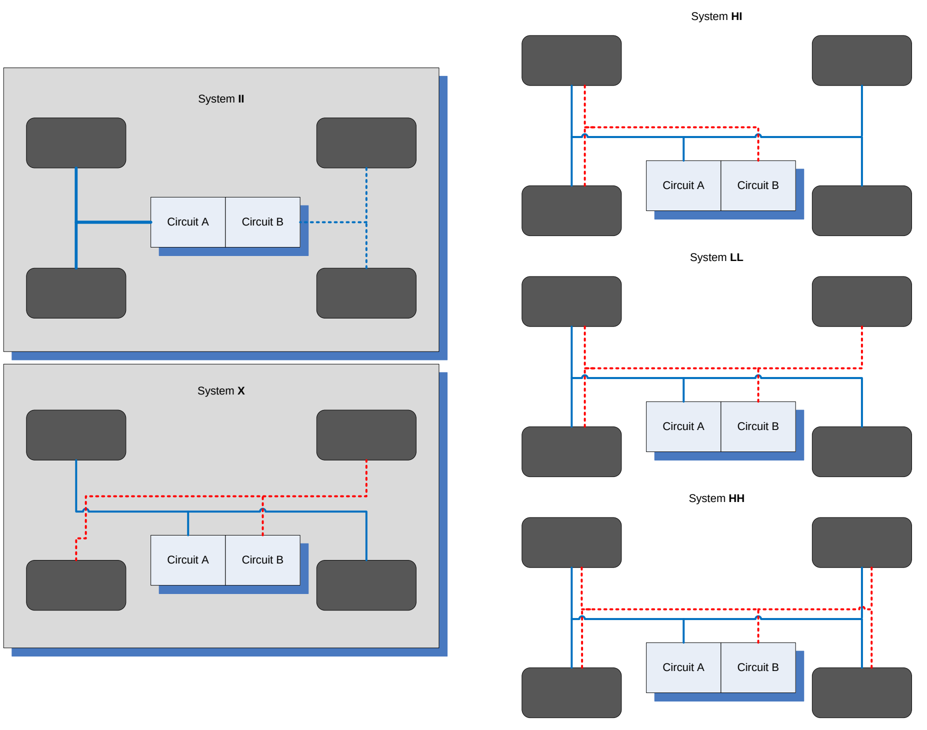

Figure 4.1 System layout

4.2 Brake system elements

Brake system consist of the elements showed on fig. 4.2:

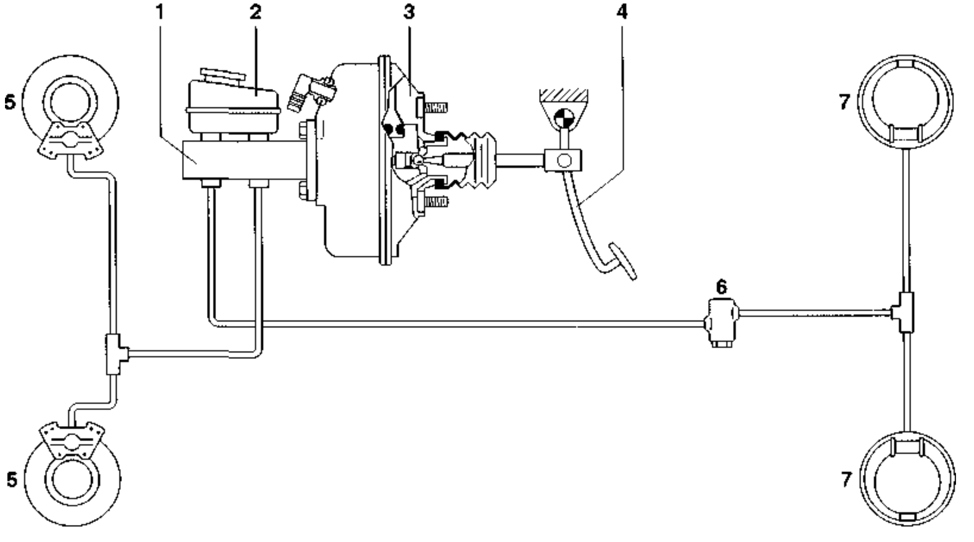

Figure 4.2 Brake system layout: 1 – master cylinder, 2 – brake fluid reservoir, 3 – power brake device, 4 – brake pedal, 5– brake disc, 6 – brake proportioning, 7 – drum brakes.

4.2.1 Wheel brake mechanisms

Basic brake system consist of: pump, power system (if used), lines, calipers-disc or drums, pressure regulators.

There are two widely used wheel brake mechanisms: more modern disc-brake solution and drum-brake. There are several types of drum brakes: simplex, duplex, duo-duplex, servo, duo-servo (fig. 4.3).

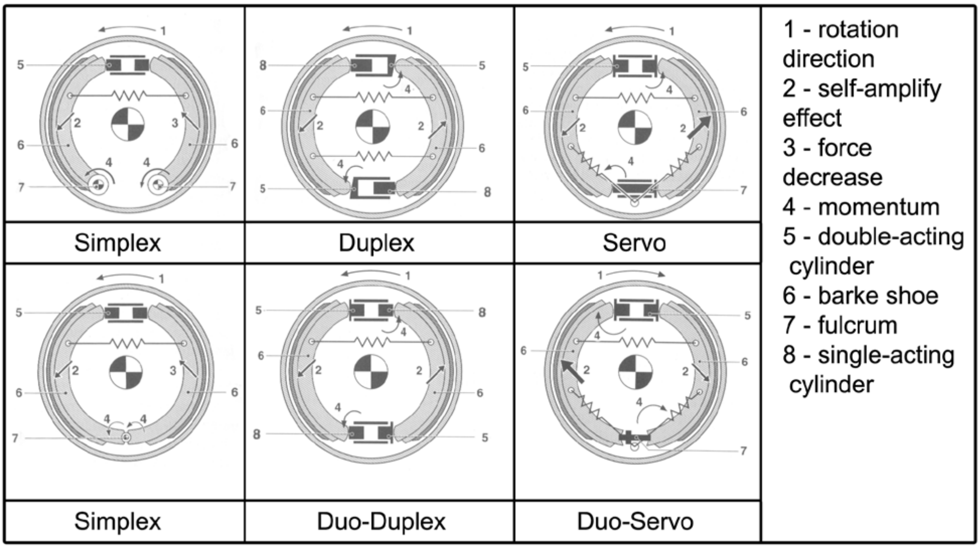

Figure 4.3 Different drum brake mechanisms [42]

Specific elements of a drum brake are shown on fig. 4.4.

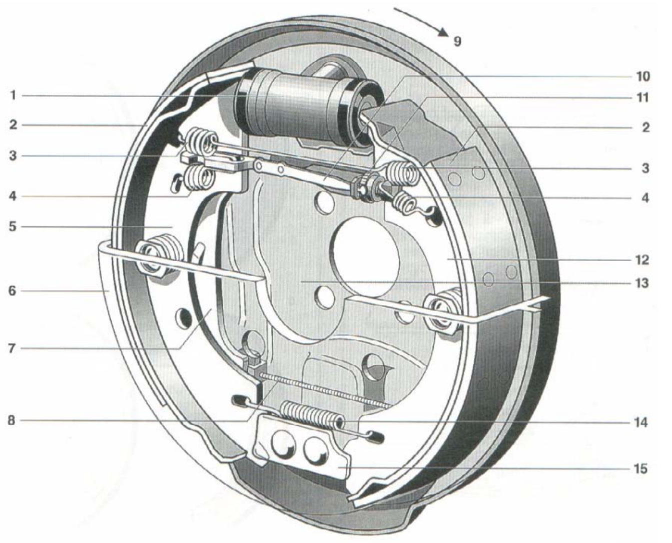

Figure 4.4 Elements of a drum brake: 1 – hydraulic cylinder, 2 – friction lining, 3 – spring, 4 – regulation mechanism spring, 5 – brake shoe (backward), 6 – brake drum, 7 – emergency brake leaver, 8 - emergency brake linkage, 9 – drum direction of rotation, 10 – regulation mechanism thermal element, 11 – regulation nut, 12 brake shoe (concurrent), 13 – carrier, 14 – spring, 15 – brake shoe support [2]

Two different calipers are being used in disc brakes: fixed caliper and floating caliper. Fixed caliper is used in heavy or high performance cars. Floating caliper advantage is that it is less expensive, more tolerant of rotor imperfections and it is smaller. On the other hand floating caliper can handle lower braking forces and is more vulnerable to dirt – the mechanism can get stuck and keep the brake engaged after the pedal was released. One the fig. 4.5 typical disc brake mechanisms are presented.

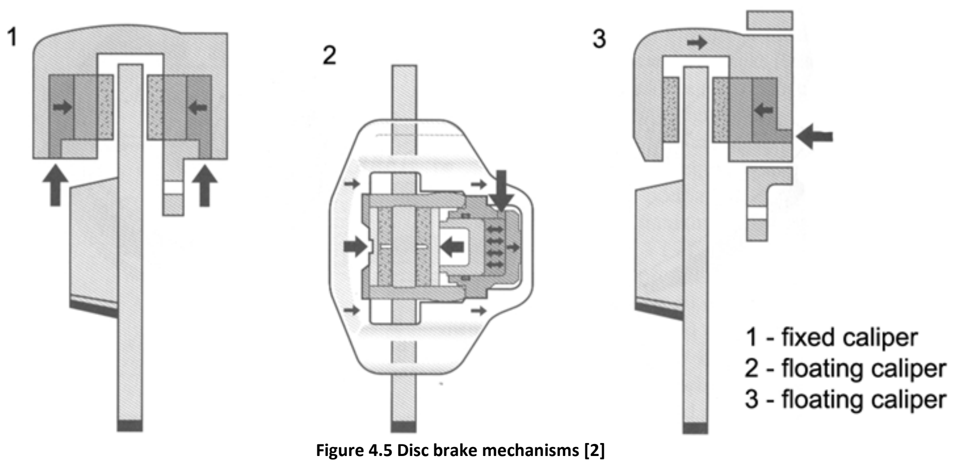

Figure 4.5 Disc brake mechanisms [2]

4.2.2 Power brakes

Because of high weigh of cars and trucks power brakes were introduced to increase safety and drivers comfort. There are two main types of brakes booster used in the industry.

- Vacuum brake booster (fig. 4.6)
- Hydraulic brake booster (fig. 4.7)

The first solution uses vacuum either from the engines intake system or from a separate pump to increase the pressure on the master cylinder – therefore increase braking force.

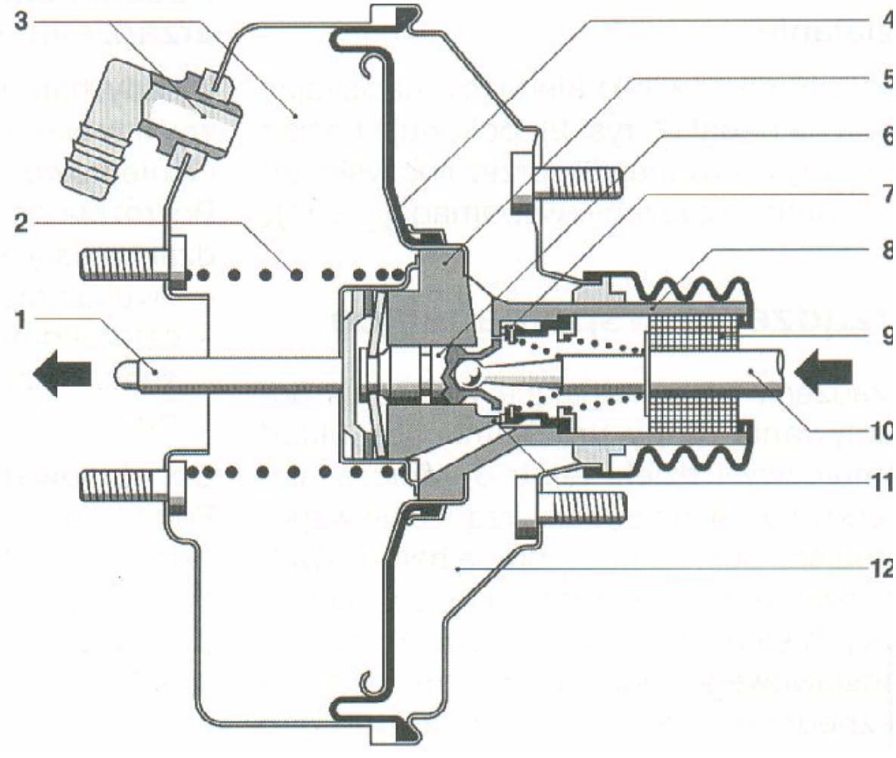

Figure 4.6 Two-chamber brake booster: 1 – pushrod (force acting on the master cylinder), 2 – spring, 3 – partial vacuum chamber, 4 – membrane with shield, 5 – work piston, 6 – control piston, 7 – valve, 8 – valve housing, 9 – air filter, 10 – piston rod, 11 – valve seat, 12 – work chamber. [2]

Apart from two-chamber boosters there are also four-chamber booster being used. Such solution provides higher pressure increase.

The second solution uses hydraulic pomp to increase pressure in the brake system. Such solution is usually used tighter with other systems which require pressure – like power steering. Higher pressures can be reached with hydraulic booster.

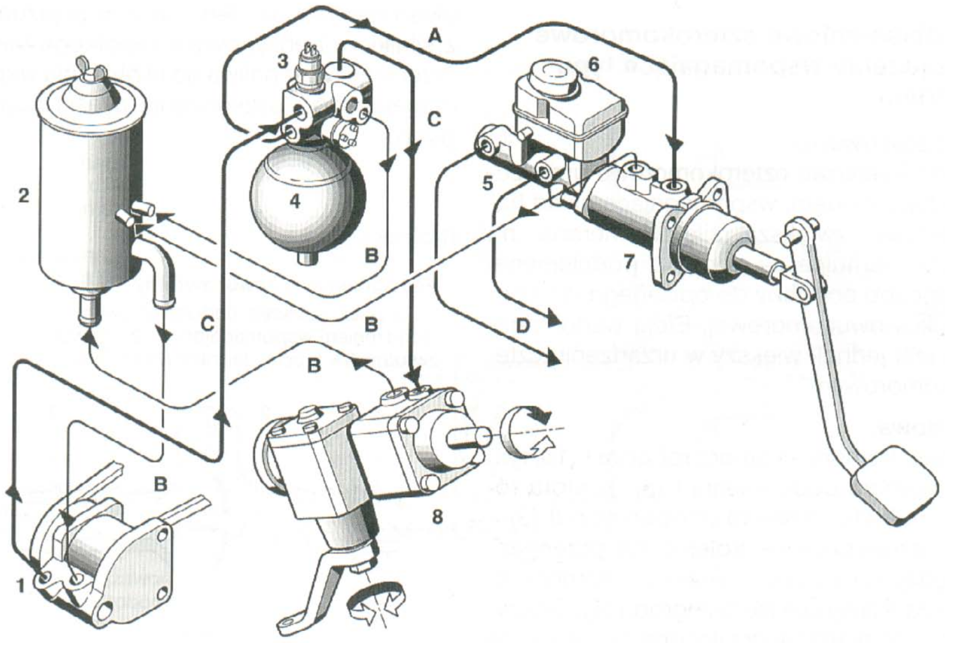

Figure 4.7 Hydraulic power brakes layout: 1 – hydraulic pump, 2 – tank with air filter, 3 – flow regulator (pressure operated), 4 – hydraulic accumulator, 5 – master cylinder, 6 – brake fluid reservoir, 7 – hydraulic brake booster, 8 – hydraulic steering booster, A – power brake circuit, B – oil supply lines, C – power steering circuit, D – brake system circuit. [2]

4.2.3 Brake proportioning

Due to dynamic longitudinal weight transfer maximum transferable brake force of the rear axle is lower than in the front. Therefore there is a need for reducing rear brake force to prevent locking of the rear wheels – which would lead to vehicle stability loss.

Brake force reducing valves are being used in systems with separate rear circuit (like II – front-rear), and in tandem systems with common connection point (like X systems).

There are three main types of pressure reducing valves:

- Pressure sensitive (fig. 4.8)
- Load sensitive – its position depend on the load on the rear axle (fig. 4.9)

- Acceleration sensitive – detects vehicles acceleration and provide required pressure reduction (fig. 4.10)

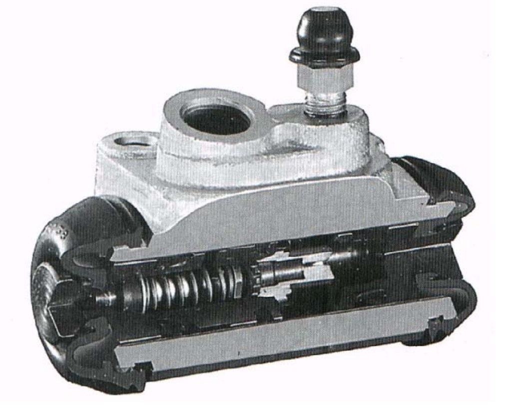

Figure 4.8 Pressure sensitive pressure reducing valve [2]

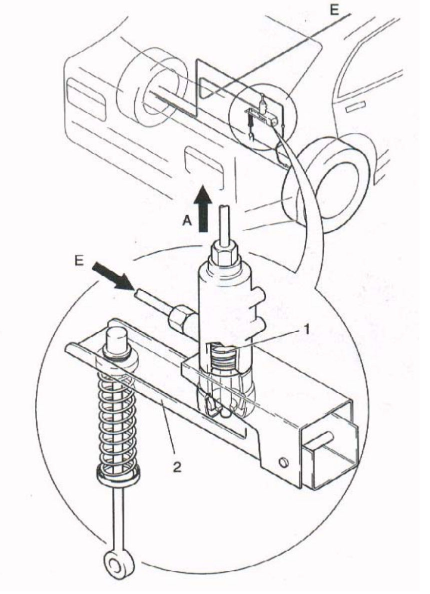

Figure 4.9 Load sensitive pressure reducing valve [2]

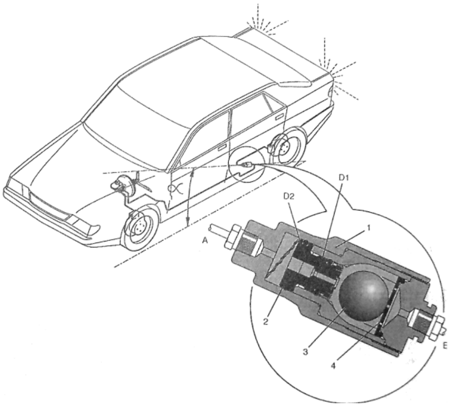

Figure 4.10 Acceleration sensitive pressure reducing valve [2]

4.3 Safety systems

4.3.1 Anti-lock brakes

ABS (Antiblocking Brakeing System) prevents the locking of the car's wheels during emergency braking (fig. 4.11). The essence of ABS is to prevent the continuous blocking of the wheel, and the latest solutions to maintain the partial slip. ABS consists of three basic elements: sensor, control unit and modulators. Sensors placed in the wheel hubs record the revolutions and send a signal to lock the wheel to the control unit. The latter is a fast-working computer, which is based on information from the sensors and next send signals to modulator. Modulator is a solenoid valve which decreasing or maintaining break pressure in brake cylinders.

Figure 4.11 Comparison of car steerability with and without ABS

4.3.2 Brake Assistant System

BAS (Brake Assistant System) is a system of emergency braking assistance in a emergency situations. This system works in cooperation with ABS. The detects situations where the driver wants to brake quickly, then BAS increases maximum pressure in the brake system to get a bigger braking power. In some vehicles during the operation of the BAS also include emergency lights to warn other driver about sudden braking. BAS placed between the brake pedal and master cylinder sensor detects the driver’s foot movement vehemence (and therefore an emergency situation that requires – which suggest the urgency of pressing the pedal) and induces electronic means for the maximum pressure in the break system. The idea of the system is the result of research which shows that most drivers even in an emergency, press the brake pedal with too little force to the braking distance has the minimum length. With the BAS’s attention, its length may be shortened from 20 to (in extreme cases) even 30 %.

4.3.3 Pyrobrake

Pryobrake is an emergency brake engaging system. By the use of small amount of explosives brake system pressure can be build up very quickly. The system scans the road in front of the vehicle and simultaneously calculates distance to an object and time remaining to crash. After the system decides that the car had reached a point when crash is unavoidable – the explosives are detonated, fast increase in brake fluid pressure immediately enables to brake the vehicle as fast as possible to reduce crash consequences.

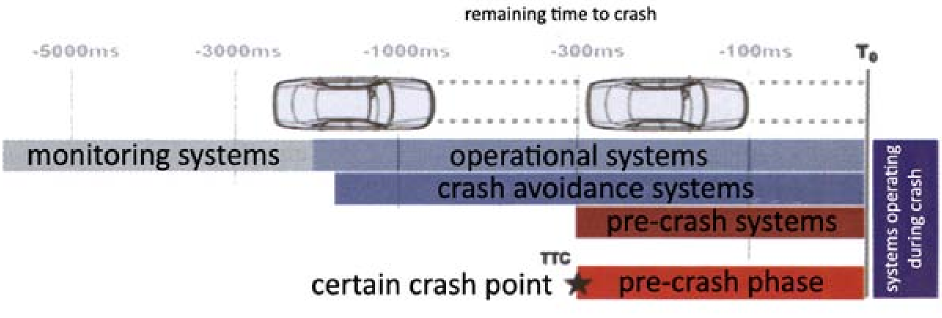

Figure 4.12 Pyrobrake initiation scheme

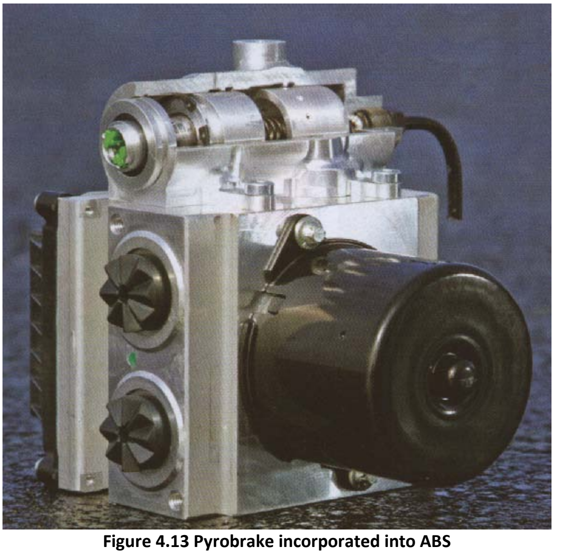

Figure 4.13 Pyrobrake incorporated into ABS

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5 SUSPENSION SYSTEMS

Suspension system connects vehicles body with wheel and its tyre and allow vertical movement of the wheel in relation to body. There are four main functions of the suspension system;

- Forces need to be transferred to the chassis.
- Bump forces resulting from road surface should be reduced – to improve ride comfort
- Reduction in vehicle body vertical movement and rolling
- Damping the natural and forced vibrations of the vehicles body

First all the forces and moments acting on the vehicle during its movement should be discussed.

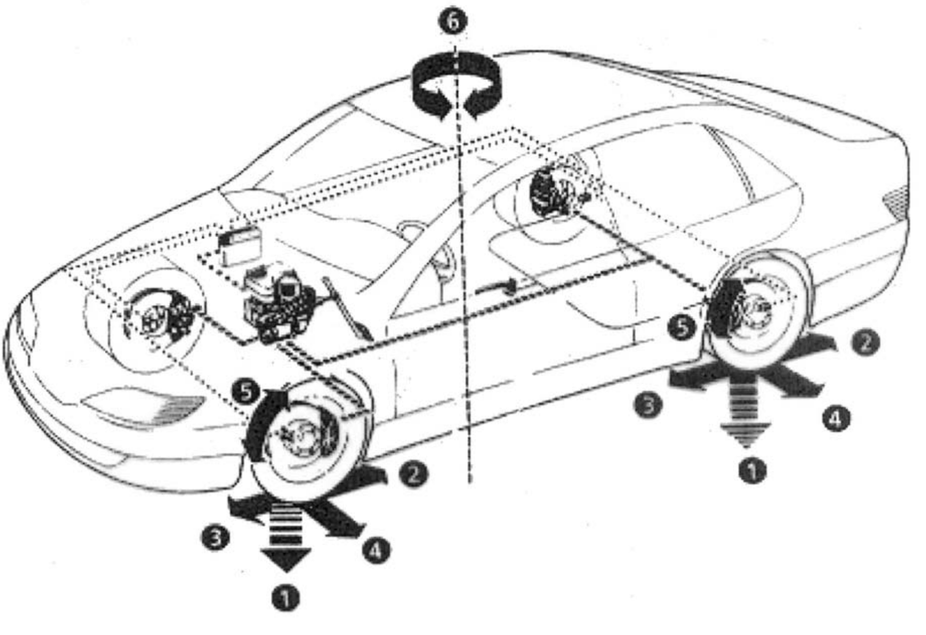


Figure 5.1 Forces and moments acting on vehicle: 1 – wheel load resulting from vehicle weight, 2 – drive forces (resulting from engine power), 3 – brake forces (longitudinal), 4 – lateral forces (present during cornering), 5 – drive and brake moments, 6 – yaw moment

Vehicle elements affecting its dynamics can be divided into several parts (fig. 5.2):

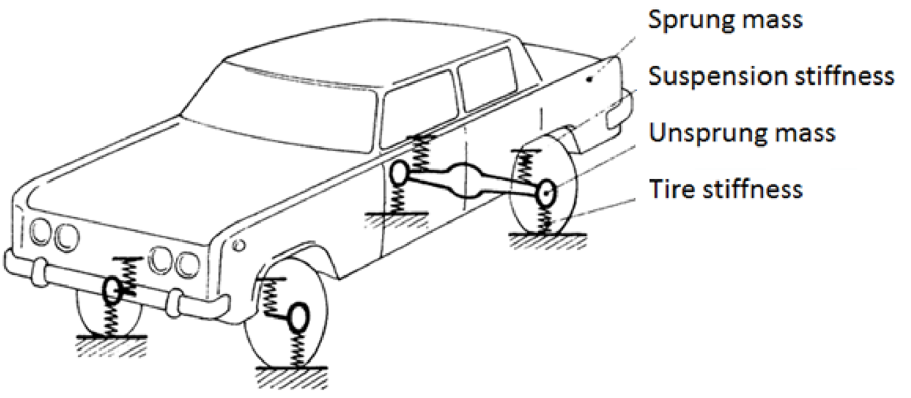


Figure 5.2 Car spring-mass model

The suspension system can be simplified to a spring-mass model (fig. 5.3). Unsprung mass is defined as the weight of all moving elements connected with the wheel – that is the wheel itself, upright, hub, brake mechanism and approx. half the weight of wishbones, tie-rods, spring and damper. The unsprung mass is suspended on the tyre – which has a specified stiffness pressure dependent. Sprung mass is connected to the unsprung mass through a spring with a specified rate. This creates a double spring-mass model – which can be used to calculate elements displacement, velocities and accelerations.

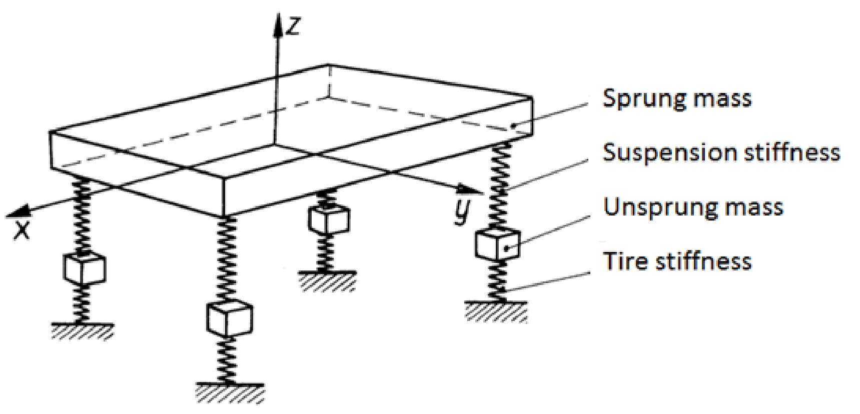


Figure 5.3 Spring-mass model

There are three main types of suspension elements (fig. 5.4)

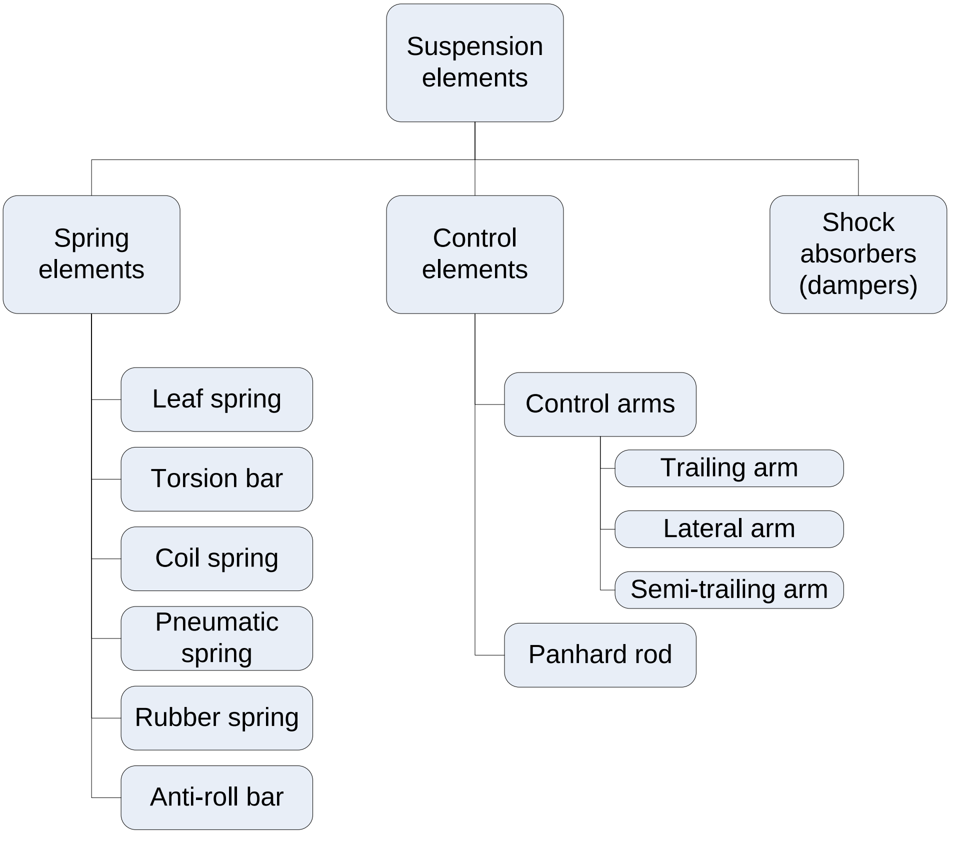


Figure 5.4 Suspension elements division

5.1 Suspension geometry

Each spatial body with six degrees of freedom can be constrained with suitable elements – like rod links – to reduce the number of DoF. A suspension system should provide one degree of freedom for the wheel. This can be done in different ways – for example by adding 5 rod link – each of which would "fix" one degree of freedom. In real suspension systems links constrain wheel carriers – which can be "carry" one or more wheels.

Taking as the reference wheel carriers, suspension systems can be divided into: rigid-axle, independent, compound and tandem wheel. Each solution has numerous layout being used in the industry.

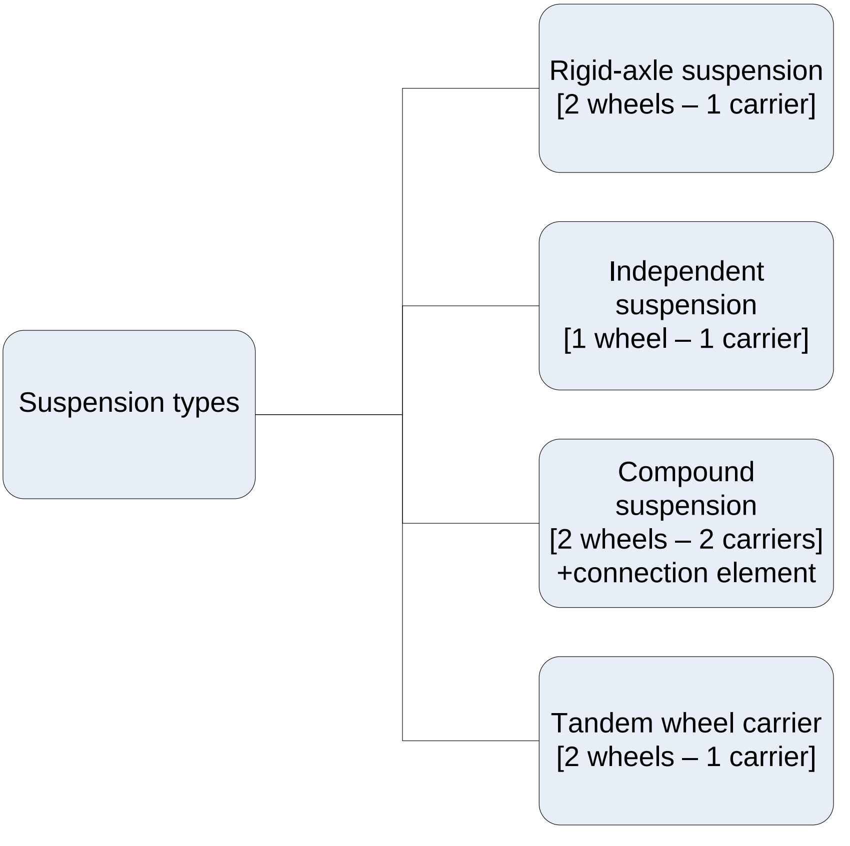


Figure 5.5 Suspension types

5.1.1 Rigid axle

Rigid axle is using single wheel carrier for two wheels. That leads to the point that rigid axle requires two free DoF – as two wheels are part of the mechanism. This solution is rarely used in modern passenger cars – because of its weight, amount of space required and poor dynamic properties. On the other hand this solution can handle high loads and is inexpensive.

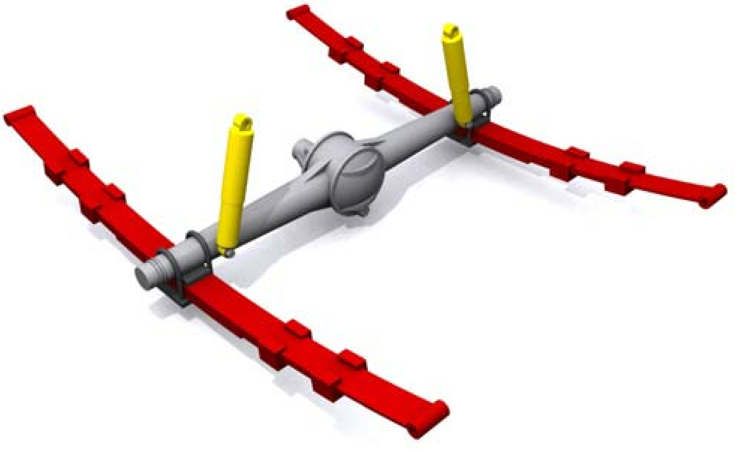


Figure 5.6 Solid axle with leaf spring

Solid axle with leaf springs was widely used for rear suspension of a rear wheel drive cars and trucks. Even nowadays this solution is used in heavy trucks.

Figure 5.7 Four-link solid axle layout

To increase the rigidity of the suspension 4-bar solution was introduced. In comparison to simple leaf spring layout show above, 4-link suspension provide better transferability of lateral forces and more firm suspension response.

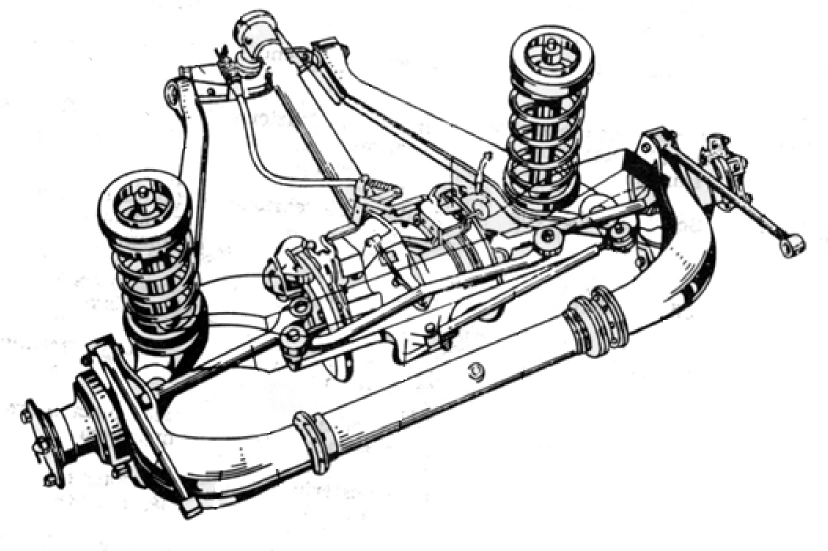


Figure 5.8 DeDion rear suspension layout

DeDion axle is different than most solid axle solutions (fig. 5.8). Here the differential is mounted to the body, therefore unsprung mass is significantly reduced. This solution was first used in sports cars and afterwards propagated to all cars.

5.1.2 Independent suspension

Independent movement of each wheels in an axle was desired in order to improve car handling in comparison to rigid axle solution. Independent suspension has several advantages over rigid axle suspension:

- Reduced unspring mass
- Driving stability increased thanks to better kinematics of the system
- Possibility of incorporating softer springs

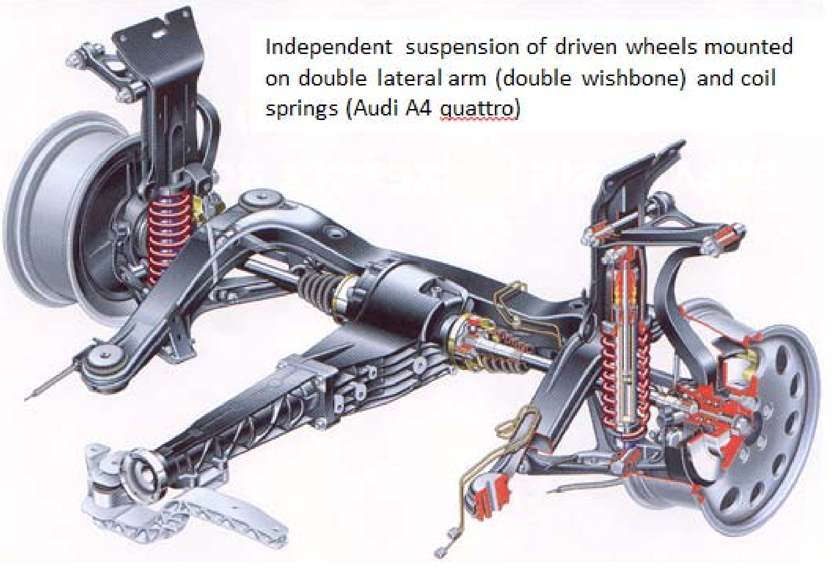


Figure 5.9 Double wishbone Audi rear suspension

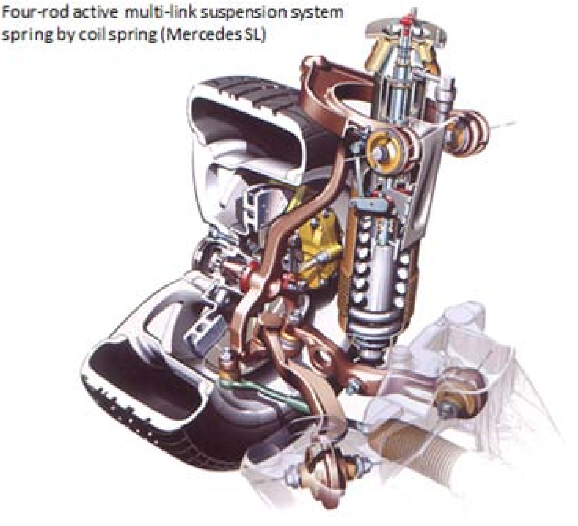


Figure 5.10 Double wishbone Mercedes front suspension

Double wishbones suspension is widely used in high performance cars, to provide good structure rigidity, and vide range of roll centre positions. In most passenger cars the front suspension is using leading column – also called MacPherson strut. With its low space requirements - more space in the engine bay - and inexpensiveness it is the most widely used solution in popular cars.

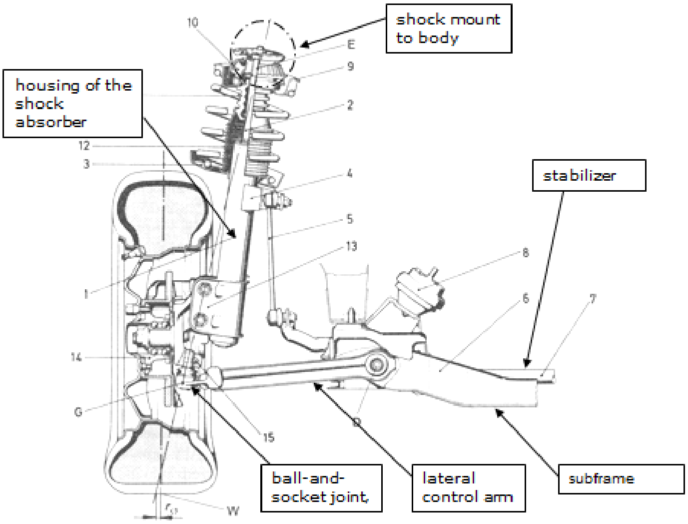


Figure 5.11 MacPherson strut suspension

5.1.3 Compound suspension

Compound or sometimes called semi-independent suspension has common characteristics with both rigid axle and independed suspension. Each wheel has its own wheel carrier and one axle wheels movement is not dependent of each other, though there is a connection between the wheels. These kind of suspensions are widely used in passenger cars because of its inexpensiveness and low space requirements.

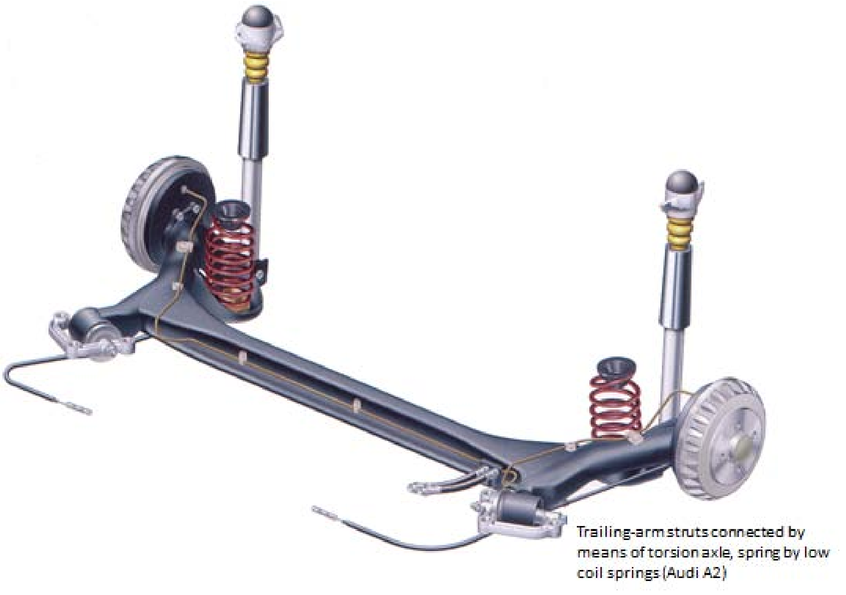


Figure 5.12 Audi compound suspension

5.2 Wheel allignment

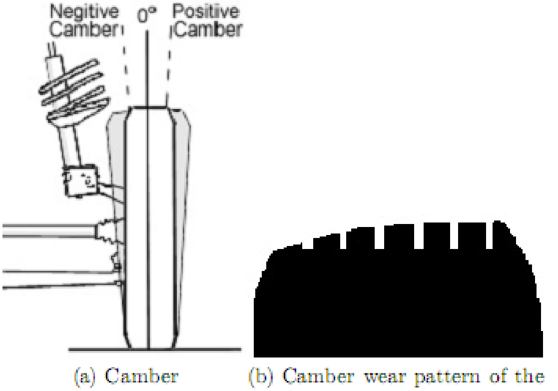
Wheel alignment is the adjusting of the wheels so that they are perpendicular to the ground and parallel to each other, whereas the object of this is to maximise tire life and a vehicle that tracks straight. Wheel alignment is often confused with wheel balancing. The two really have nothing to do with each other except for the fact that they affect ride and handling. If a wheel is out of balance, it will cause a vibration at highway speeds that can be felt in the steering wheel and/or the seat. If the alignment is out, it can cause excessive tire wear and steering or tracking problems. The adjusted angles with their specific names are Camber, Caster and Toe-in, are described In the next few sections.

5.2.1 Camber angle

Camber angle γ is the angle between the tire plane and the vertical plane measured about the x-axis. If the top of the wheel is farther out than the bottom (that is, away from the axle), it is called positive camber; if the bottom of the wheel is farther out than the top, it is called negative camber.

Camber angle alters the handling qualities of a particular suspension design; in particular, negative camber improves grip when cornering. This is because it places the tire at a more optimal angle to the road, transmitting the forces through the vertical plane of the tire, rather than through a shear force across it. Another reason for negative camber is that a rubber tire tends to roll on itself while cornering. If the tire had zero camber, the inside edge of the contact patch would begin to lift off of the ground, thereby reducing the area of the contact patch. By applying negative camber, this effect is reduced, thereby maximizing the contact patch area. Note that this is only true for the outside tire during the turn; the inside tire would benefit most from positive camber.

On the other hand, for maximum straight-line acceleration, the greatest traction will be attained when the camber angle is zero and the tread is flat on the road. Proper management of camber angle is a major factor in suspension design, and must incorporate not only idealized geometric models, but also real-life behaviour of the components.



(a) Camber (b) Camber wear pattern of the tire

Figure 5.13 Camber angle

5.2.2 Caster angle

Caster angle is the angular displacement from the vertical axis of the suspension of a steered wheel in a car, bicycle or other vehicle, measured in the longitudinal direction. It is the angle between the pivot line (in a car - an imaginary line that runs through the centre of the upper ball joint to the centre of the lower ball joint) and vertical. Car racers sometimes adjust caster angle to optimize their car's handling characteristics in particular driving situations.

When an automotive vehicle's front suspension is aligned, caster is adjusted to achieve the self-centring action of steering, which affects the vehicle's straight-line stability. Improper caster settings will cause the driver to move the steering wheel both into and out of each turn, making it difficult to maintain a straight line.

The pivot points of the steering are angled such that a line drawn through them intersects the road surface slightly ahead of the contact point of the wheel. The purpose of this is to provide a degree of self-centring for the steering - the wheel casters around so as to trail behind the axis of steering. This makes a car easier to drive and improves its directional stability (reducing its tendency to wander). Excessive caster angle will make the steering heavier and less responsive, although, in racing, large caster angles are used to improve camber gain in cornering. Caster angles over 10 degrees with radial tires are common. Power steering is usually necessary to overcome the jacking effect from the high caster angle.

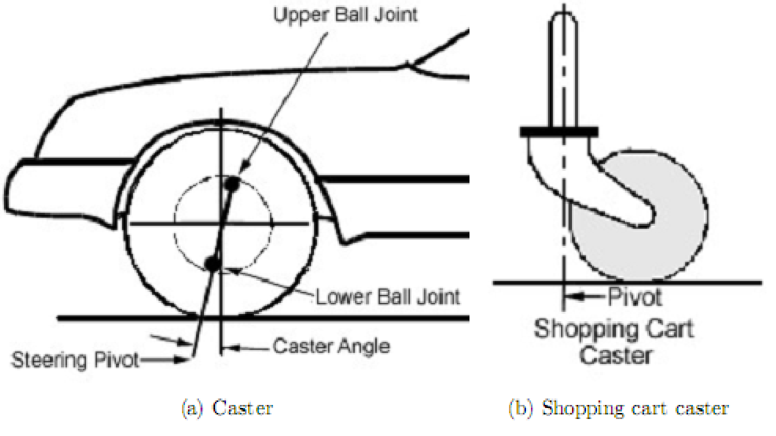


Figure 5.14 Caster angle

5.2.3 KPI inclination

King pin inclination – also called steering inclination – is the angle between the upright rotation axis and vehicles symmetry plane in the front or back view. It has an effect on the wheels inclination to return to straight position. The bigger the KPI angle the faster the wheels will try to return to straight position. On the other hand during turning the wheel on the inside of the corner because of the KPI is reducing camber – which may is some cases – reduce the tyre patch contact with the road.

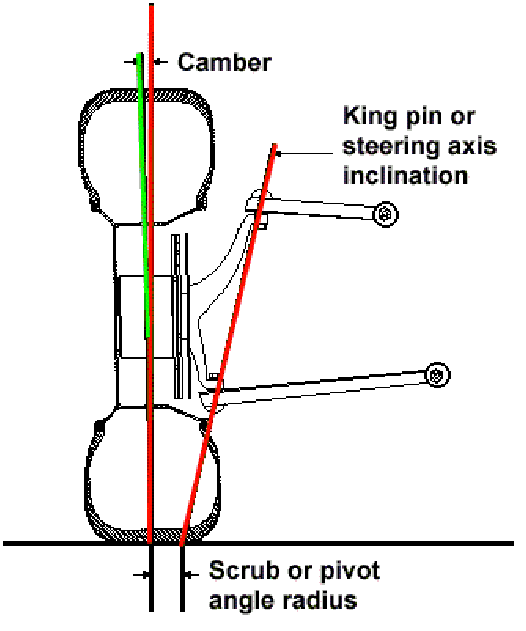


Figure 5.15 KPI inclination

5.2.4 Scrub radius

Scrub radius is the distance between point in which the SAI intersects the ground and the centre of the tyre. This distance must be exactly the same from side to side or the vehicle will pull strongly at all speeds. While included angle problems will affect the scrub radius, it is not the only thing that will affect it. Different wheels or tires from side to side will cause differences in scrub radius as well as a tyre that is low on air. Positive scrub radius is when the tire contact patch is outside of the SAI pivot, while negative scrub radius is when the contact patch is inboard of the SAI pivot (front wheel drive vehicles usually have negative scrub radius).

5.2.5 Toe angle

When a pair of wheels is set so that their leading edges are pointed slightly towards each other, the wheel pair is said to have toe-in. If the leading edges point away from each other, the pair is said to have toe-out. The amount of toe can be expressed in degrees as the angle to which the wheels are out of parallel, or more commonly, as the difference between the track widths as measured at the leading and trailing edges of the tires or wheels. Toe settings affect three major areas of performance: tire wear, straight-line stability and corner entry handling characteristics.

For minimum tire wear and power loss, the wheels on a given axle of a car should point directly ahead when the car is running in a straight line. Excessive toe-in or toe-out causes the tires to scrub, since they are always turned relative to the direction of travel. Too much toe-in causes accelerated wear at the outboard edges of the tires, while too much toe-out causes wear at the inboard edges.

When the wheel on one side of the car encounters a disturbance, that wheel is pulled rearward about its steering axis. This action also pulls the other wheel in the same steering direction. If the wheels have toe-in they absorb the irregularity without significantly changing the direction of the vehicle. In this way, toe-in enhances straight-line stability.

If the car is set up with toe-out, however, the front wheels are aligned so that slight disturbances cause the wheel pair to assume rolling directions that do describe a turn. Any minute steering angle beyond the perfectly centred position will cause the inner wheel to steer in a tighter turn radius than the outer wheel. Thus, the car will always be trying to enter a turn, rather than maintaining a straight line of travel. So it's clear that toe-out encourages the initiation of a turn, while toe-in discourages it.

The toe setting on a particular car becomes a trade-off between the straight-line stability afforded by toe-in and the quick steering response promoted by toe-out. But popular cars are generally set up with toe-in, while race cars are often set up with toe-out as racers are willing to sacrifice a bit of stability on the straightaway for a sharper turn-in to the corners.

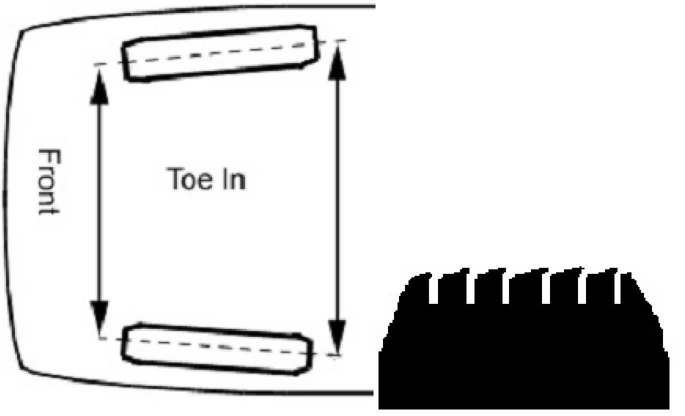


Figure 5.16 Toe angle

5.2.6 Trust angle

Thrust angle is the direction that the rear wheels are pointing in relation to the centre line of the vehicle. If the thrust angle is not zero, then the vehicle will "dogtrack", and the steering wheel will not be centred (fig. 5.17).

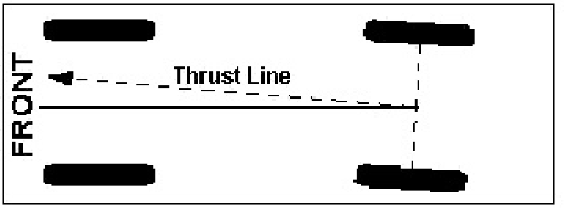


Figure 5.17 Thrust angle

5.3 Roll centre

Roll axis, which is defined by front and rear roll-centre is the axis over which vehicles body is rolling during cornering. Lateral forces applied in the centre of mass act with an leverage equal to the distance between centre of gravity and roll axis. This results in rolling moment which can be noticed as angular displacement of the body in relation to the road surface. Roll centre position is strictly determined by suspension geometry and designation of the RC is conducted differently for various suspension types.

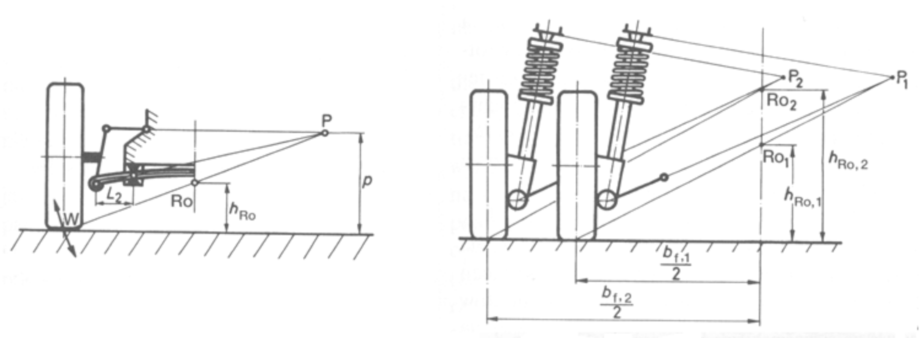


Figure 5.18 Roll centre

5.4 Pitch centre

When longitudinal accelerations are acting on the vehicle, pitching occurs. During acceleration front of the vehicle body is being raised, while the rear – lowered. The opposite is happening when the vehicle brakes. Pitch centre designation procedure depends on the type of suspension used.

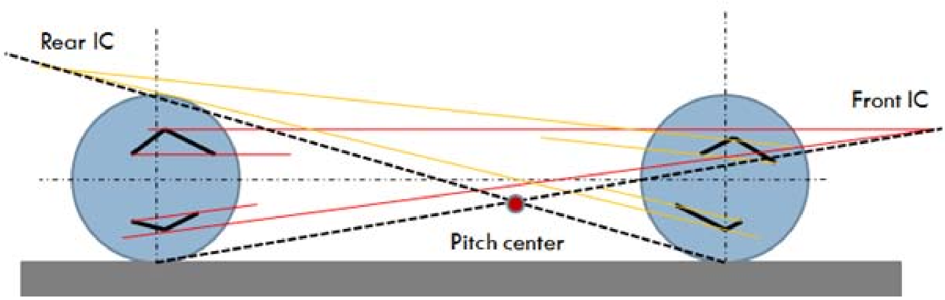


Figure 5.19 Double wishbone suspension pitch centre

5.5 Suspension variables

During wheel vertical movement vehicle parameters like wheelbase, track, or wheel camber angle are changing. The why these variables are being changed depend on the suspension design.

5.5.1 Wheelbase

Wheelbase is the distance between front and rear axle. Large wheelbase ensures more space for passengers and lower tendency for longitudinal leaning of the vehicle. On the other hand smaller wheelbase give better manoeuvrability, Typical wheel base to vehicle length ratio values depending on the vehicle type:

- estate and limousine i_l=0.57 – 0.67
- sedan i_l=0.56 – 0.61
- coupe i_l=0.56

Wheelbase can be dynamically changing when anti-features are present in the suspension design. The wheel is not being moved only vertically. During vertical movement the distance between the axles is rising and decreasing.

5.5.2 Track

Track is defined as the distance between wheel centres of one axle – left and right wheel. During wheel movement track change occur because of the change in wishbone/tie-rods angular position. For 0 degrees of the control arm to the ground the track is at its maximum.

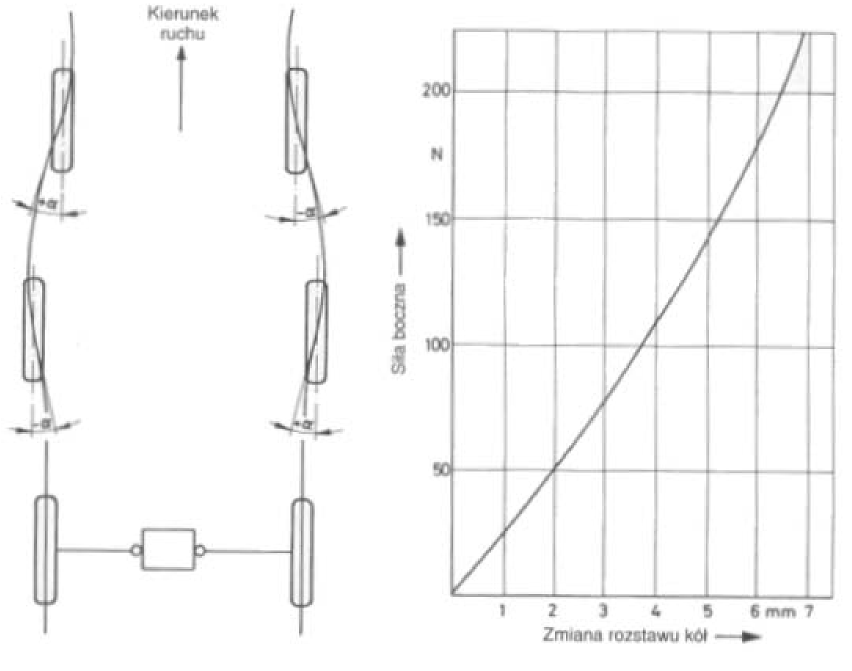


Figure 5.20 Track change in relation to lateral force

5.5.3 Camber angle

Camber change is directly connected with control arm layout. Negative camber increase during upward wheel movement is beneficial for cornering – increased corner speed and vehicle stability. On the other hand during braking because of the camber change – tyre patch is being reduced – therefore braking force is reduced and tyre wear non-uniform.

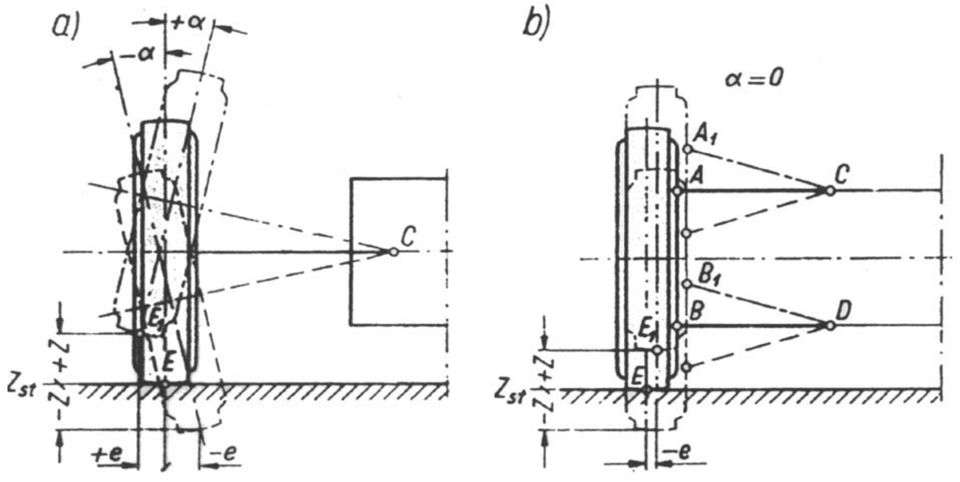


Figure 5.21 Camber angle change

In most sports cars SLA system is used – which stands for Short Long Arm. Thanks to longer lower arm and shorter upper arm – negative camber angle increase can be obtained.

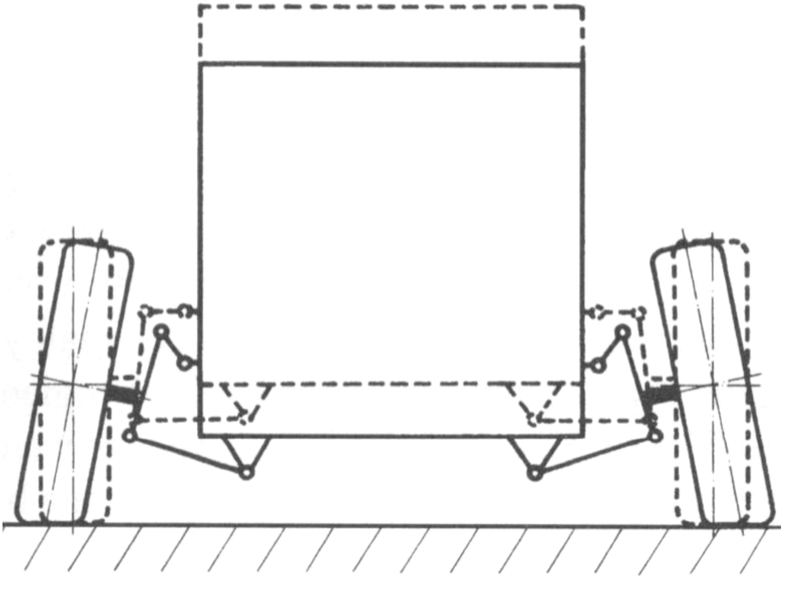


Figure 5.22 Camber gain in SLA layout

5.6 Anti-features

Anti-features can be implemented in the suspension geometry to reduce pitching of the vehicle. Wheel load change caused by weight transfer is acting not only on the spring but also on the „non-deformable" wishbones. Thanks to that front of the car is "diving" less during braking and the rear end squatting less when the car accelerates. One of the main defined geometries are:

- Anti-dive – front suspension geometry, reduces diving during braking
- Anti-squat – rear suspension geometry, reduces squat during accelerating – on rear-wheel drive cars only
- Anti-lift – front suspension geometry, reduces lifting on the front suspension when accelerating
- Anti-lift – in rear suspension geometry, reduces lift (droop) in forward braking.

5.7 Springs and dampers

As it was mentioned before vehicle can be simplified to spring-mass model. Primary purpose of spring and damper is to protect vehicle from impact forces (bump) and high accelerations caused by road irregularities.

$$m\ddot{y} - k(y_t - y_c) - c(\dot{y}_t - \dot{y}_c) = 0$$

$$m_t\ddot{y}_t + k(y_t - y_c) + c(\dot{y}_t - \dot{y}_c) - k_t(y_t - y_c) = 0$$

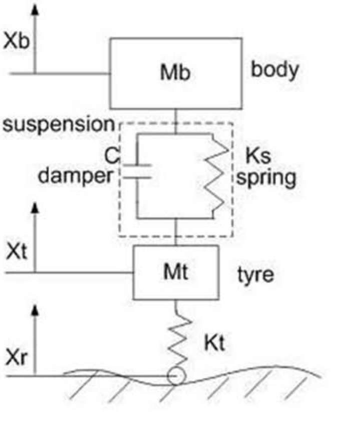


Figure 5.23 Spring-mass model equation on motion

5.7.1 Springs

There are several types of springs used in automotive applications. Leaf springs – the oldest solutions that was used way before the first automobile was built. Nowadays mostly used in heavily loaded vehicles – like trucks, trains, heavy duty off road vehicles. It should be mentioned that Chevrolet Corvette C6 uses leaf springs in rear suspension – although those are specially designed composite leaf springs. Torsion bars are a cheap and very compact solution. Those springs were used in various cars with great success (VW Beetle, Peugeot 205, etc.). Coil springs are the typical springs used in the industry – most of the produced cars use those springs. Thanks to its performance and possibility of incorporating progressive spring rate it is used so widely.

5.7.2 Shock absorbers

Shock absorbers (dampers) role is to damp natural and forced vibrations. It can be divided depending on several factors – from the construction of the damper, through work fluid, and damping coefficient, to pressure present in the damper (fig. 5.24).

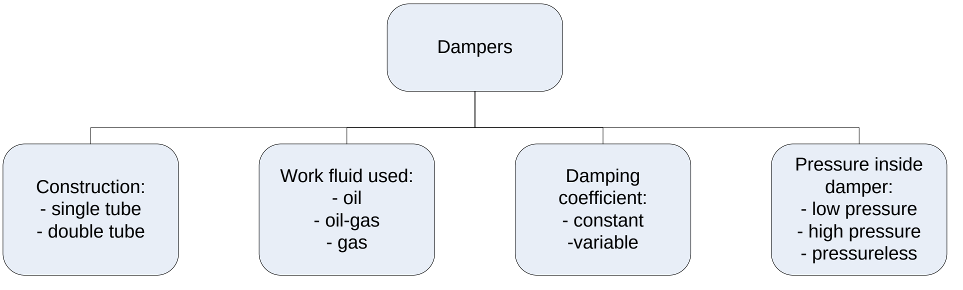


Figure 5.24 Dampers division

Types of damper characteristics can be seen on the figure 5.25:

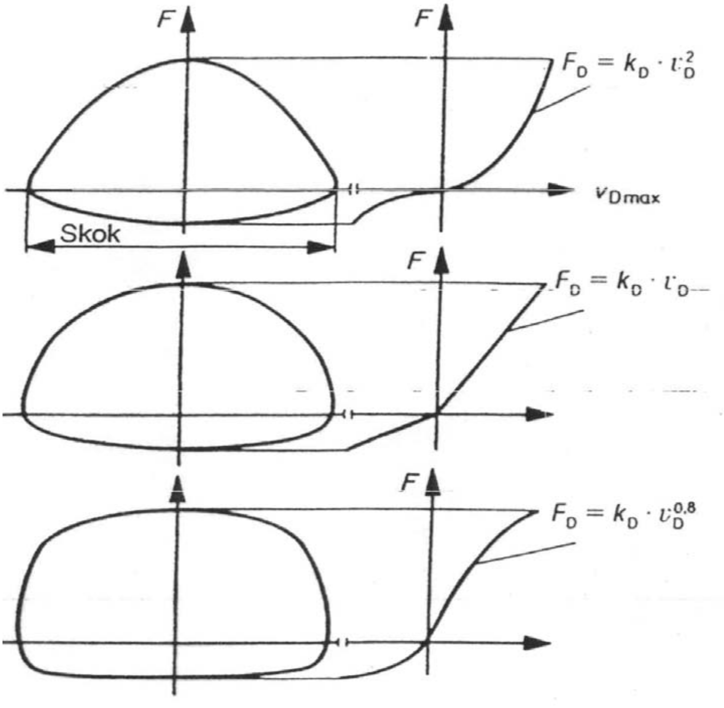


Figure 5.25 Damper characteristics: progressive, linear and depressive

Typical shock absorber consist of:

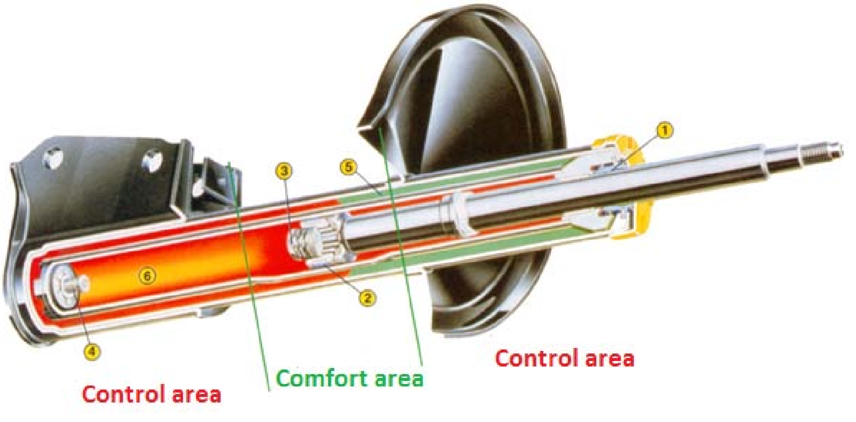


Figure 5.26 Shock absorber components: 1 – low friction seal, 2 – guide bar, 3 – piston with multistage valves, 4 – multistage valve system at the bottom of the damper, 5 – gas at low pressure, 6 – hydraulic fluid

MagneRide™

New system which is being used in sports cars provides variable damping coefficient thanks to special type of fluid used in the shock absorber. Fluid which in response to magnetic field is changing its flow properties, together with electronic system monitoring wheels movement, is adjusting damping coefficient dynamically. (Fig. 5.27)

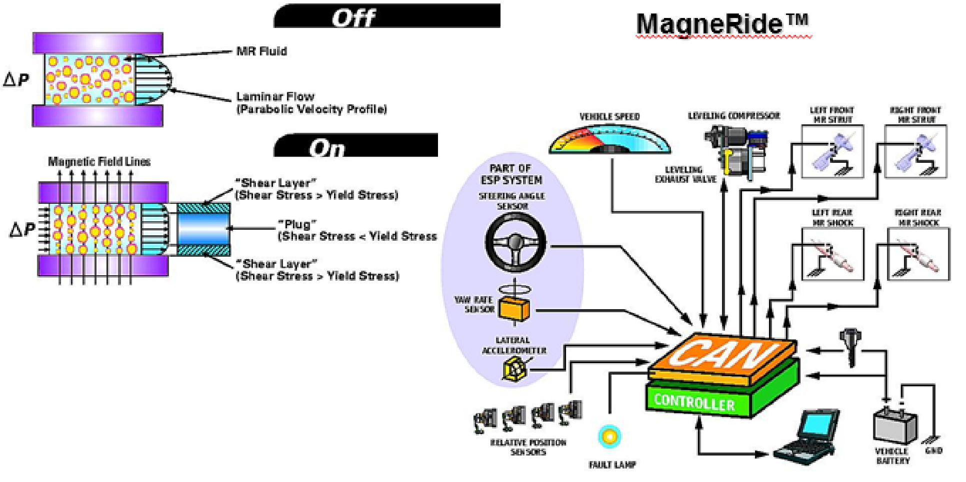


Figure 5.27 Magna ride working layout and principle

5.8 Anti-roll bar

To prevent vehicle rolling during cornering an anti-roll bar (sway bar) was designed. Levers of the sway bar are connected to wheel carrier or wishbone and transfer force to the bar which is being twisted. Depending on the stiffness of the bar – roll rate of the vehicle can be affected.

Anti-roll bar design procedure is the following:

- Define max angular roll of the vehicle during defined cornering (g's, friction coef. of the road, etc),
- Find the motion ration – wheel-lever arm
- Calculate required diameter of the bar.

$$M_{\varphi} = \frac{\pi d^4 G b_2^2 \beta}{64 b_1 r_w^2} \tag{5.1}$$

$$d = \sqrt[4]{\frac{64 \zeta M_b b_1 r_w^2}{\pi G b_2^2 \beta}} \tag{5.2}$$

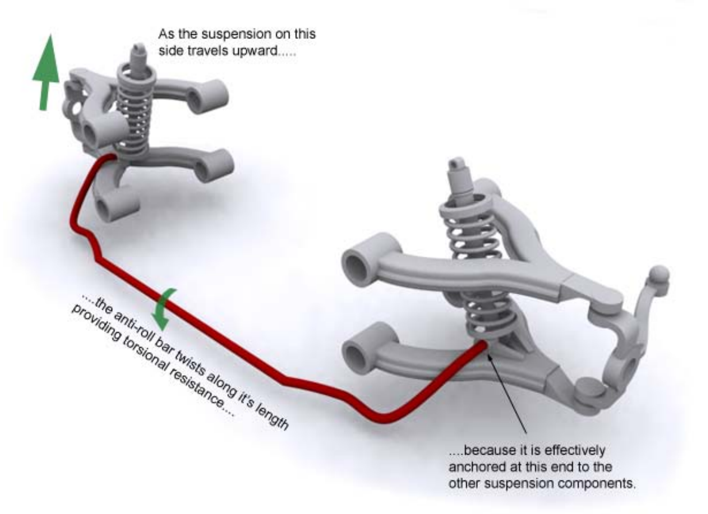


Figure 5.28 Anti-roll bar

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6 OVERALL ROAD LOAD IMPACT OF FUEL ECONOMY

The automotive industry can be classified as one of the most fuel consuming sector. Therefore fuel economy is becoming the aspect of great importance. Only 15% of fuel is efficiently used for car motion or supplying additional accessories (e.g. air conditioning). Rest of energy produced by fuel combustion is wasted in power drive inefficiencies and idling [9]. Unfortunately, it is impossible to entirely predict the fuel consumption of a given vehicle due to the fact that it is dependent on various factors e.g. driving style, climate, traffic patterns (rush hours vs. midday or weekends), road loads, weather, vehicle maintenance. Even such accessories like air conditioning, radio, GPS requires power supply and in consequence addition part of fuel has to be consumed.

Another factor which significantly affects vehicle's fuel economy is the ambient temperature during the year season. It appears that lower ambient temperature have negative influence on fuel economy. Another words larger amount of gasoline will be consumed during winter driving comparing with another seasons. It is because greater friction between vehicle's parts, greater rolling resistance caused by poor road conditions, greater power load on the engine and longer time of cold engine operating mode as well as longer time spend on idling. The fuel is usually produced with different composition, depending on season. The winter gasoline mixture is less dense and with lower molecular weight hydrocarbons in order to ensure greater volatility for engine operating in lower winter temperature. A volatility defines the evaporation properties of a fluid. The higher volatility the more intensive evaporation of the fluid will occur. The incorrect volatility of a gasoline can be reason of engine problems. Due to low volatility the insufficient mixing of fuel and air may occur, as well as insufficient fuel reaching the combustion chamber. While, to high volatility can cause the warm engine operating problems, because the fuel can turn into gas before reaching the carburetor or injectors and consequently intruding the flow. This is called "vapor lock". The less dense winter gasoline however, can produce less energy and therefore negatively affect the fuel economy.

Apart from the season condition the fuel economy is different for urban driving and highway driving. In case of urban driving, the braking is main reason of fuel consumption as it consumes the momentum of the vehicle which than has to be replenished by acceleration. Moreover, very frequent during city driving, idling also causes increase fuel consumption. On the other hand, the aerodynamic drag is predominant factor in terms of fuel consumption during highway driving

[10]. Both of the driving types are used in order to determine the fuel consumption in engine test house. The variability of velocity are described by ECE 15 European standard [9].

Generally, fuel consumption is dependent on power demanded to overcome every forces resisting motion of the vehicle (i.e. inertia, road grade, tire rolling resistance and aerodynamic loss). The rolling resistance however, consumes small part of energy output in consequence consuming only 4 – 7 % of total energy expanded by the vehicle [13]. The schematic model of the fuel consumption is shown in *figure 6.1* [11].

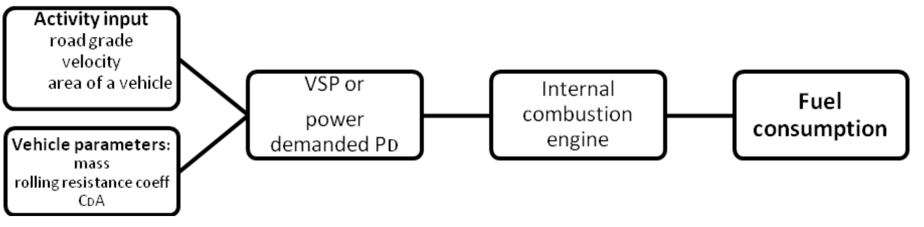

Figure 6.1 The schematic model of a fuel consumption [11]

Understandably, the fuel consumption depends upon the velocity, whereas the velocity is dependent upon engine's rotational speed. The rotational speed is controlled by means of gear. The faster vehicle drives without changing the gear the more fuel will be consumed. Knowing the gear ratio it is possible to calculate the velocity on each gear using the *equation 6.1*.

$$v_i = \frac{2\pi r_w n}{i_i} \tag{6.1}$$

Where:
r_w is the radius of the drive wheel.
n is the rotational speed of the engine.
i_i is the gear ratio.

The demanded power can therefore, be expressed as a product of all resisting forces (F_R) and the velocity of the car (v_i).

$$P_D = F_R v_i, W \tag{6.2}$$

$$F_R = Qf\cos\alpha + Q\sin\alpha + \frac{\varrho v_i^2}{2} c_D A \tag{6.3}$$

The fuel rate (*FR*) is defined as the amount of fuel required to generate one kilowatt-hour of energy. This value is usually determined with aid of the characteristic of the particular engine. Knowing the value of percentage shear of maximum available power that the engine produces operating at given rotational speed it is possible to determine the FR. The approximate value of amount of fuel consumed within the distance of 100 km is given by *equation 6.4*

$$q = \frac{FR \cdot F_R}{3600 \eta_{PT} \rho_f}, \ {}^{L}/_{100km} \tag{6.4}$$

Where:

η_{PT} is the efficiency of the power transmission system

It is obvious that the lower drag the better fuel economy (less fuel consumed). Sovran & Bohn has proposed a relationship between change of drag coefficient and the change of fuel savings during the same driving cycle. This dependence is shown in *equations 6.5* [9].

$$\%fuel\ saving = \zeta \cdot \%\ drag\ reduction$$

$$\zeta = \frac{0{,}89}{1 + \frac{m(0{,}081 f_R + 0{,}000126)}{c_D A}} \tag{6.5}$$

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7 CAD/CAM/CAE SOFTWARE IN VEHICLE ENGINEERING

7.1 Usage of computers in automotive engineering

Computers with proper software may be used in automotive engineering as as:

- **monitoring and control of processes** – for example in continuous process industrie (like Fiat factory in Tychy) a number of processes parameters must be monitored and controlled. Additionally its verification is also possible.

- **design and manufacturing operations by supporting applications –** computers provide successful completion of manufacturing operations

7.2 Design and manufacturing application

- CAD
  - Computer Aided Design
- CAE
  - Computer Aided Engineering
- CAM
  - Computer Aided Manufacturing
- CAPP
  - Computer Aided Process Planing
    - link between CAD and CAM.
- CATD
  - Computer Aided Tool Design
    - used for developing tools for manufacture
- CAP
  - Computer Aidded Planning
    - used for planning functions

7.3 Computer Aided Design

Computer-aided design (CAD), also known as computer-aided design and drafting (CADD), is the use of computer technology for the process of design and design-documentation. Computer Aided Drafting describes the process of drafting with a computer. CADD software, or environments, provides the user with input-tools for the purpose of streamlining design processes; drafting, documentation, and manufacturing processes. CADD output is often in the form of electronic files for print or machining operations. The development of CADD-based software is in direct correlation with the processes it seeks to economize; industry-based software (construction, manufacturing, etc.) typically uses vector-based (linear) environments whereas graphic-based software utilizes raster-based (pixelated) environments.

CADD environments often involve more than just shapes. As in the manual drafting of technical and engineering drawings, the output of CAD must convey information, such as materials, processes, dimensions, and tolerances, according to application-specific conventions.

CAD may be used to design curves and figures in two-dimensional (2D) space; or curves, surfaces, and solids in three-dimensional (3D) objects

CAD is an important industrial art extensively used in many applications, including automotive, shipbuilding, and aerospace industries, industrial and architectural design, prosthetics, and many more. CAD is also widely used to produce computer animation for special effects in movies, advertising and technical manuals. The modern ubiquity and power of computers means that even perfume bottles and shampoo dispensers are designed using techniques unheard of by engineers of the 1960s. Because of its enormous economic importance, CAD has been a major driving force for research in computational geometry, computer graphics (both hardware and software), and discrete differential geometry.

Figure 7.1 CAD in suspension design

Current computer-aided design software packages range from 2D vector-based drafting systems to 3D solid and surface modellers. Modern CAD packages can also frequently allow rotations in three dimensions, allowing viewing of a designed object from any desired angle, even from the inside looking out.

There are several different types of CAD. Each of these different types of CAD systems require the operator to think differently about how he or she will use them and he or she must design their virtual components in a different manner for each.

There are many producers of the lower-end 2D systems, including a number of free and open source programs. These provide an approach to the drawing process without all the fuss over scale and placement on the drawing sheet that accompanied hand drafting, since these can be adjusted as required during the creation of the final draft.

- 3D wireframe is basically an extension of 2D drafting. Each line has to be manually inserted into the drawing. The final product has no mass properties associated with it and cannot have features directly added to it, such as holes. The operator approaches these in a similar fashion to the 2D systems, although many 3D systems allow using the wireframe model to make the final engineering drawing views.
- 3D "dumb" solids (programs incorporating this technology include AutoCAD) are created in a way analogous to manipulations of real world objects. Basic three-dimensional geometric forms (prisms, cylinders, spheres, and so on) have solid volumes added or subtracted from them, as if assembling or cutting real-world objects. Two-dimensional projected views can easily be generated from the models. Basic 3D solids don't usually

include tools to easily allow motion of components, set limits to their motion, or identify interference between components.

- 3D parametric solid modeling requires the operator to use what is referred to as "design intent". The objects and features created are adjustable. Any future modifications will be simple, difficult, or nearly impossible, depending on how the original part was created. One must think of this as being a "perfect world" representation of the component. If a feature was intended to be located from the center of the part, the operator needs to locate it from the center of the model, not, perhaps, from a more convenient edge or an arbitrary point, as he could when using "dumb" solids. Parametric solids require the operator to consider the consequences of his actions carefully.

7.4 Computer Aided Engineering

Software tools that have been developed to support these activities are considered CAE tools. CAE tools are being used, for example, to analyze the robustness and performance of components and assemblies. The term encompasses simulation, validation, and optimization of products and manufacturing tools. In the future, CAE systems will be major providers of information to help support design teams in decision making.

In regard to information networks, CAE systems are individually considered a single node on a total information network and each node may interact with other nodes on the network.

CAE systems can provide support to businesses. This is achieved by the use of reference architectures and their ability to place information views on the business process. Reference architecture is the basis from which information model, especially product and manufacturing models.

7.4.1 Branches of CAE

CAE

- FEA
- CFD
- MES
- Optimisation

7.4.2 CAE phases

In general, there are three phases in any computer-aided engineering task:

- Pre-processing – defining the model and environmental factors to be applied to it. (typically a finite element model, but facet, voxel and thin sheet methods are also used)
- Analysis solver (usually performed on high powered computers)
- Post-processing of results (using visualization tools)

This cycle is iterated, often many times, either manually or with the use of commercial optimization software.

7.4.3 CAE in the automotive industry

CAE tools are very widely used in the automotive industry. In fact, their use has enabled the automakers to reduce product development cost and time while improving the safety, comfort, and durability of the vehicles they produce. The predictive capability of CAE tools has progressed to the point where much of the design verification is now done using computer simulations rather than physical prototype testing. CAE dependability is based upon all proper assumptions as inputs and must identify critical inputs (BJ). Even though there have been many advances in CAE and it is widely used in the engineering field. Physical testing is still used as a final confirmation for subsystems due to the fact that CAE cannot predict all variables in complex assemblies (i.e. metal stretch, thinning).

7.5 Finite Element Method Analysis

The finite element method (FEM) (its practical application often known as finite element analysis (FEA)) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as of integral equations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then numerically integrated using standard techniques such as Euler's method, Runge-Kutta, etc.

In solving partial differential equations, the primary challenge is to create an equation that approximates the equation to be studied, but is numerically stable, meaning that errors in the input and intermediate calculations do not accumulate and cause the resulting output to be meaningless. There are many ways of doing this, all with advantages and disadvantages. The Finite Element Method is a good choice for solving partial differential equations over complicated domains (like cars and oil pipelines), when the domain changes (as during a solid state reaction with a moving boundary), when the desired precision varies over the entire domain, or when the solution lacks smoothness.

This powerful design tool has significantly improved both the standard of engineering designs and the methodology of the design process in many industrial applications. The introduction of FEM has substantially decreased the time to take products from concept to the production line. It is primarily through improved initial prototype designs using FEM that testing and development have been accelerated. In summary, benefits of FEM include increased accuracy, enhanced design and better insight into critical design parameters, virtual prototyping, fewer hardware prototypes, a faster and less expensive design cycle, increased productivity, and increased revenue.

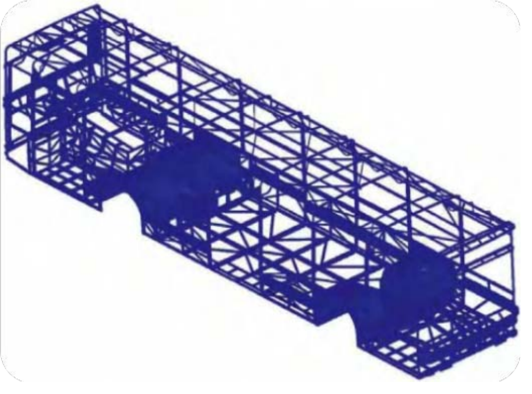


Figure 7.2 Bus created as a wireframe with beam elements prepared for analysis

7.5.1 FEM Steps

1. Geometrical model
2. Discrete model
3. Boundary conditions
4. Solution
5. Post-processing

7.5.2 FEM – Geometrical model

Simplifications of:

- geometry, shape, complexity

Material

- choose of its properties and its properties, usually assumption of linearity

7.5.3 FEM – Discrete model

Discretization:

- process of transferring continuous models into discrete parts. This process is usually carried out as the first step towards making them suitable for numerical calculations.

Assumption:

- Finite numbers of points are representative for the structure made of infinite number of points.

Reason

- Making number variables smaller and possible to calculate

7.5.4 FEM – Boundary conditions

Restrains

- Fixing of a model

Loads

- Forces
- Bending / torsional moments
- Continuous loads
- Pressure
- Temperature
- Displacements

7.5.5 FEM - Solution

Assumption

- Solution approach is based either on eliminating the differential equation completely or rendering the partial differential equations into an approximating system of ordinary differential equations, which are then numerically integrated.

Techniques

- Euler's method
- Runge-Kutta method

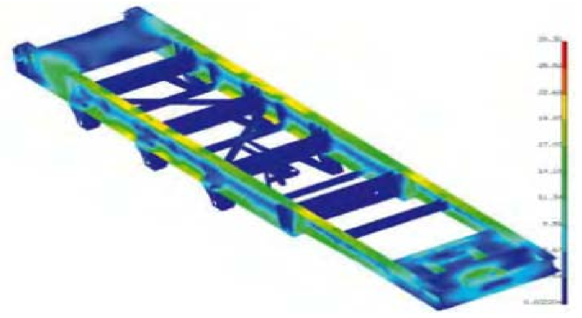


Figure 7.3 Bus frame strength calculation

7.5.6 FEM – Troubles

Geometrical model

- must be made in specific way

Discrete model

- proper discretization, choosing of proper finite element, mesh size

Boundary conditions

- real forces and fixtures existing in the model must be properly defined in software

Solution

- if everything before was OK. Solution should also be OK

Post-processing

- results interpretation is difficult for beginners

7.5.7 FEM – 1D Finite Element

Degrees of freedom

- 6 DOF
    - beam element
- 3 DOF
    - truss

Associated geometry

- line
- curve

Additional properties

- cross-section
- orientation of cross-section

7.5.8 FEM – 2D Finite Element

Degrees of freedom

- 6 DOF

Associated geometry

- surface model

Additional properties

- thickness

7.5.9 FEM – 3D Finite Element

Degrees of freedom

- 3 DOF

Associated geometry

- solid model

Additional properties

- model with fully defined geometry

7.5.10 FEM – Automotive engineering

- FEM
- Static analysis
    - Strength calculations
- Dynamic analysis
    - Natural frequencies
    - Crash-tests

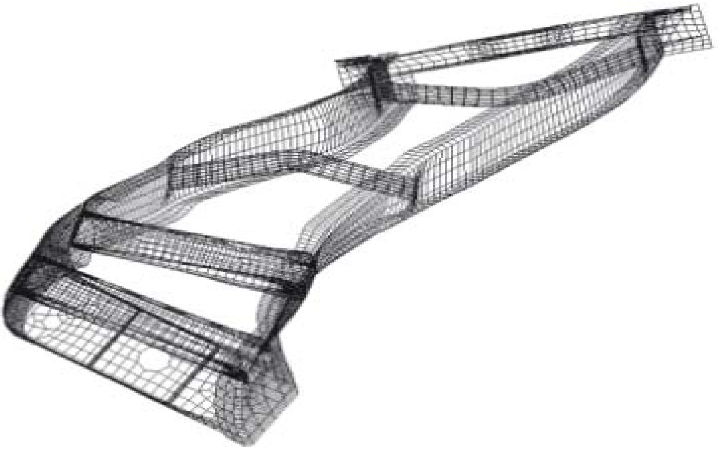


Figure 7.4 1st mode shape of a trailer frame

7.5.11 FEM – Software

Solvers

- ABAQUS
- NASTRAN
- PATRAN
- ANSYS
- MARC

All in one

- I-DEAS
- PRO/ENGINEER
- CATIA
- UNIGRAPHIX 9NX0

Pre/post processors

- HYPERMESH

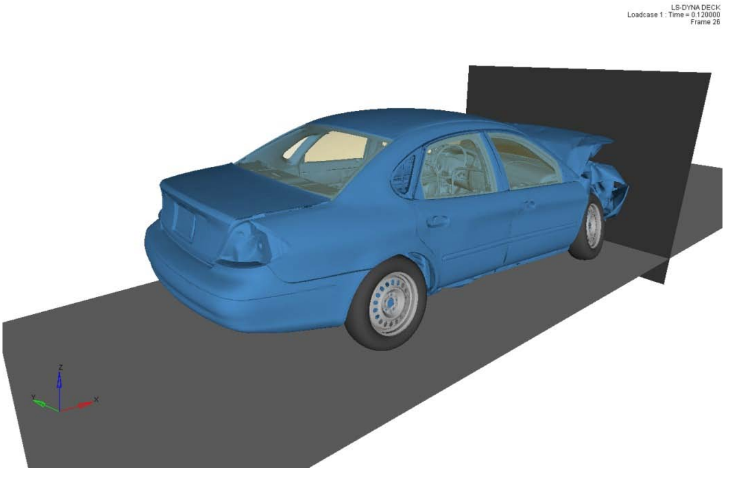


Figure 7.5 Crash in LS-DYNA - Ansys module

7.6 Computational Fluid Dynamics

Branch of fluid mechanics that uses numerical methods and algorithms to solve and analyse problems that involve fluid flows. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions. With high-speed supercomputers, better solutions can be achieved. On-going research, however, yield software that improves the accuracy and speed of complex simulation scenarios such as transonic or turbulent flows.

7.6.1 CFD Steps

In all of these approaches the same basic procedure is followed.

During pre-processing

- The geometry (physical bounds) of the problem is defined.
- The volume occupied by the fluid is divided into discrete cells (the mesh). The mesh may be uniform or non-uniform.
- The physical modelling is defined – for example, the equations of motions + enthalpy + radiation + species conservation
- Boundary conditions are defined. This involves specifying the fluid behaviour and properties at the boundaries of the problem. For transient problems, the initial conditions are also defined.

The simulation is started and the equations are solved iteratively as a steady-state or transient.

Finally a postprocessor is used for the analysis and visualization of the resulting solution.

7.6.2 CFD - Calculation

DNS

Direct numerical simulation (DNS) resolves the entire range of turbulent length scales. This marginalizes the effect of models, but is extremely expensive. The computational cost is proportional to Re3.DNS is intractable for flows with complex geometries or flow configurations.

LES

Large eddy simulation (LES) is a technique in which the smallest scales of the flow are removed through a filtering operation, and their effect modelled using sub-grid scale models. This allows the largest and most important scales of the turbulence to be resolved, while greatly reducing the computational cost incurred by the smallest scales. This method requires greater computational resources than RANS methods, but is far cheaper than DNS.

DES

Detached eddy simulations (DES) is a modification of a RANS model in which the model switches to a sub-grid scale formulation in regions fine enough for LES calculations. Regions near solid boundaries and where the turbulent length scale is less than the maximum grid dimension are assigned the RANS mode of solution. As the turbulent length scale exceeds the grid dimension, the regions are solved using the LES mode. Therefore the grid resolution for DES is not as demanding as pure LES, thereby considerably cutting down the cost of the computation. Though DES was initially formulated for the Spalart-Allmaras model (Spalart et al., 1997), it can be implemented with other RANS models (Strelets, 2001), by appropriately modifying the length scale which is explicitly or implicitly involved in the RANS model. So while Spalart-Allmaras model based DES acts as LES with a wall model, DES based on other models (like two equation models) behave as a hybrid RANS-LES model. Grid generation is more complicated than for a simple RANS or LES case due to the RANS-LES switch. DES is a non-zonal approach and provides a single smooth velocity field across the RANS and the LES regions of the solutions.

7.7 Optimisation

Uses optimization methods to solve design problems incorporating a number of disciplines. It is also known as multidisciplinary optimization and multidisciplinary system design optimization (MSDO).

MDO allows designers to incorporate all relevant disciplines simultaneously. The optimum of the simultaneous problem is superior to the design found by optimizing each discipline sequentially, since it can exploit the interactions between the disciplines. However, including all disciplines simultaneously significantly increases the complexity of the problem.

These techniques have been used in a number of fields, including automobile design, naval architecture, electronics, computers, and electricity distribution. However, the largest number of applications has been in the field of aerospace engineering, such as aircraft and spacecraft design. For example, the proposed Boeing blended wing body (BWB) aircraft concept has used MDO extensively in the conceptual and preliminary design stages. The disciplines considered in the BWB design are aerodynamics, structural analysis, propulsion, control theory, and economics.

7.8 Computer aided manufacturing

Computer-aided manufacturing (CAM) is the use of computer software to control machine tools and related machinery in the manufacturing of work-piece. This is not the only definition for CAM, but it is the most common. CAM may also refer to the use of a computer to assist in all operations of a manufacturing plant, including planning, management, transportation and storage. Its primary purpose is to create a faster production process and components and tooling with more precise dimensions and material consistency, which in some cases, uses only the required amount of raw material (thus minimizing waste), while simultaneously reducing energy consumption.

CAM is a subsequent computer-aided process after computer-aided design (CAD) and sometimes computer-aided engineering (CAE), as the model generated in CAD and verified in CAE can be input into CAM software, which then controls the machine tool.

7.8.1 CAM – Areas of concern

- High Speed Machining, including streamlining of tool paths
- Multi-function Machining
- 5 Axis Machining
- Feature recognition and machining
- Automation of Machining processes
- Ease of Use

7.8.2 CAM – Automotive Engineering

Machining of parts used in vehicles.

- Engine parts.
- Suspension Parts.

Machining of dies for cast of parts used in vehicles.

- Dies for all kind of metal casted parts.
- Dies for plastic parts.
- Dies for sheet metal forming.
- Dies for forging.

7.9 Computer aided car body optimization

Designing a new vehicle is an extremely challenging and a complex process as it requires close coordination and inputs from a number of disciplines in developing a number of systems and sub-systems in the vehicle. Further, the numerous systems in a vehicle must fit within the specified vehicle space, function and provide the customers an acceptable combination of all relevant vehicle attributes (e.g. appearance, performance, safety, ride, comfort, etc.).

The development of a car body is a very complex process, as various functional requirements have to be considered. Different targets with sometimes competitive issues define a challenge for the development engineers. Today, numerical simulation is a well-established and widespread tool to predict, analyse and optimize the functional characteristics of a car. The engineer describes a design and the simulation will predict the performance of this design. Numerical optimization is performed considering different simulation discipline. Once an optimal design has been found it has to be verified for all other relevant disciplines. In most cases this process leads to significant design changes. Therefore, it requires a great effort of all participating engineers to find an optimum design that satisfies all relevant disciplines. This process is very time consuming and often ends up in an unsatisfying design compromise. Therefore, it is very necessary to get the best design parameters in the conceptual design phase by optimization which would avoid complicated modification in anaphase design.

7.9.1 VCD system

VCD system is a new methodology for auto-body structure concept design. It is based on template method, knowledge-based engineering (KBE), and parametrical method. This technology builds up body parametrical conceptual model and FE mesh model quickly and easily by VCD template model. Since the present VCD system has been integrated into the UGS platform, UGS/NX Nastran can be directly made use of solving the strength, stiffness and modal solution within the VCD user interface. The post-process view and product design report could be rapidly generated to afford designer needed data and information. It can optimize the beam and panel's thickness and beam's section shapes in order to make these auto-body.

Parametric body modelling

VCD system is fully parameterized. The full parameterization technology means that the change of parameter's values in design process could update other module of VCD system. If user changes the parameter after the optimization process, the changes can update the body's geometry mode automatically to keep the consistency of the model. What is more geometry mesh and optimization model is parametric.

Design with body template

The formation of template comes from object-oriented technology. Its basic idea is to abstract a background of model. A car body part is the object in VCD system. The template can produce new instance with analogous attributes. Because there are much analogous attributes among car-body, the conception of template is afforded to build a new auto-body model. Many types of auto-body templates are afforded in the VCD system, such as car, bus and truck. The vehicle template is built by parametric design method. So its shape is changeable. User need not build auto-body geometry model from point or line, which would save much time.

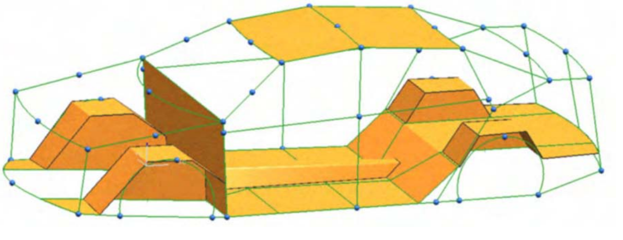


Figure 7.6 VCD template part

There are many parameters defined in the template object, such as control point's position and auto-body's dimensions. There are also some dimension constraints defined to control whole sizes and standards of the auto-body. These dimensions are defined in the template according to SAE standard. These parameter values are recorded in spread sheet linked with the template part.

NX integration

The VCD system is built on NX platform (UGS). It integrates NX Nastran and Hyper Opt solver needed in structure and optimization calculations. All of the parameters and results in VCD processes will be updated into VCD parameter database.

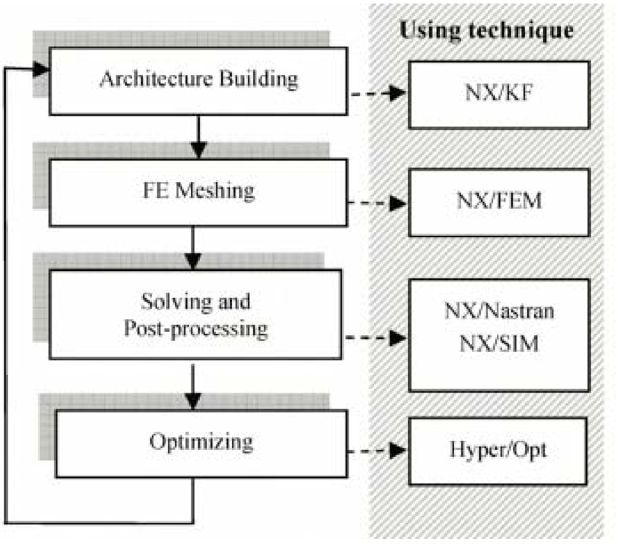


Figure 7.7 VCD system program flowchart

7.9.2 Development of optimization volume

VCD system can implement body structural parameter optimization. Its' optimization module can be used to optimize the solution result, which is built in FE Meshing procedure. User can define auto-body weight, stiffness and frequency as the optimized objects to optimize sheet thickness of auto-body structure and section shape of beam. Multi-constrains is supported in the system. Weight, stiffness, stress and frequency could be selected as constrains. The optimization result can be used to update auto-body' parametric model. In optimization process, the system allows user to define design variable link to reduce variable number and show optimization iterative process by spread sheet. Figure below shows the procedure of the VCD optimization module developed in this study.

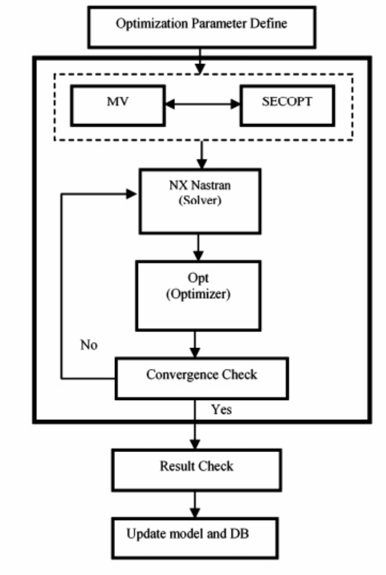


Figure 7.8 Procedure of the VCD optimization module

As shown in diagram, NX/NASTRAN is used to perform the structure analysis of B.I.W. is the module to modify model by response variables. SECOPT is used to calculate the section properties of the pillars. After optimization, user can check result and update design model and parameter database (DB) automatically.

7.9.3 Example of application

VCD system has been used in R&D centre of FAW Group of china.

FAW is a maker of automobiles, buses; light, medium, and heavy-duty trucks; and auto parts, FAW became China's first automobile manufacturer when it unveiled the nation's first domestically-produced passenger car, the Hong Qi, in 1958. Selling 2.07 million units in 2010 made it the third-most productive vehicle manufacturer in China that year. FAW has its headquarters in Changchun, a tree-lined city on the Northeastern Songliao Plain.

VCD system can make concept structure design easier for new type body development.

Stiffness and modal analysis

The detail model and VCD model of the car body were calculated according at the same solution. The results of calculations are show in the table below.

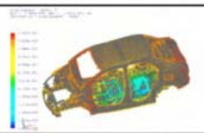
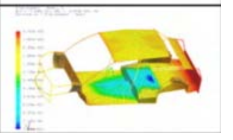
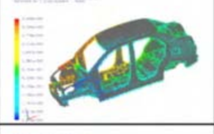
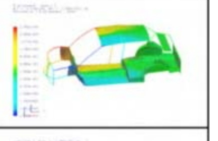
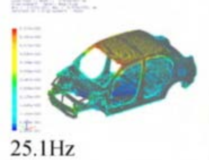
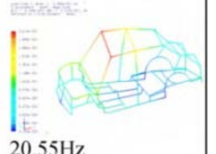
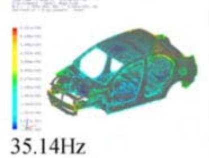
	Detail model	VCD model
Bend maximum displacement	0.24mm	0.214mm
Bend stiffness	26778 N/mm	29906 N/mm
Bend displacement figure		
Torsion stiffness	7906N.m/deg	9221N.m/deg
Torsion displacement figure		
Modal: 1st bend	 25.1Hz	 20.55Hz
Modal: 1st torsion	 35.14Hz	 36.15Hz

Figure 7.9 Result of structural calculation

From value of table above, we can see VCD model has bigger stiffness value and lower modal value. The error between detail model and VCD model would not be more than 20%, such as 11.68% for bend stiffness, 16.63% for torsion stiffness and 18.13% for modal of 1st bend. The main reason of bigger error of modal is that the panels were erased form VCD modal to avoid local vibration.

Optimization design by VCD system

Optimization design by VCD system was to decrease the weight of car as well as keeping enough stiffness, the mass is selected as design objective and the stiffness is defined as design constraint. The optimization setup is as following.

- Design Objective: Minimize mass
- Design Constraint: Bend stiffness>28000N/M
- Design Variables were defined in table below
- Relative Convergence: 2.5%
- Absolute Convergence: 0.001

Table 7.1 Reference values and boundaries for optimization

Beam or panel name	Design variable	Variabl e Name	Lower bound	Initial value	Upper bound
B5	Thickness of B5	T_B5	0.5	1.7	3
	Parameter of section shape	O_B5	0.5	1	2
B9	Thickness of B9	T_B9	0.5	2.4	3
	Parameter of section shape	O_B9	0.5	1	2
B14	Thickness of B14	T_B14	0.5	1	3
	Parameter of section shape	OB14	0.5	1	2
Roof _ panel	Thickness of Roof_panel	T_ Roof_ panel	0.5	1.4	3

7.9.4 Results

When the optimization process is finished, VCD system will provide a report by spread sheet with needed optimal data. It can be found from the figure that the mass of the auto-body decreases about 30kg from 319.842kg. Down to 289.622 kg and the stiffness of the optimized body is 28905.4 Nm, which is fit to the defined design constraint that is larger than 28000N.m.

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8 NOISE, VIBRATION AND HASHNESS (NVH) IN VEHICLES

In the chapter number 4 fundamental description of the noise, vibration and hashness problems in vehicles are described. Sources of noise and vibrations are shown in the pictures. Types of vibrations are defined and well known practical methods of reducing noise, vibration and hashness in vehicles are presented. The chapter was prepared with the help of Prof. Wiesław Fiebieg from Wrocław Technical University, who is also leading the company wibroakustyka.pl. We would like to thank for his help and assistance in preparing this chapter.

8.1 Sources of noise and vibration

The noise from a car originates from:

- engine noise
- tyre-road noise
- aerodynamical noise

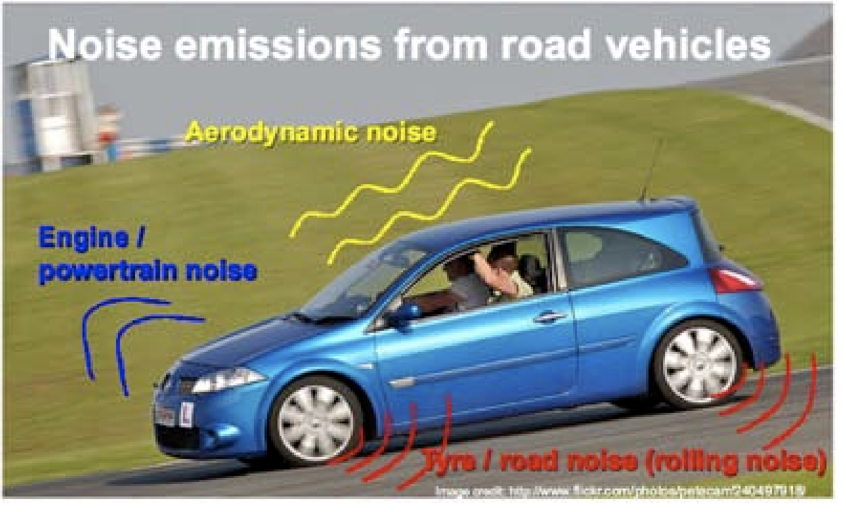

Figure 8.1 The main noise sources in the road vehicle

According to the directive 70/157/EEC, which regulates the technical approval of new vehicles and currently sets noise emission limits of 74dB(A) for passenger cars and 80dB(A) for trucks. The standards for trucks have been somewhat more effective than for cars. The noise limits include all sources of noise from the vehicle.

Tire/road noise has become increasingly important for overall sound quality perception due to the ongoing and successful reduction of powertrain and driveline noise. Road noise generally starts to be noticeable at vehicle speeds above 30 mph, but its contribution to overall interior noise is maximum between 40 and 60 mph and then decreases at higher speeds, where aerodynamic noise becomes predominant. For this reason, tests for road noise are generally conducted at constant conditions, typically 50 mph and in coast down on different road surfaces. Road noise is generated by the interaction between the tire and the road surface and excites the vehicle through both structural and airborne paths.

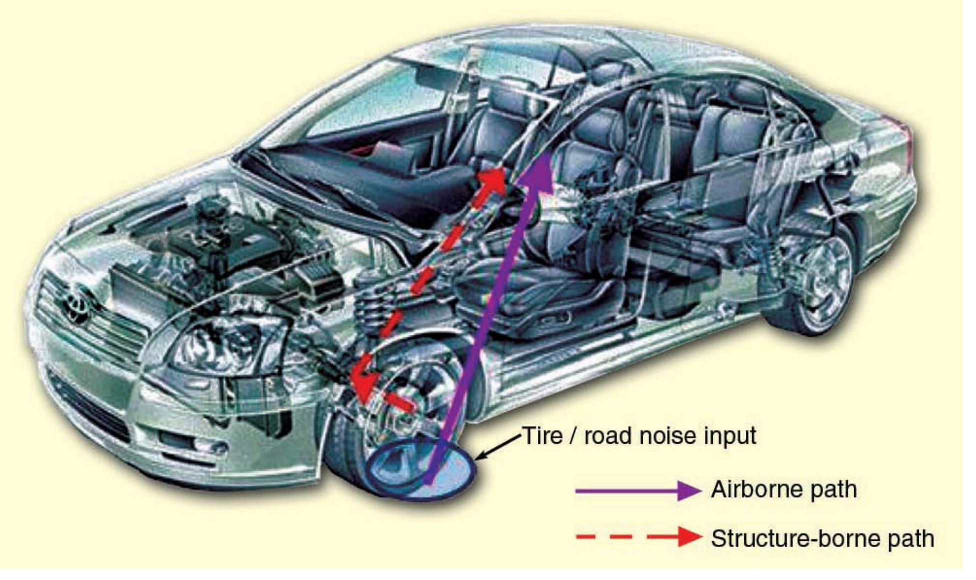

Figure 8.2 Airborne and structure borne noise in the vehicle

There are many sources and causes of noise and vibration in a vehicle— engine, driveline, tires, brakes, wind and electrical drives. While some NVH characteristics are actually admired in certain cases most noise and vibration in the vehicle must be minimised in order to ensure a totally comfortable driving experience.

Engine noise consists from:

- accessory noise
- combustion noise
- mechanical noise
- flow noise

Combustion noise -periodic variation of cylinder pressure acting on the engine structure.

Mechanical noise -piston slap impacts and other mechanical impacts in gears and bearings.

Most common sources of mechanical noise :

- Piston slap which is generated when the piston moves from side to side during the engine cycle.
- Timing gear rattle which is generated when the teeth of the gears impact.
- It is coupled to torsional vibrations of the crankshaft.
- Bearing noise which is generated by impact and vibration caused by fluctuating forces during the engine cycle.

8.1.1 Exhaust and intake noise

The fundamental frequency for a *n* cylinder four stroke engine running with speed *N* rpm: where *N*/60 is the engine speed expressed in Hz and the division with 2 has to do with the fact that the crankshaft makes 2 rotations per engine cycle. Exhaust and intake noise spectra are often expressed in engine orders where the frequency has been normalized by *N*/60, the engine speed expressed in Hz. For a four-cylinder engine the 2nd order would therefore be the fundamental and for a six-cylinder engine it would be the 3rd order

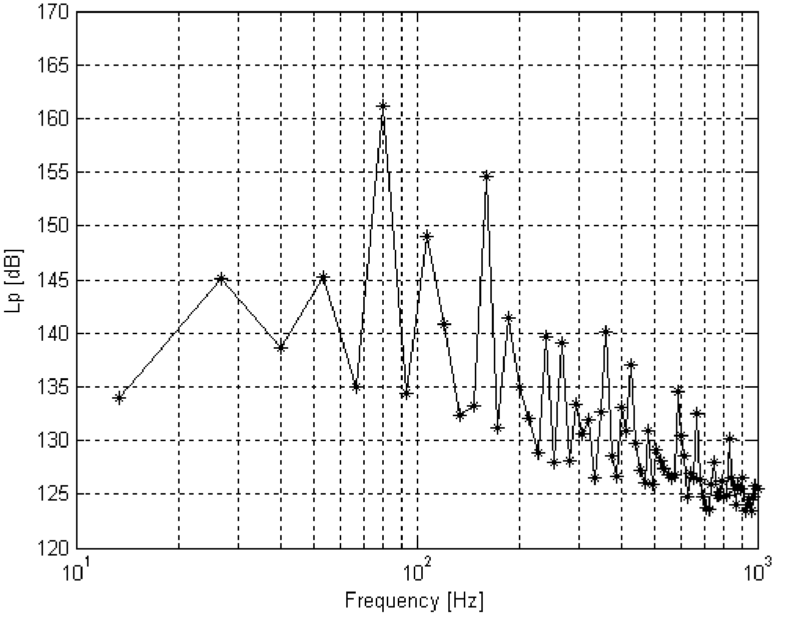

Figure 8.3 Typical shape of exhaust noise spectrum.

8.1.2 Tire noise

Tire casing excitation at roadway roughness frequency

- Tread block Impact
- Air Pumping
- Stick-slip (friction)
- Stick-snap (adhesion)
- Horn effects (amplification)
- Absorption (source strength and propagation)
- Closed cavity effects

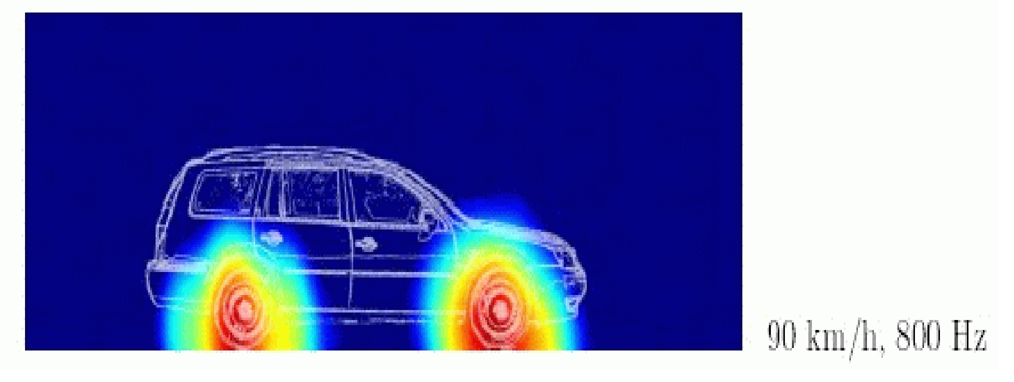

Figure 8.4 Tire road interaction noise at 800 Hz for a car driving at 90 km/h.

8.1.3 Brake noise

Brake system is another important source of noise and vibration. The brake system NVH will be used to demonstrate the complexity and the challenges that the automotive industry is facing in one of the NVH areas. General NVH characteristics of brake system include high frequency squeal, low frequency squeal, moan/groan noise and vehicle judder or roughness. The industry achieved great success in detecting and to some degree good understanding of the mechanisms of these NVH attributes. However, many challenges still exist regarding how to control and prevent them at the component design level.

8.1.4 Aerodynamic noise

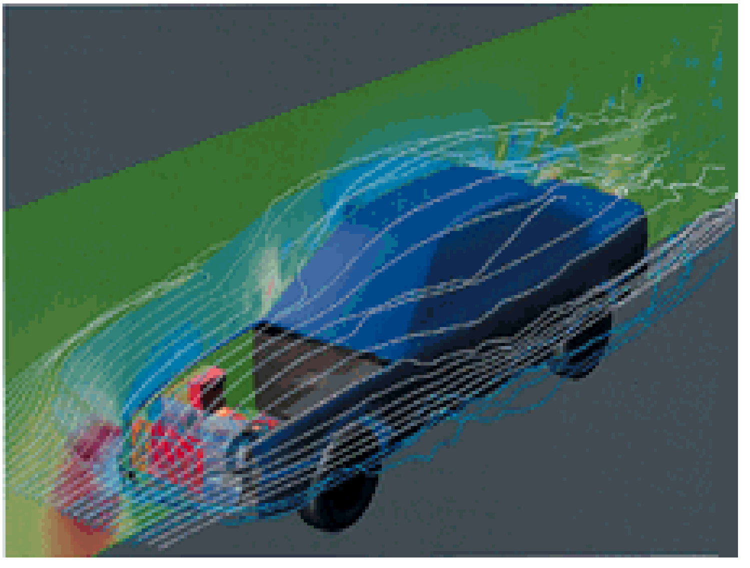

Figure 8.5 Simulation of flow field around a car causing aerodynamic noise generation at high vehicle speeds

Aerodynamic noise is the predominant component of interior vehicle noise at speeds above 100 km/h. It is typically tested at steady vehicle speeds between 100 and 160 km/h, either on the road or

in a wind tunnel. Wind noise refers to the following noise and conditions:

• Aerodynamic noise made by the vehicle as it moves at high speed through a steady medium (air). This is related to the aerodynamic (or drag) coefficient of the vehicle, which is a function of the vehicle shape and its cross-sectional area.

• Aerodynamic noise due to turbulence through “holes,” which is correlated to how tightly sealed the vehicle is (around doors, hood, windshield etc.).

• Aerodynamic noise due to exterior varying wind conditions, such as cross-wind on a highway. This is different from the previous two, since this type of wind noise is fluctuating.

8.2 Control techniques to vehicle noise and vibration

Number of ways in which the final sound radiation may be influenced:

- Reduction at the source of combustion forces and mechanical forces.
- Reduction of the vibration transmission between the sources and the outer surface.
- Reduction of the sound radiation of the outer surface.

8.2.1 Engine noise control

Engine noise is caused by various types of force generation within the engine and is transmitted to the radiating outer surfaces. The transmission path properties are determined by the vibration modes of the structure. The properties of the outer surface will also influence the sound radiation. number of ways in which the final sound radiation may be influenced:

- Reduction at the source of combustion forces and mechanical forces.
- Reduction of the vibration transmission between the sources and the outer surface.
- Reduction of the sound radiation of the outer surface.

Reduction of combustion pressures is intimately coupled to changes in the combustion process or combustion chamber shape. Since any changes to the design of the combustion chamber or to the combustion process will also have an effect on engine performance and exhaust gas emissions this is a difficult path for a noise control engineer. Unfortunately most design changes which would reduce noise would also increase exhaust emissions. Piston slap can be reduced by redesign of the piston and cylinder or by oil film injection

Gear and bearing noise can be reduced by improved design of these components for instance attention to gear tooth profiles and bearing clearances. To reduce the transmission of vibrations to the engine outer surface the crank case and cylinder block can be redesigned. One example is strengthening and stiffening of the structure. Other more advanced redesigns can be made involving extensive simulation the dynamics using for instance finite element modeling.

8.2.2 Exhaust and intake noise control

Exhaust and intake system noise originates, from the pressure pulsations caused by the operation of the engine and additional flow generated noise. To control the noise generation at the source involves making changes to the combustion process, combustion chamber or exhaust or intake valves which also influences engine performance and exhaust gas emissions. Instead often mufflers or silencers are used which are placed in a flow duct to prevent sound from reaching the openings of the duct and radiating as far-field sound. Reactive silencers do this by reflecting sound back towards the source while absorptive silencers attenuate sound using absorbing material. No car or truck can pass the standard noise tests required by legislation or compete on the market without them. There are three basic requirements for a modern exhaust systems;

- compact outer geometry,
- sufficient attenuation,
- low pressure drop.

Different practical methods are used for noise reduction. Mufflers or silencers which are devices used in a flow duct to prevent sound from reaching the openings of the duct and radiating as far-field sound reduction in mufflers: use of sound absorbing materials in which sound energy is converted into heat mainly by viscous processes. Typical sound absorbing materials used are rock wool, glass wool and plastic foams, steel wool, perforated steel, sheet mineral wool.

The exhaust system can be a source of noise. Exhaust noises include:

- Exhaust gas sounds
- Muffler and pipe

Exhaust gas sounds are further subdivided into three categories:

- Pulsating
- Air column resonance
- Air stream sounds

Exhaust gases are discharged each time opened exhaust valves are creating a pulsating sound. The sound is cyclic and is associated with engine speed and the number of cylinders. The sound is relatively low-pitched, consisting mainly of this basic frequency.

Air-column resonance consists of sounds produced in exhaust pipes and mufflers. Pipe length and the cross-sectional area of the pipe determine the frequency. Air stream sounds can be produced by high-speed exhaust. An example of this is turbulence caused by air going through the muffler, or jet noise when exhaust is discharged from the tailpipe.

Muffler and exhaust noises can be caused by exhaust system misalignment, incorrectly installed or damaged mounting brackets, or failed hangers. This can cause a variety of annoying noises that can be located with a thorough visual inspection.

Any of these sounds in the exhaust system can be carried, or transmitted, throughout many of the exhaust system components and into the passenger compartment. The compelling force of exhaust exiting each cylinder creates a pulsation on the exhaust manifold. The pulsating pressure at the exhaust manifold produces acoustic energy, which is transmitted to the exhaust pipe. The pulsating sound waves traveling through the exhaust pipe are a source of vibration for the exhaust system. Exhaust vibrations can become amplified by resonating with engine firing frequencies and vibrations caused by the reciprocal motion of the pistons. The combination of these vibrations can produce unwanted NVH concerns.

8.2.3 Interior noise and vibration control

This kind of control is motivated by driver and passenger sound and vibration comfort. Sound level in dB(A) is therefore not a good measure for interior noise. Instead sound quality measures such as loudness or harshness are used. Main source: engine noise. Other sources: tyre-road noise, exhaust noise, intake noise, cooling system noise and ventilation system noise.

8.3 Effects on hearing

Not all the effects of vehicle noise pollution are related to stress, however. Millions of citizens worldwide are hampered by hearing loss, much of which is sound induced. A quick description of how the ear works is necessary at this point: When noise reaches the inner ear it is transduced by hair cells, which line the inner ear, into nerve impulses and then is transmitted to the brain, where it is perceived as sound.

There are essentially two ways in which sound induced hearing loss occurs: short term exposure and long-term exposure. Short term exposure refers to loud bursts of sound that exceed 120dB (threshold for pain), such as rock concerts or gun shots. Fortunately, this hearing loss is often only temporary.

Such is not the case, however, with long term exposure. "Hearing loss due to prolonged noise exposure is generally associated with destruction of the hair cells of the inner ear. The number of hair cells damaged or destroyed increases with increasing intensity and duration of noise and, in general, progressive loss of hair cells is accompanied by progressive loss of hearing." (World Health Organization, Sweden 1995) One then, cannot simply expect that by disappearing into the silence of the forests for a while, the effects of hearing loss will be reversed. Once the hair cells are destroyed, they do not grow back, and your hearing will never be what it once was.

8.4 Ride Comfort

Ride comfort plays a large part in a customer’s satisfaction with their vehicle. Avoiding abnormal vibrations ensures a quality ride comfort level.

Normal vehicle vibrations are a result of road roughness. During normal operation, the vehicle experiences vibration between the sprung components (body and suspension) and the unsprung components (axles, tires, and wheels). This is an acceptable condition unless the sprung or unsprung components become defective, worn, or damaged.

When unsprung components resonate with the sprung components, the result is poor ride comfort. Ride comfort vibrations may cause the vehicle to roll, pitch, and bounce, which may cause a customer concern. Poor ride comfort can be minimized by ensuring suspension and steering components are not damaged or worn.

8.5 Types of Vibration

Vibrations in a vehicle can be any one of the following types:

- Shake
- Shimmy
- Brake Vibration/Shudder

8.5.1 Shake

Vibrations at the steering wheel or seat, or an annoying vibration at the floor, are indicators of "shake." Shake generally has a frequency of 10 to 30 Hz. There are two types of shake:

- Vertical (up-and-down)
- Lateral (side-to-side)

Vertical shake is severe vertical vibration of the body, seats, and steering wheel. A trembling engine hood or rearview mirror also can be a vertical shake symptom.

Lateral shake is side-to-side vibration of the body, seats, and steering wheel. A trembling vibration in the driver's waist or hips may be a symptom of a lateral shake.

The major vibration sources of vertical and lateral shake are:

- Roughness of road
- Tire unbalance
- Non-uniform tires
- Bent or out-of-round wheels
- Driveline

• Engine

8.5.2 Shimmy

Vibration that causes the steering wheel to oscillate is known as “shimmy.” The body of the vehicle also may vibrate laterally. Shimmy generally has a frequency of 5 to 15 Hz. There are two types of shimmy:

• High-speed shimmy

• Low-speed shimmy

High-speed shimmy occurs when driving on smooth roads at high speeds. High-speed shimmy typically has a limited speed range in which symptoms are noticeable.

Low-speed shimmy occurs when the steering wheel begins to vibrate as the vehicle is driven across a bump at low speeds.

The major vibration sources of high-speed and low-speed shimmy are:

• Roughness of road

• Tire unbalance

• Non-uniform tires

• Bent or out-of-round wheels

For example, a tire with excessive runout, out-of-balance, or out-of-round may cause high or low-speed shimmy. This is because the tire fault generates a vibration at a particular frequency. When the vibration of the tire reaches the natural frequency of the vehicle’s front unsprung components (such as the front axle, tires, and wheels), they start to vibrate. When the frequency of the front unsprung components matches the natural frequency of the steering system, resonance occurs. This resonance causes the steering wheel to vibrate heavily in the turning direction.

8.5.3 Brake Vibration/Shudder

Brake vibration/shudder is transmitted through the brake hydraulic lines to the suspension system, steering system, and the brake pedal. Brake pedal pulsation is generated when applying a brake with a non-uniform diameter drum or a disc brake with non-uniform brake disc thickness.

Brake shudder causes the instrument panel, steering wheel, and sometimes the entire body to vibrate vertically and back-and-forth during braking. It also may result in brake pedal pulsation related to wheel rotation and can occur during any braking condition or vehicle speed. Normally, brake shudder has a peak at 40 to 50 mph (60 to 80 kph) and has a frequency of 5 to 30 Hz.

If the disc rotor has excessive thickness variation, friction force on the braking surface varies during brake application. The change in the braking force generates a vibration at a certain frequency. This vibration is transmitted to the suspension, steering, and brake pedal. The vibration can also transmit to the body, causing it to resonate.

8.6 Vibrations

Vibrations are noticeable at the steering wheel, seats, and floor. The level and intensity of the vibration changes with the suspension type and the bushings used.

Longitudinal impact forces are transmitted to the lower arms where suspension bushings dampen the vibration. The dampened vibration then transmits through the suspension crossmember and strut insulators to the body.

The rigidity of the bushings and insulators in the vibration transmission path has a large influence on harshness. The use of low-rigidity bushings and insulators to provide greater fore-aft suspension compliance softens the impact force effectively, but results in less responsive steering.

Along with suspension system designs, tire characteristics influence the amount of harshness. When tires experience an impact force from a pavement joint, they deform to cushion the rest of the vehicle. The tires absorb the force to some extent and are an important factor when dealing with harshness. At the same time, the deformation transmits vibration to other portions of the tires. This causes the tires to undergo complicated resilient vibrations. A tire that can absorb vibrations from impact is efficient in controlling the problem of harshness.

It is extremely difficult for a tire to absorb harshness vibrations completely. These vibrations involve not only tire type, but also inflation pressure. The technician should always ensure proper tire inflation pressure when troubleshooting a harshness concern.

8.6.1 Engine Mounts

The first components that isolate vibration from the engine to the passenger compartment are engine mounts. Engine and transmission mounting consists of a number of relatively small parts and are sometimes ignored when troubleshooting. However, these parts are extremely important in preventing noise and vibration produced by the engine.

Torque reaction force of the engine acts directly on the transmission, causing engine mounts to be subjected to a large force. Therefore, engine mounts must be rigid to stabilize the engine. On the other hand, to minimize inherent engine vibrations and noise during all engine speeds, the engine mounts also must be soft. Any fault in the engine mounting system can lead directly to noise and vibration.

Inspect engine mounts for cracks or damage to the insulator and the bracket. Grounded, the engine mounted bracket contacting the frame-mounted bracket, or strained engine mounts may not isolate engine vibrations.

Engine mounts must be installed correctly. If the mounts are installed incorrectly or incorrect parts are used, they cannot absorb engine vibration. These valves soften or stiffen the engine mounts depending upon engine operating characteristics.

8.6.2 Suspension Components

The suspension system plays a large role in the effects of NVH on a vehicle. Similar to the steering system components, the suspension of a vehicle can easily act as the medium through which vibration can be transferred. When the source of an NVH concern is determined, you usually can find the cause by performing a thorough visual inspection.

Passengers of a vehicle are sensitive to the effects of a harsh riding suspension. For this reason, various measures are taken in the design and development phase of a vehicle’s suspension. For example, springs and bushings are carefully selected to provide a smooth ride and also to reduce the impact of steering forces.

The vibration of a vehicle’s suspension system consists of vertical and longitudinal forces. The vertical forces are controlled by well-proportioned sprung and unsprung weights, shocks, and coil springs.

Longitudinal forces are controlled by careful selection of suspension bushings. The suspension of a particular vehicle is designed so that the lower control arms absorb longitudinal impacts from the tires. Tire impact is further dampened by bushings before being transmitted through the crossmember to the body. This longitudinal compliance depends on the spring constant rate of the rubber insulator bushings.

8.7 Noise measurements

The noise measurements are carried out in the anechoic or semianechoic chambers. The conditions in such rooms are similar like in free field. The influence of different design measures on the noise levels can be evaluated. Noise sources identification will be performed with the sound intensity method or with the acoustic holography. For measurements on instationary noise sources the beamforming method will be used.

Figure 8.6 Semi anechoic chamber for noise measurements

References

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9 MATERIALS IN VEHICLES

In the past vehicles were mainly made out of steel. Nowadays more and more new materials come into the automotive industry to improve vehicle efficiency and safety. Steel is still the main material used for cars but its properties had risen.

9.1 Metals

9.1.1 Place of metals in modern car

The very major direction the present-day car industry looks on now is to reduce weight, save costs and make cars more environmentally friendly. Despite of the fact that polymers are being used more often and implemented into wider group of car body elements, metals have not been defeated yet.

Regarding metals, there has been a trend toward the use of light metals and their alloys in automotive components, particularly automotive bodies. The most commonly used materials are aluminium, magnesium, and their alloys.0T The amount of high- and medium-strength steel, aluminium, and plastics used in automobiles has been growing, while the amount of regular steel sheet and iron has declined. The change to lighter weight materials has helped the average automobile to be about 250 pounds lighter in 2004 than in 1977. Steel (all kinds combined) makes up more than half of the weight of an automobile.

On the other side there has been strong competition from the steel industry, which for obvious reasons would prefer that steel be retained as the primary automotive material. On modern cars, most of the weight comes from steel. In 2007, for example, the average car contained 1,090 kilograms of steel, and the average pickup truck or SUV used nearly 1,360 kilograms. Consider that most SUVs now weigh around 1,810 kilograms.

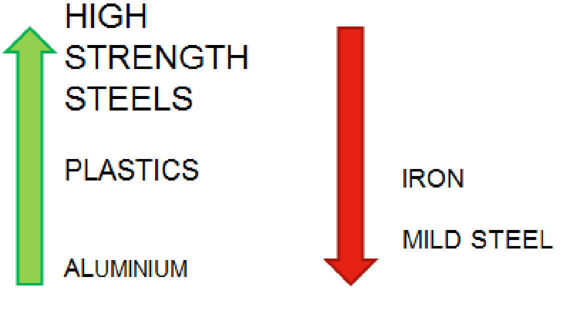

Figure 9.1 Current tendency in vehicle materials

As the automotive industry addresses environmental concerns, the problem of fuel consumption and weight reduction has come to the fore.

Advanced materials, including metals, polymers, composites, and intermetallic compounds, can play an important role in improving the efficiency of transportation engines and vehicles. Weight reduction is one of the most effective ways to increase the fuel economy of vehicles while reducing exhaust emissions.

The use of lightweight, high-performance materials will contribute to the development of vehicles that provide better fuel economy, yet are comparable in size, comfort, and safety to today's vehicles. The development of propulsion materials and enabling technologies will help reduce costs while improving the durability, efficiency, and performance of advanced internal combustion, diesel, hybrid, and fuel-cell-powered vehicles. The weight, size and design (including materials) of a car body all contribute to its behaviour in an accident (crashworthiness) and the safety of the passengers.

9.1.2 Material use relation

In the front and rear of the vehicle, where there is more room to absorb the energy of a collision, steels that deform more easily, and get stronger as they do, might be used. Even body panels, which are usually made from milder steel, are bake hardenable, getting stronger as they are heated during paint curing to resist denting better.

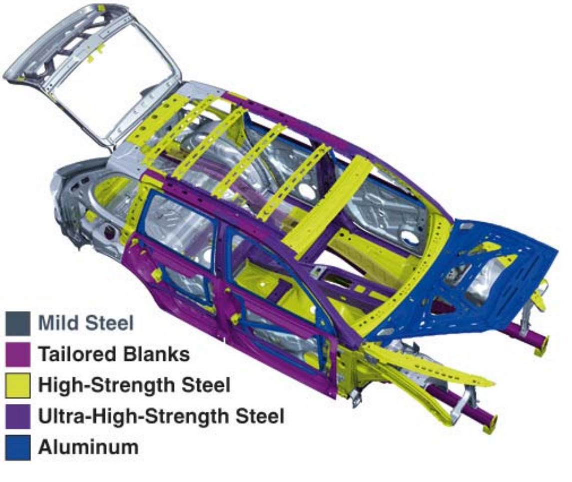

Figure 9.2 Materials used in vehicle chassis

The weight, size and design (including materials) of a car body all contribute to its behaviour in an accident (crashworthiness) and the safety of the passengers. Aluminium structures are capable of absorbing energy equivalent to those using steels. But the proven performance of steel-bodied cars in a wide range of countless real life crashes cannot be reproduced easily. Aluminium costs five times as much as mild steel, however, and while bare aluminium is undoubtedly more resistant to atmospheric corrosion than bare steel, new coatings and galvanised steels mean that corrosion is no longer the primary determinant of a car's lifespan.

9.1.3 Steel

On modern cars, most of the weight comes from steel. In 2007, for example, the average car contained 1,090 kilograms of steel, and the average pickup truck or SUV used nearly 1,360 kilograms. Consider that most SUVs now weigh around 1,810 kilograms.

Often disregarded, ease and cost of repair are now increasingly important as newer grades of high strength steels and aluminium and other materials emerge, requiring precise retreatments involving more sophisticated equipment to ensure original standards can be met. These procedures can have an important bearing on the insurance category derived for specific vehicles.

In cars, steel is used to create the underlying chassis or cage beneath the body that forms the skeleton of the vehicle and protects people in the event of a crash. Door beams, roofs and even body panels created during auto manufacturing are made of steel on most cars today. Steel is also used in a variety of areas throughout the body to accommodate the engine or other parts. Exhausts are often made from stainless steel, for example.

Steel improvements

The steel industry has worked to develop new products, to keep steel in cars and to fend off the increased use of aluminium and other materials. They have been developing higher-strength products for decades, beginning with high-strength low-alloy, or H.S.L.A., steels. But most were costly to produce and difficult to make into parts.

Only in recent years have researchers and manufacturers figured out how to make economical high-strength steels that are pliant enough to be stamped and formed and that can be welded or otherwise joined to other parts in the complex auto assembly process, where time is money.

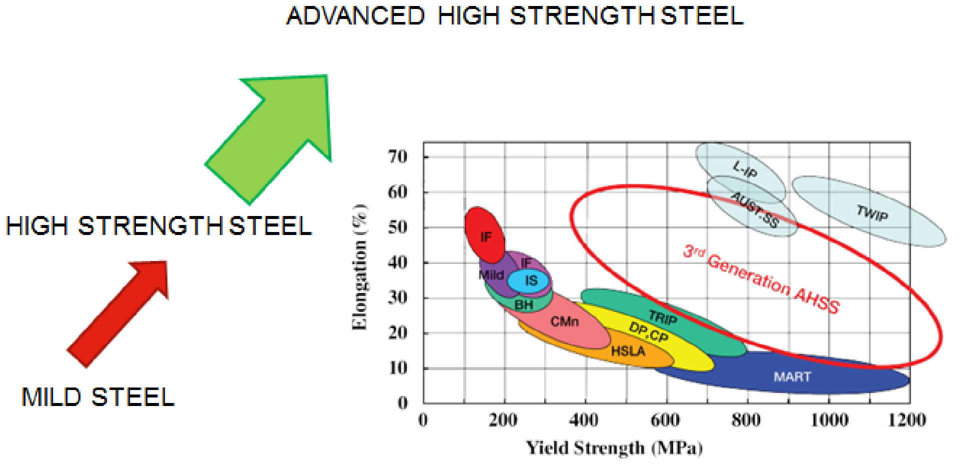

Figure 9.3 Steels used in vehicles evolution

Mild steel

Mild steel is the most common material in use in the car industry. Despite the in-roads made by aluminium and polymers, steel still offers great value in mass production and more simpler production processes than its rivals. Improved galvanising and painting techniques mean that steel is holding more ground than ever before in terms of corrosion, one of its biggest problems. Steel is cheap to obtain by comparison to Aluminium and is readily available through recycling.

- Pros:

  Cheap, available and well understood. Mild steel can be prepared and processed using traditional tooling and responds well to standard joining processes such as spot and seam welding as well as bolting. Steel's simplicity and ubiquity make it easy to repair.

- Cons:

  Steel requires good preparation to ensure it does not corrode in the presence of air and water, especially in cold climates where salt is also present on roads. Steel is heavier than its immediate rivals and adds to vehicle weight; this affects fuel consumption and to a lesser extent momentum (in impact) and handling.

Figure 9.4 Exhausts - typical use of mild steel

High strength steel

Using higher-strength steel, a part can be downgauged — made from thinner stock — to help improve fuel economy. But some parts, notably exterior panels, are already so thin — less than a millimetre, or one twenty-fifth of an inch, in many cases — that stiffness becomes an issue. “You reach a thickness where you can’t go even thinner,”

The advanced steels go by names like D.P. (for dual phase) and TRIP (for transformation-induced plasticity). The strongest ones are used in parts like door beams, where the aim is to stop a foreign object (like another car’s bumper) from entering the passenger compartment, and windshield pillars, where the goal is to prevent the roof from flattening like a pancake in a rollover.

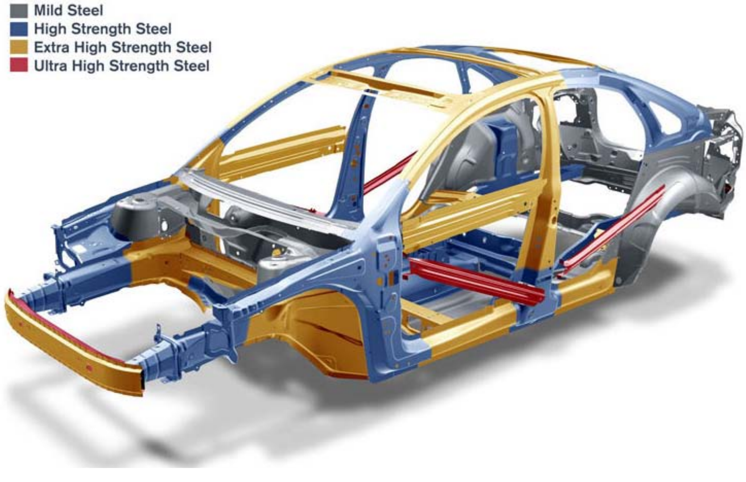

Figure 9.5 Steels used for car chassis elements

9.1.4 Magnesium alloys

Magnesium (Mg) alloys, with the lowest density of all structural metals, have the potential to reduce component weight by greater than 60%. However, significant technical barriers limit the use of Mg to approximately 1% of the average vehicle by weight. These barriers include high raw material cost and price volatility, relatively low specific stiffness, difficulty in forming sheet at low temperatures, low ductility of finished components, and a limited alloy set, among others. In addition, using Mg in multi-material systems introduces joining, corrosion, repair, and recycling issues that must be addressed.

9.1.5 Aluminium alloys

Aluminium (AL) alloys represent a middle-ground in the structural light metals spectrum. Years of development within the aerospace, construction, and automotive industries have led to a well-developed and reasonably well understood alloy and processing set. Applications of Al within automotive design include hoods and panels, power train components, and even entire vehicle body-in-white (BIW) structures. There are several barriers to the increased use of Al in vehicle weight reduction applications such as material cost, room temperature formability, and limitations within the existing manufacturing infrastructure. As with Mg, the addition of significant amounts of AL to the automotive manufacturing stream presents added multilateral challenges in joining, corrosion, paint and coatings, repair, and recycling.

Aluminium is hailed by Audi as a suitable material for all elements of a vehicle's body. Whilst it can be used to save weight and resist corrosion, it does have its own concerns. Firstly, slightly more complex TIG welding processes need to be used in production (and repair) making it (currently) more costly than steel to work with. Where space-frame construction is employed, aluminium is often bonded with epoxy which creates strong bonds with good force distribution but is very difficult to repair.

In North America legislative pressure to reduce fuel consumption has sparked the search for a lighter car. Aluminium and plastics can indeed produce vehicles that are lighter than current steel models. And these lighter vehicles also have other benefits, such as fewer parts, by using space frame construction though steel could also do this. An aluminium panel weighs about half as much as a steel panel of equivalent strength, and using more aluminium could, it is claimed, also meet other criteria, although the past 15 years have seen considerable savings (albeit offset by luxury fittings and safety features) achieved through rationalisation of car body structures and the use of lighter gauge, higher strength steels.

At present, alternative materials are most competitive in low volume production where tooling, rather than materials, most affects unit cost. Aluminium could reduce body weight by up to 40%, but new steel technologies promise reductions of up to 35%, leaving aluminium only just ahead.

Figure 9.6 Audi A2 full aluminium body

- <u>Pros:</u>

  Aluminium is light and resistant to corrosion. Large weight savings can be made when using aluminium for the engine block.

- <u>Cons:</u>

  Aluminium is currently more expensive to manufacture with than steel. Corrosion can still take place if aluminium is placed on a less reactive metal or exposed to salt. Less well-established joining and bonding techniques can make aluminium awkward to repair. Aluminium welding poses special problems. It requires more welding spots to compensate for lower fatigue strength. Fusion welding is difficult because of oxide formation, frequently making MIG/TIG welding necessary. And aluminium's higher surface reflectivity makes laser welding more difficult. Steels require lower welding currents and lower contact pressures, while electrode life is longer, and energy consumption three times less.

Figure 9.7 Audi R8 - aluminium alloys used in chassis and body

9.1.6 Environmental causes

Environmental issues increasingly influence automotive design, particularly noise, recycling and life-cycle pollution. In theory, an aluminium structure’s resistance to vibration is one third that of an identical steel structure. This means 10 db more noise, while steel's higher sound-damping capacity reduces road noise. Aluminium cars can compete by alternative designs or insulation, but the penalty is extra cost in materials and more weight. Passenger comfort may also be reduced: aluminium's thermal conductivity is five times that of steel. This presents problems with external and internal temperatures on hot days so air conditioning may become a necessary extra.

Recyclability

Legislation in Germany and North America appears to be heading towards the 100% recyclable car. Marketing campaigns have presented aluminium as a highly recyclable material. Pound for pound it has ten times the value of steel sheet scrap. Aluminium, the material, is potentially recyclable with an appropriate infrastructure and a limited mix of alloys.

But recycling aluminium cars is highly complex; the different products and alloys likely to be used are incompatible for recycling into wrought products for car bodies, and it will take 10-15 years to create an adequate stockpile of scrap, by which time the alloys in use today may be obsolete.

But steel is the world’s most recycled material and recycling is as old as steel production itself. Over 400m tons are recycled worldwide each year. This is about ten times the combined total of all other automotive body materials. Secondary aluminium production in Europe in 1991 was only 1.6m tonnes. These 400m tons include 20m tons of automotive steel scrap in the USA and Europe using existing recycling networks. Plastics recycling is minimal and automotive plastics are causing problems with landfill space. Steel can be separated magnetically, while plastic resins or aluminium alloys must be segregated with 100% accuracy, because of their low tolerance to impurities. Different molten aluminium alloys could be mixed, but only at the expense of downgrading.

Some 80% of today's car is recycled (including over 95% of the steel), requiring much less energy than refining from ore. The energy required for primary aluminium production is five times that of steel, and producing a part from aluminium, rather than steel, needs almost three times the energy. Although the excess energy in vehicle parts manufacture is 'paid back' during its life, there is an overall deficit because of increased emission of greenhouse gases in primary aluminium production, and other problems, such as the caustic 'red mud' produced in alumina extraction, and disposal of toxic pot liners containing cyanides and fluorides.

Infrastructure

The largest, and most immediate problem in high-volume production with alternative materials is the expense of a new infrastructure to handle design, manufacture and repair. Much of that used for steel cars is either unsuitable or incompatible. For example, aluminium and plastics cannot use presses with magnetic handling so new handling and post stamping facilities would be needed. Plastics pose problems with fixing and painting and, like aluminium, are difficult to integrate with monocoque steel body design, so new joining techniques would be needed.

9.1.7 A look into the future

Car manufacturers are examining future options, but have already decided the way ahead for the near future. Very few of the future models are known to include significant amounts of aluminium or plastic body panels. Meanwhile, Chrysler is replacing the plastic wing on its LHS with steel, and problems with the Viper's plastic bumper led to 66% of 1993 production being lost.

Radical new designs, such as space-frames, could make aluminium and plastic panels viable, but are still at the experimental stage. Monocoque bodies - the most economical construction - are designed for steel panels and substituting alternatives would lead to problems. Few manufacturers have worked with aluminium in mass production, and introducing new product forms and working practices will take time and money. While aluminium alloys and plastics undoubtedly offer some benefits, car manufacturers are nothing if not realistic and the bottom line is: alternative materials may exist, but are they cost-effective in producing safer, 'greener' cars? With today's steels, an annual run of just 25,000 units could save US$140 per car (IISI study Competition between steel and aluminium for the passenger car), or US$525 on a run of 200,000 units.

By the time the claims made for aluminium, plastic and composite exteriors have been examined and proven (or not) steel will probably have evolved still further, maintaining, or even improving, its current advantage and preventing mass penetration of the all-important high volume markets.

AHSS – future of car materials

Conventional iron and steel alloys are prominent in existing vehicle architectures, making up over 70% of the weight of a vehicle. Despite the relatively high density of iron based materials, the exceptional strength and ductility of advanced steel offer the potential for efficient structural designs and reduced weight. Application of a new generation of AHSS has the potential to reduce component weight by up to 25%, particularly in strength limited designs. Steel components are also generally compatible with existing manufacturing infrastructure and vehicle materials, making them a likely candidate for near-term weight reduction. Steel development and research in the LM activity is focused on introducing the so-called "3rd generation". 3rd generation AHSS are targeted to properties in between 1st and 2nd generation AHSS with high strength, improved ductility, and low cost.

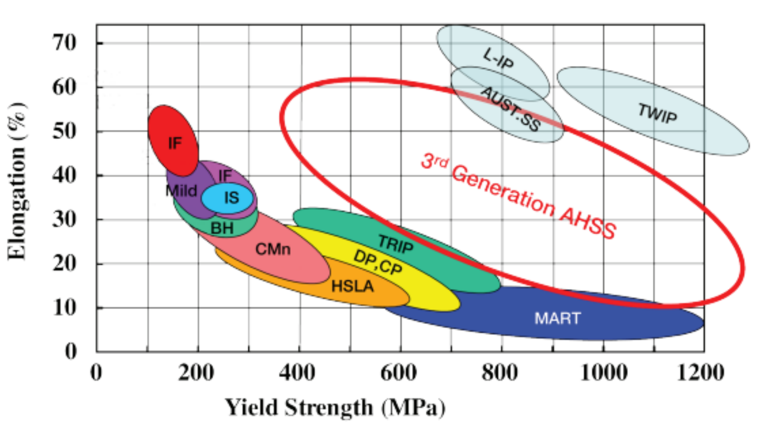

Figure 9.8 AHSS – future of car materials

9.2 Plastics

Throughout the whole vehicles plastics make a decisive contribution to safety, reliability, comfort and eco-efficiency.

Plastics are being used in:

- engine and chassis;
- tanks, wires, cables, ventilators, steering wheels, suspension, mechanisms;
- car body;
- buffers, heating and cooling system, interior outfit;
- electronics;
- ignition system, lightning, wires.

9.2.1 Comfort, safety and economics

People spend a lot of time in their cars these days. In fact, in the western world, drivers and passengers spend a significant part of their lives in cars. In Europe, the figure is 274 hours per person a year, while in the US it is 541 hours a year.

That is why the comfort, safety and economics are most important requirements in today's automotive industry.

Figure 9.9 Use of plastics in passenger compartment

Good economical properties can be obtained by reducing mass of the vehicles. Usage of plastics and component materials is one of ways to reduce mass of the vehicle without decreasing its strength.

Increased fuel efficiency is not the only benefit. A lighter weight of the vehicle reduces the CO2 created by the engine.

Global environmental regulations set stringent CO2 emissions standards for the automotive industry to meet by 2012. As a result, engineers are faced with making significant vehicle design changes in a tight timeframe; all without sacrificing the comfort and quality consumers expect.

Customers desire comfort with quality. Comfort and esthetics for most customers are the primary criteria of buying a car. Plastics meets those requirements.

Esthetics of vehicles is a study of artistic values of vehicle. We assess this vehicles in the case of appearance. This is very useful when we want to formulate some criterion of worth of vehicle.

9.2.2 Use of plastics in vehicles

Laminated windows

Weight reduction: Laminated windscreens can reduce vehicle weight, by allowing the use of thinner glass in the glazing configuration. The typical weight reduction for a standard 2.3/2.3mm windscreen when replaced with a 2.1mm/1.6mm windscreen is 3.5kgs.

Safety: PVB interlayer can meet or exceed emerging penetration standards for vehicle side window intrusion resistance and occupant ejection mitigation.

Intrusion resistance: With standard tempered side windows, it takes a thief less than two seconds to break the glass and enter your car. But laminated glass provide up to 2 minutes of intrusion resistance

It takes a thief 20 to 30 seconds of continuous pounding to break laminated glass made with protective interlayer. By the time it takes them to enter the car, most opportunistic thieves just give up.

Figure 9.10 Laminated window

Acoustic windshields

Decreasing the noise inside the cabin is a result of the special acoustic interlayer found inside the windshield.

With so much time spent in cars drivers would like to have a quieter experience when driving. Wind and road noise can feel distracting, and even fatiguing after long hours spent driving.

Acoustic windshields are the same as a standard windshield, but they have been made with a thin, sound absorbing technology between the glass that reduces the interior noise by 3dB overall, and even more in the frequency where people "hear" the human voice.

Figure 9.11 Acoustic windshield

Illuminated licence plates

Illuminated license plates are much more visible than standard license plates. A polycarbonate coat emits much more light. Better visibility increases safety on the road.

Figure 9.12 Difference in license plate visibility - from left side: standard, polycarbonate coated

Wheels

Run flat tire is an innovative solution but there was even better idea found. An innovative polymer net is used to design a new type of tire which needs no air at all. This invention was made for military vehicle needs.

Its design is based on honeycomb shape. The aim was to reduce the difference between the stiffness in the tire and to make it more uniform and homogenous. This special design makes it more bombproof.

Figure 9.13 Tyre designed for military vehicles

Body

Carbon fibre reinforced plastics (CFRP) is lightweight, torsion resistant material contributed significantly to reducing vehicles weight.

Figure 9.14 CFRP body and aerodynamics of a race car

BMW Z4 GT3 (shown in fig 9.14) uses rear wings, bonnet, roof, fender and many other components made out of CFRP.

Carbon brake discs

Brakes both in race cars and road cars convert movement energy to heat.

This brake rotors are a PA based and Carbon Fibres filled material.

Formula One cars must sometimes decelerate in a matter of seconds from 350 km/h to about 70 km/h. During such heavy braking, the temperature of the brake rotor and pads can warm up from 400°C to more than 1000°C. These 1000°C occurs at the very end of the braking, and is approximately the highest temperature a carbon brake disc. Propertied of such disc are very high at very low weight.

Figure 9.15 Red-hot ceramics brakes on a F1 car

Lightweight clutch

Vehicle clutch works very similar to a brakes. Thanks to the friction momentum can be transferred from the drivetrain to the wheels. Carbon fibre is also used for covers.

Advantages of ceramic clutch:

- very light;
- very good friction coefficient, so you can use a fairly soft spring = light pedal;
- high internal damping - engage as smoothly as an OEM organic clutch;
- contain solid lubricants - light on the flywheel facing too;
- huge amounts of heat can be dissipated;
- more chemical than mechanical based contact – good wear rates.

Figure 9.16 Carbon fibre clutch

9.2.3 Carbon-neutral plastics

The carbon-neutral plastics are being called "ecological plastic." These plastics are being used in scuff plates, headliners, seat cushions and other interior vehicle parts.

Ecological Plastic is responsible for less carbon dioxide (CO2) emissions during a product's lifecycle (from manufacture to disposal) than plastic made solely from petroleum.

There are two types of ecological plastic--that produced completely from plant-derived materials and that produced from a combination of plant- and petroleum-derived materials.

9.3 Ceramics

Engine designers are always looking for alternative approaches to lower cost and emissions and increase fuel economy and performance of automobiles. One approach to improve automobile designs is through material substitutions. Ceramics may be the enabling technology for many critical components in engines of the future because of their unique heat, wear, and corrosion resistance, light weight, and electrical and heat insulation. Ceramics are inorganic and non-metallic materials that are commonly electrical and thermal insulators, brittle and composed of more than one element (e.g., two in Al2O3). As ceramics are composed of two or more elements, their crystal structures are generally more complex than those of metals.

Ceramics characteristics:

- high compression strength considerable higher than tensile strength
- high Young’s modulus (10-10^3 GPa)
- high hardness and high abrasion resistance

- brittleness
- lower density in comparison with metals
- very high temperature of melting point (2000-4000˚C) and connected with this, resistance to very high (but constant) temperatures
- high chemical stability and corrosion resistance
- low thermal and electrical conductivity
- creep resistance

Using a ceramic composite takes advantage of a material with outstanding hardness (and potentially long life) and an ability to retain its strength and shape at temperatures that would melt conventional iron brake material into a glowing puddle.

Use of ceramics in vehicles

Spark plugs

A spark plug is composed of a shell, insulator and the central conductor. It pierces the wall of the combustion chamber and therefore must also seal the combustion chamber against high pressures and temperatures without deteriorating, over long periods of time and extended use.

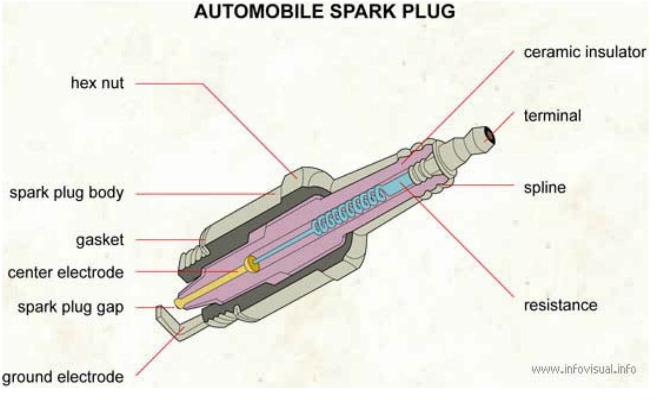

Figure 9.17 Spark plug elements

Brake discs

Porsche was the first automaker to use ceramic brakes on a production car; in 2001, it offered discs made of a novel ceramic composite material to reduce the weight of a special sport model. Nowadays more and more car manufacturers offer ceramic brakes in their production cars. In most professional motorsport series ceramic brakes are very widely used.

Figure 9.18 Audi ceramic brake disc

Piston rings

In short, the following requirements for piston rings can be identified:

- Low friction, for supporting a high power efficiency rate
- Low wear of the ring, for ensuring a long operational lifetime
- Low wear of the cylinder liner, for retaining the desired surface texture of the liner
- Emission suppression, by limiting the flow of engine oil to the combustion chamber
- Good sealing capability and low blow-by for supporting the power efficiency rate
- Good resistance against mechano-thermal fatigue, chemical attacks and hot erosion
- Reliable operation and cost effectiveness for a significantly long time

Ceramic piston rings meet those requirements perfectly.

Figure 9.19 Ceramic piston rings

Engine valves

Ceramic valves have been standard components in racing cars for decades. They offer several advantages in comparison with valves made of metal: thanks to their lower weight, higher revs are possible, and thus improved engine performance. They lower fuel consumption and lessen emissions of carbon dioxide. They are also quieter.

Figure 9.20 Ceramic valve

Pressure sensor

A pressure sensor measures pressure, typically of gases or liquids. Pressure is an expression of the force required to stop a fluid from expanding, and is usually stated in terms of force per unit area. A pressure sensor usually acts as a transducer; it generates a signal as a function of the pressure imposed.

A pressure sensor is just one application of ceramics in electronics – there are far more examples.

Catalytic converters

Catalytic converters have a ceramic grid inside that the exhaust gases flow through. This grid is coated with a metal, usually platinum or palladium, that acts as the catalyst with the exhaust gases.

The grid also has to be heated up to very high temperatures to work correctly. Cats start working at 1200F and are most efficient around 2200-2300F.

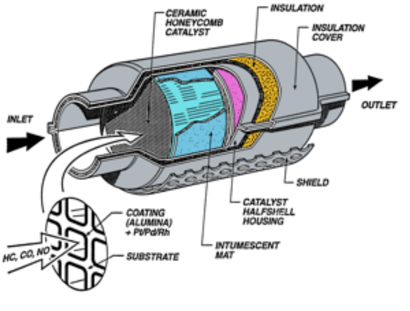

Figure 9.21 Catalytic converter built

Vibration sensors

Piezoceramics have the special ability to turn electrical energy into mechanical force or vice versa. This has led them to be used for sensor and transducer applications in many demanding conditions. The alteration in shape is a natural phase change so the material is left undamaged.

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10 VEHICLE LIFE CYCLE

10.1 Introduction

In chapter number 7 last elements of vehicle life cycle are described. Contemporary knowledge about safety investigation realized in different part of the world is presented. Legal regulations, typical crashworthiness and comparison of real accidents and crash tests are demonstrated. Vehicle repair problems are mentioned. Recycling of different materials as metals, plastics, glass, fluid coming from the body, all systems, elements and parts of vehicle are described. European Directive is cited and on this basis tendencies in car recycling are formulated.

10.2 Vehicle usage

Vehicle from the time it reaches its owner has to fulfil all kinds of tasks to assure safe and comfortable usage. Among others one of the most important is the passenger safety. In a situation of crash impact structures are being deformed to decrease deceleration actuated on the passengers. For not serious crashes cars can be repaired and provide the same amount of safety to its driver as a new one. In the case of serious crash car should not be let to function on the road again and should be recycled.

10.3 Vehicle safety

10.3.1 Legal regulations

The legal regulations are written in terms of minimal vehicle requirements and minimum safety performance requirements for motor vehicles. The difference between the US and European procedures is that US procedures consist mainly of occupant protection and European protocols consists of car design improvement. The American safety regulations are established by Federal Motor Vehicle Safety Standard (FMVSS). The European safety regulations are established by Economic Commission for Europe (ECE). The ECE regulations are directed by the "United Nations Economic Commission for Europe".

Each of them include a detailed set of requirements such as vehicle preparations, speeds, barriers, dummies used, and general crash testing procedures which were previously described. After the tests completed the data acquired from the dummies is downloaded and the forces acting on the neck, head, chest, and lower parts of the body are measured. This is then compared to the injury criteria, a comprehensive guide based on the Cadaver Project, that regulates the minimum requirements needed to pass the legal standard. There are over 500 federal regulations which cover everything from door locks to air brake systems. The European legal regulations are constructed in a different way: the test procedures are written with all details about vehicle and dummies.

Both standards are quite similar; however important differences exist. For example, in frontal impact there are standards like R 12 (protection against steering wheel) or R 16 (safety belt restrain system) but there is not any regulations describing the cars structure like American FMVSS 205 and 219 which are related to windshield design glazing materials. Furthermore, for side impact European norms ECE R 11 and R 95 describe the performance of doors and their locks while American standards like FMVSS 214 (Side Impact Protection) or FMVSS 206 (Seating Systems) consider performance requirements for protection of occupants in side impact crashes, and requirements for seats, attachment assemblies, and installation, to minimize the possibility of failure as a result of forces acting on the seat in vehicle impact. Each vehicle is tested with respect to several legal regulations which must be fulfilled. The following standards are related to safety testing in Europe and US.

Since crash tests have become a legal requirement for each car, two main companies emerged in order to carry out these tests. National Highway Traffic Safety Administration (NHTSA) is an American company which was founded in 1978. Their crash test results are published as New Car Assessment Program (NCAP). NCAP is an international program established to provide information to consumers. NCAP can be separated into European-NCAP and USNCAP; however, a new institution has been established in the US to aid USNCAP in publishing crash test data. This new project is called the Insurance Institute for Highway Safety (IIHS). IIHS is an independent, nonprofits, scientific, and educational organization which was created in order to reduce the possibility of death, injury, and property damage from crashes in the US. The Institute is wholly supported by auto insurer.

Euro NCAP was established in 1997. This program has become significant in safety improvements for new cars. NCAP in general, includes reports of the following tests: frontal impact, side impact, pedestrian impact, and pole test.

The European Enhanced Vehicle-Safety Committee (EEVC) was founded in 1979 and it is an international program on Experimental Safety Vehicles. The EEVC steering committee, consisting of representatives from several European Nations, initiates research work in a number of automotive Working Groups. These groups operate for several years and give advice to the Steering Committee who, in collaboration with other governmental bodies, recommend future action in order to improve vehicles safety [20]. EEVC carries out the crash test but not for approval purposes. The aim of the tests is to provide results for technical research and development in order to improve vehicle standards.

The Federal Motor Vehicle Safety Standards (FMVSS) have been established by The National Highway Traffic Safety Administration. FMVSS is a set of regulations and standards which are the minimum safety performance requirements for motor vehicles or items of motor vehicle equipment valid in the US. These federal standards also include a brief description of other federal consumer information regulations and requirements [21].

ECE R European Regulations relates crash testing with the European climate. These norms are established by United Nations Economic Commission for Europe [22], [23].

Japan and Korea as well as Australia have their own NCAP programs too. It seems that in the time of globalization one standard should exist in the world in the future.

10.3.2 Vehicles crashworthiness

In order to prove the vehicle safety it is necessarily to carry out test which consist on destroying the tested vehicle and simulating a real collision. Those test are called crash test. Crash tests are sets of measurements taken in order to determine the safety of the car. Due to the lack of procedures for ensuring maximum car safety in the past, many fatal accidents occurred. Presently, the constantly increasing number of cars requires that procedures are performed and presented to the consumer to ensure crucial safety regulations are met. During crash tests a vehicle is totally destroyed. Tests simulate real collisions such as frontal impact, side impact, rear impact, and pedestrian impact. The aim of the crash test is to ensure the driver, as well as passengers, with maximum safety in the event of an accident. A crash test dummy is utilized to replicate human behaviour under such circumstances. The dummies manufacture nowadays are called ATD which means Anthropomorphic Testing Device. The ATD are highly advanced devices capable of simulating human behaviour during the impact.

10.3.3 Biomechanics

During each test the dummy injury is always measured and calculated. Relationships between forces and moments which act on the dummy relate to the human body's durability and hardiness, which was determined during the Cadaver Project. The calculation equations for the US and Europe are quite similar due to the resemblance of dummies; however, there are differences in side impact dummy injury criteria due to the fact that in Europe the Euro SID II dummy is used (according to Directive 96/27/EC), while according to American standards SID II is used. The ES 2 is able to measure more parameters then SID II. SID measures acceleration of the ribs, spine and pelvis while the ES 2 measures this plus force, displacement, and acceleration [24].

The endurance and humans body ability to restrain injuries has been investigated for a very long time. For instance, Hippocrates described the head injuries in 400 year b.c. Sir Hugh de Haven is acknowledged as first, who systematically measured the humans body ability of restraining injuries. His investigation started during convalescence caused by aircraft crash. Haven noticed that rigidity of the pilots compartment as well as seatbelts prevent him from hitting the stiff items installed in pilots compartment and consequently save his live. However, the inappropriate construction of the seat along with the seatbelts can cause dangerous scenarios.

J.P. Stapp found that, as long as passenger is correctly protected by means of passive safety equipment i.e. seatbelts, the human body can take huge load without serious injuries. The test consisted in accelerating specially designed for this enterprise carriages to tremendous velocities and then rapid stopping. It has been found that human body can take 30g of acceleration acting within the time of 5 s without death threatening injuries. Furthermore, the acceleration resulting in concussion has been set for 45 g. On the basis of barbeque and cadaver projects, engineers managed to describe several types of injuries for whole body of occupant [25].

AIS scale (**A**bbreviated **I**njury **S**cale) has been established in 1969. AIS determines the scale of humans injuries in terms of medicine and contains 6 following levels:

1. **AIS = 1** $\rightarrow$ minor injuries (bruising, skin sore, broken nose, broken ribs, cut wounds)
2. **AIS = 2** $\rightarrow$ medium injuries (deep wounds, concussion with conscious lose for les the 15 minutes, broken breastbone, extensive ribs braking)
3. **AIS = 3** $\rightarrow$ major injuries (concussion with conscious lose for more than one hour, broken shoulder, broken diaphragm)
4. **AIS =4** $\rightarrow$ heavy injuries (apoplexy with conscious lose for less than 24 hours, spleen injure, stomach injure, the leg loss above the knee)
5. **AIS = 5** $\rightarrow$ critical injuries (apoplexy with conscious lose for more than 24 hours, intestine injure, liver injure, hart injure, spine injure)
6. **AIS =6** $\rightarrow$ deathly injuries (skull brake, chest squash, spine injure above third vertebra)

The AIS scale is also used to determine the cars' velocity during the pedestrian impact. It also apply to the situation in which the passenger compartment intrusion occurs [25]. Based on random 66 test the empirical equation describing relationship between the AIS scale and cars' velocity during the impact with pedestrian, has been set *(equation 10.1)*.

$$AIS = a + bv + cv^2 \tag{10.1}$$

Where:

v is cars' velocity during impact.
a, **b** and **c** are coefficient depends on the type of cars' body type and age of impacted pedestrian.

<u>Head injury criterion (HIC)</u>

The head injury is the most dangerous and most common type injury which occurs during the cars accidents. The reason of head injuries occurrence is head's deceleration. Moreover, the local skull deformation should be taken into account, due to the fact that skull deformation causes the brain damage as well as increase of the brain pressure, which affects the area beyond the local skull displacement. It has been determined that human's head is capable of tolerating very high acceleration as long as duration of acceleration is short. Generally, the shorter time of deceleration acting on the head, the greater deceleration can be tolerated. *Figure 10.1* shows the curve which represents the limits of human's head ability of tolerating the deceleration. This curve was first proposed by Lissner, later was improved by Patrick. Nowadays, chard shown in figure 10.1 is called **W**ayne **S**tate **T**olerance **C**urve (WSTC) [25].

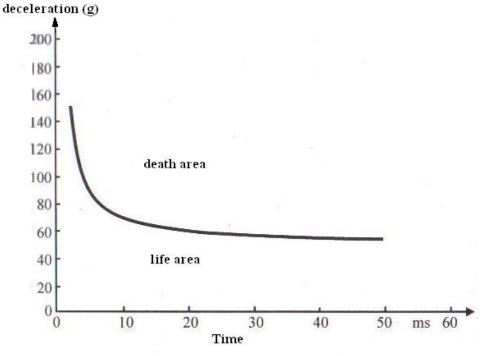

Figure 10.1 WSTC curve [25]

The head injury criterion **HIC** *(equation 10.2)* value is standardized maximum integral value of the head acceleration during the impact [27]. As it was already said, during HIC consideration the duration of the head's deceleration should be taken into account. Therefore, the American directive FMVSS 208 normalizes the duration of the deceleration for average humans size i.e. adult man (represented by 50% ATD), small woman (represented by 5% ATD) and child (represented by Carbi ATDs). There are two normalized time intervals: 15 ms and 36 ms. The time is added to designation "*HIC*" as an index. For example, designation HIC_{36} mans the head injury criteria obtained in time interval of 36 ms .

$$HIC = (t_2 - t_1)\left[\frac{\int_{t_1}^{t_2} A_R dt}{(t_2 - t_1)}\right]^{2.5} \quad (10.2)$$

Where:
(t_2-t_1) is a time interval during which the head is in contact with obstacle.
A_R is acceleration of the head's centre of gravity.

The limits for head injury criteria for different deceleration duration as well as various dummies are as follow:

- HIC_{36}= 1000 $\rightarrow$ man (50%ATD), woman (5%ATD), 6 years old child (Cabri ATD)
- HIC_{36}=900 $\rightarrow$ 3 years old child (Cabri ATD)
- HIC_{36}=660 $\rightarrow$ 12 mouth old child (Cabri ATD)

The head injury criteria HIC for time interval of 15 ms:

- HIC_{15}=700 $\rightarrow$ man (50%ATD), woman (5%ATD), 6 years old child (Cabri ATD)
- HIC_{15}=570 $\rightarrow$ 3 years old child (Cabri ATD)
- HIC_{15}=390 $\rightarrow$ 12 mouth old child (Cabri ATD)

Neck injury criterion (NIC)

The neck injury occurs, most often, as a consequence of the rear impact. However, frontal as well as side impact can cause injury of the neck. During the frontal impact, neck is not submitted to large stresses due to the fact that the jaw rests on the chest, therefore the head's moment forward is limited (*see Figure 10.2 a*). The backward movement is more dangerous as far as neck injury is concern, due to the fact that this kind of movement produce greater stress (*see Figure 10.2 b*). However, nowadays almost every car is equip in head restrain which significantly reduces probability of fatality, but it impossible to completely avoid the risk of neck injury during any type of impact [26].

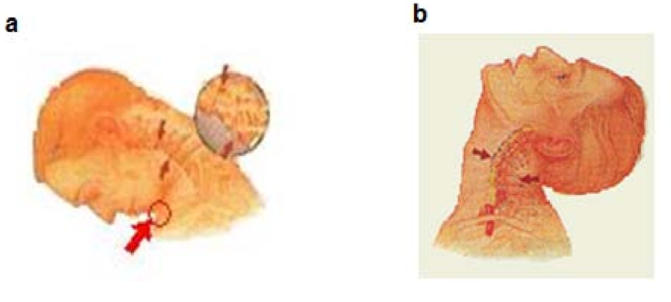

Figure 10.2 Neck tension during: a) forword movemetn, b) backard movement [26]

The estimation of the neck injury that occurs as a consequence of impact is described as neck injure criterion. According to ECE regulation, EuroNCAP protocol and FMVSS standards the neck injury criteria is determined by using the axial compression force, the axial tensile force and the shear forces at the head and/or neck as well as the time interval during which human is submitted to those forces. Mathematically NIC is defined in *equation 10.3* [25]:

$$NIC = 0{,}2a_{relattive} + v_{relative}^{2} \tag{10.3}$$

Where:

$a_{relative}$ and **$v_{relative}$** are the relative values of acceleration and velocity between the chest vertebra (T_1) and the neck vertebra (C_1).

The **0,2** constant represents the length of the neck part of spine.

$$a_{relative} = a_x^{T_1} - a_x^{head} \tag{10.4}$$

$$v_{relative} = \int a_{relative} \tag{10.5}$$

Where:

$a_x^{T_1}$ is a acceleration of the first chest vertebra in x-direction in m/s^2

a_x^{head} is the head's acceleration at the height of centre of gravity in x-direction in m/s^2

In order to determine complicated neck stress during the accident the **Nji (Normalized neck injury criterion**) has been employed. Considering the Nji neck injuries are calculated by using the axial compression force, the axial tensile force, shearing forces at the intersection of the head with the neck in kN, duration of all those forces in milliseconds. The neck bending moment criterion is determined by the bending moment in kN, around a lateral axis at the transition from the head to the neck [27].

Nji includes four neck criteria (neck injury predictor) [27] :

- N_{TE} – tension –extension
- N_{TF} – tension – flexion
- N_{CE} – compression – extension
- N_{CF} – compression – flexion

Mathematically Nji is exhibited in: [25], [27].

$$Nji = \frac{F_z}{F_{\mathrm{int}}} + \frac{M_Y}{M_{\mathrm{int}}} \tag{10.6}$$

Where:

F_Z, M_Y are axial load (compression- tension) at the intersection of the head and neck.

$F_{\mathrm{int}}, M_{\mathrm{int}}$ are the critical values of the force and moment.

It has been determined that the value of Nji =1,0 corresponds to the 15% possibility of serious injury, Nji= 1,4 corresponds to 30% possibility of serious injury. Furthermore, the relationship between AIS scale and Nji has been established. Those relationships are shown in following equations [25]:

$$P_{(AIS\geq 2)} = \frac{1}{1+e^{2,054-1,195Nji}}$$

$$P_{(AIS\geq 3)} = \frac{1}{1+e^{3,227-1,969Nji}}$$

$$P_{(AIS\geq 4)} = \frac{1}{1+e^{2,693-1,195Nji}} \tag{10.9}$$

$$P_{(AIS\geq 5)} = \frac{1}{1+e^{1,817-1,195Nji}} \tag{10.10}$$

Using the equations mentioned above it is possible to estimate the risk of injury. For example, assuming that Nji=1,0 it is easy to estimate the risk of the injury for 22%.

Chest injury criterion

The tests shown that human's chest is cable of tolerating loads of 8kN as long as the load is distributed uniformly on shoulders and on the chest. Moreover, the acceptable chest's deceleration is equal to 60 g in time shorter than 3 milliseconds and acceptable chest deformation has been established on 76mm [25].

Based on those test and limits the chest injury criterion has been introduced. Considering chest injury criteria it is crucial to determine the chest deflection and duration of deflection process. This phenomenon is defined as Viscous Criterion (VC). The basis of the criterion is the velocity (V) of the chest compression (C). It appears that the greatest allowed value of deflection occurs for time less than 3 m/s. Above displacement velocity of 25 m/s, instant muscular tissue injury and consequently life threatening injures occurs.

Another chest injury criterion is Thorax Compression Criterion (TCC). TCC is the criterion of the compression of the thorax between the sternum and the spine. TCC can be calculated by using the absolute value of the thorax compression and extension expressed in millimetres [27]. According to this criterion the chest deflection should not exceed 50 millimetres, while Thorax Performance Criterion (THPC) establishes the limit for 75 millimetres [25]. The THPC is the criterion for chest strain with side impact. Calculation of this criterion includes the Rib Deflection Criterion as well as Viscous Criterion [25].

Nowadays, most common criterion is Combined Thoracic Index (CTI) [25]. CTI represents the chests injury during frontal impact [27]. The calculation of this criterion is curried out according to *equation 10.11*.

$$CTI = \frac{A_{\max}}{A_{\mathrm{int}}} + \frac{D_{\max}}{D_{\mathrm{int}}} \tag{10.11}$$

Where:

$A_{\max}, D_{\max}$ are maximum values of the acceleration and the chest deflection

$A_{\mathrm{int}}, D_{\mathrm{int}}$ are the greatest allowed values of the acceleration and the chest deflection

Like in case of Nji, the relationship between CTI and AIS scale. This relationship are described by *equation 10.12* and *equation 10.13.*

$$P_{(AIS \geq 3)} = \frac{1}{1 + e^{7,529 - 6,431 \cdot CTI}} \tag{10.12}$$

$$P_{(AIS \geq 3)} = \frac{1}{1 + e^{7,529 - 6,431 \cdot CTI}} \tag{10.13}$$

Criteria for the Lower Extremities

The lower extremities criteria includes the **A**bdominal **P**eak **F**orce (APF), **P**ubic **S**ymphysis **P**eak **F**orce (PSPF), **F**emur **P**erformance **C**riterion (FPC) **F**emur **F**orce **C**riterion (FFC), **T**ibia **I**ndex (TI), **T**ibia **C**ompression **F**orce **C**riterion (TCFC).

The **APF** represents the greatest value of a sum of three forces acting on the abdomen, measured during the side impact (*equation 10.14*). Generally, APF is maximum side abdominal strain criterion [27].

$$APF = \max\left|F_{yFront} + F_{yMiddle} + F_{y\mathrm{Re}ar}\right| \tag{10.14}$$

The **PSPF** represents the pelvic strain during the side impact. PSPF can be determined by the maximum pelvis's strain in kN.

According to ECE R94 directive as well as EuroNCAP protocol the **FFC** is defined as criterion of the force which acts on the femur. FFC is determined by the compression stress (expressed in kN) transmitted axially on each femur of the ATD dummy in exact amount of time (expressed in milliseconds). The only difference between the FFC described in the EuroNCAP protocol and in the ECE directive is a performance requirements. The performance requirement are exhibited as a chards in shown *Figure 10.4* [27]

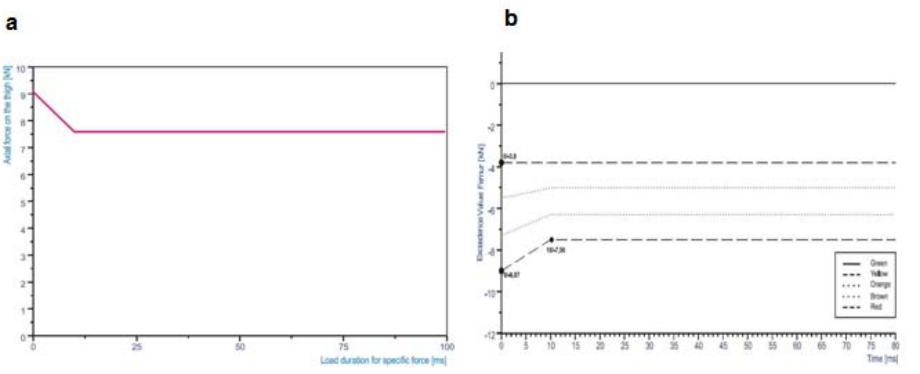

Figure 10.3 Performance requirements, a) for ECE, b) for EuroNCAP [27]

The **TCFC** is a criterion for the tibia strain and is the force acting axially on both tibias of the ATD dummy. According to the TCFC the maximum allowed axial force acting tibias should not exceed 8kN.

The **TI** includes the lower leg area and involves the bending moments around the x and y axis as well as force which acts in accordance with z axis at the top or bottom end of the tibia. The tibia index is calculated according to (*equation 10.15)* [25].

$$TI = \frac{F_Z}{(F_Z)_{\text{int}}} + \frac{M_G}{(M_G)_{\text{int}}} \tag{10.15}$$

Where:
F_Z is a compressive force acting in a *Z* direction.
$(F_Z)_{int}$ is a critical compressive force acting in a *Z* direction. This critical force equals 35,9kN [27].

The bending moments around the Y and X axis are shown below:

$$M_G = \sqrt{(M_X)^2 + (M_Y)^2}; M_X, M_Y \tag{10.16}$$

$(M_G)_{int}$ is a critical bending moment and its value amounted to 225 Nm

It has been established that allowed value of tibia index is amounted to TI $\leq$ 1,3

10.3.4 Vehicle crashworthiness determination

Frontal impact

The frontal impact is a type of accident which causes the most fatalities and serious injuries. Tests during which the vehicles were strike against the rigid wall with speed of 50 km/h shown that, passenger compartments' deceleration can reach up to 28 g and vehicles centre of mass can be displaced for 60 cm.

During the frontal impact test the car is pushed into a solid barrier. Frontal hit can be separated into impact with offset and hit in full-width wall. In this case most injuries occur due to rapid vehicle deceleration.

During this test forces acting on dummy's head, chest and legs are measured. This test provides information about dummies' injuries as well as such features as seat belt and air bag efficiency; however, vehicle damage is not assessed. Such features as car speed, legal regulations, and measurement techniques are also dependent on the country in which the test is carried out [28].

The frontal offset impact test is always on the driver side due to the fact that there is the greatest risk of injury from the pedals as well as steering wheel. Moreover, accidents simulated by this test (car-to-car) are the most common kind of collisions; hence the barrier is made of deformable aluminium due to the fact that there is no car which has the same toughness as a solid wall. This test in general provides information about a car's front ability to absorb the impact energy [28]. Moreover, the frontal impact is carried out in order to assess and reduce passengers compartment intrusion and reduce forces of inertia acting on passengers who wears the seatbelts [23]. The frontal impact is shown in Figure 10.5 [30].

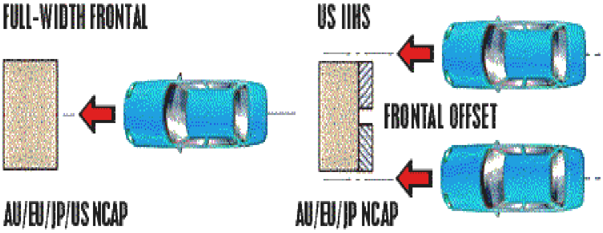

Figure 10.4 Frontal Impact [30]

According to European norms (EU Directive 96/79 EC), and Euro-NCAP testing procedures, during the frontal impact the tested vehicle strikes into a deformable barrier. This barrier is made of deformable aluminium and its design resembles a piece of honeycomb. The bumper is attached to the Main Honeycomb Block in order to simulate the impact of two vehicles. It is also the reason why the barrier is made of deformable aluminium. EU Directive 96/97 EC only monitors dummy injuries and steering column movement while Euro- NCAP monitors this in addition to foot well intrusion and structural demerits.

According to Directive 74/297EC the vehicle is pushed against a rigid wall. Standards which describe this are ECER12, R33, and R34. The vehicle speed is 48.3 km/h (30 mph). This type of frontal impact is carried out in order to determine steering column movement, passenger compartment, door opening ability, and fuel spillage.

The tested vehicle is pushed into the barrier with a speed equal to 64km/h (40mph) ± 1km/h according to the Euro NCAP testing protocol and with a speed equal to 56 km/h (35 mph) according to EU Directive 96/79 EC . The speed will be measured as near as possible to the point of impact. The impact point will be moved with 40%±20mm offset of the vehicles. The vehicle will strike on the driver side.

Barrier type Euro NCAP Barrier Face	Aluminium Honeycomb Block
Ground Height	200 mm
Barrier type 74/297 EC	Rigid wall
Euro-NCAP Speed	64 km/h (40 mph) ± 1km/h
74/297 EC Speed	48.3 km/h (30 mph)
Directive 96/79 EC speed	56 km/h (35 mph)
Target	Vehicles front with offset 40%±20mm
74/297 EC Target	Full front of the vehicle

Euro NCAP deformable barrier is designed with the following dimensions:

Main Honeycomb Block

Material:	Aluminium 3003 (BS 1470)
Height	650 mm (in direction of honeycomb ribbon (foil) axis)
Width:	1000 mm
Depth:	450 mm (in direction of honeycomb cell axes)
	All above dimensions ± 2.5 mm
Crush Strength:	0.342MPa +0% -10%
Bumper Element	
Material:	Aluminum 3003 (BS 1470)
Height:	330 mm (in direction of honeycomb ribbon axis)
Width:	1000 mm
Depth:	90mm (in direction of honeycomb cell axes)
	All above dimensions ± 2.5 mm
Crush Strength:	1.711MPa +0% -10%
Backing Sweet	
Material:	Aluminum 5251/5052 (BS 1470)
Height:	800 mm
Width:	1000 mm
	All above dimensions ± 2.5 mm

Thickness: 2.0 ± 0.1mm

The following injuries are calculated according to European 96/79/EC and American FMVSS 201 standards [27].

Head Injury Criteria (HIC) is calculated form *equation 10.17* :

Neck

$$(M_Y)_i = M_Y - fxd \qquad (10.17)$$

Where:

M_Y is a Bending Moment

fxd is a Shear Force

Chest

Viscous criteria:

$$1{,}3V_{(t)} \cdot C_{(t)} \tag{10.17}$$

$$C_{(t)} = \frac{D_{(t)}}{0{,}229} \tag{10.18}$$

$D_{(t)}$ is a chest deflection
$C_{(t)}$ is a compression related to the chest deflection
Velocity of the deflection:

$$V_{(t)} = \frac{8\lfloor D_{(t+1)} - D_{(t-1)} \rfloor - \lfloor D_{(t+2)} - D_{(t-2)} \rfloor}{12\delta t} \tag{10.19}$$

δt is a time interval between successive digital samples of $D_{(t)}$

Knees. The knees injury criteria are measured by determining the value of the displacement.

According to both American FMVSS 208 and European 96/79/EC norms the maximum dummy's injuries are as follows:

Head. Acceleration should be less than 80 g in time 3 ms
Neck. Flexion bending moment should be less than 190 Nm
Extension bending moment should be less than 57 Nm
Axial tension 3300 N
Chest. Acceleration should be less than 60 g.
Viscous criteria (V*C) = 1m/s
Chest deflection 50 mm
Lower extremity. Femur loads 9,07 kN
Axial compression of tibia 8 kN
Displacement of knee slider 15 mm

Side impact

Side impact (see *figure 10.6*) ranks second after frontal impact as the most common type of collision. In side impact the tested vehicle is hit by a trolley. The trolley is made of deformable aluminium, and it hits on the driver side. The use of the trolley opposed to another car is equally effective while more financially advantageous. During this test Euro SID II (ES 2) and SID (Side Impact Dummy) are used due to their greater number of measurement sensors. Sometimes another dummy is placed in the rear seat in order to determine the likelihood of driver and passenger injury. In general this test evaluates the following factors: driver and passenger injury, head protection, and structural performance. The measured injuries are neck, chest, abdomen, pelvis, and femur.

Another type of side impact testing is car on car hit. However, the main inconvenience of this solution is that the tested cars must be precisely verified.

The most significant parameter during the side impact testing is the rapid vertical acceleration of the struck vehicle at the moment of contact.

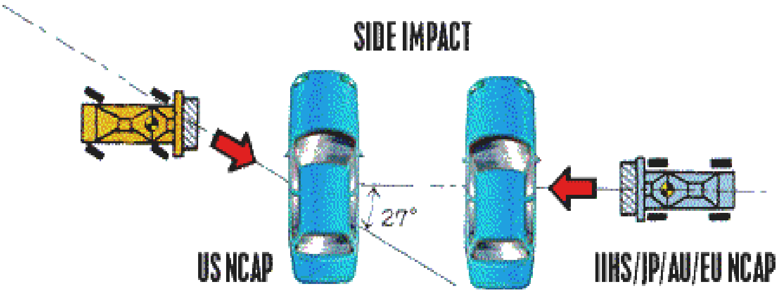

Figure 10.5 Side impact [30]

According to European (EU 96/27 EC) norms, and Euro NCAP testing protocol, during the side impact the tested vehicle is struck by a Moving Deformable Barrier (MDB) in the driver side of the car. The barrier is made of deformable aluminium in a Honeycomb Block design. As in the case of frontal impact the bumper is attached to the main barrier. The precise dimensions of the barrier are shown on *Figure 10.7*. These procedures are identical except that Euro NCAP adds data for door opening ability in addition to the dummy injury monitored by both. In order to eliminate secondary impact between the vehicle and barrier, the trolley braking system must be installed

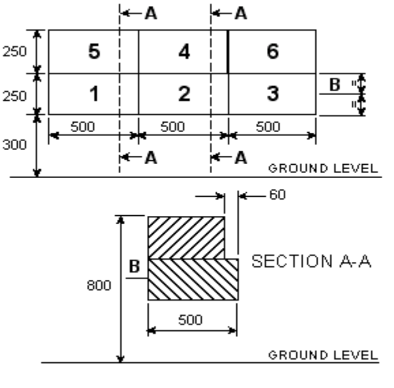

Figure 10.6 design of side impact moving deformable barier [31]

The barrier is installed on the trolley which is then pushed into the tested vehicle with a speed equal to 50km/h ± 1km/h. The speed will be measured as near as possible to the point of impact [1].

Barrier type	MDB Aluminum Honeycomb Block (m=950 kg)
Barrier Face	
Ground Height	300 mm
Speed	50km/h ± 1km/h (31,7 mph)
Target	Vehicles R-point
Impact angle	90º±3 º

<u>Pole test</u>

Pole test is another type of side impact. In real conditions this collision occurs either when one car runs into the side of another or runs into a stationary object such as a tree, lamppost or bridge support. These kinds of collisions are the cause of serious injuries. Pole tests provide the information relevant to heat protection devices like side airbags. Without these kinds of devices and appropriate car design, probability of death or serious injury is extremely high.

During pole tests the car is pushed into a stationary column which creates the main difference between this test and side impact.

The whole energy of the impact is concentrated on a much smaller area which produces a larger crash depth and consequently greater danger to the driver [31].

This test is directed by Euro NCAP testing protocol. The largest difference between this test and side impact is that in the overall rankings only dummy head injuries are considered. The limited lower value of head injury is the same as in side impact. During the pole impact test, the tested vehicle is placed on a carrier which is pushed into a stationary column. In order to reduce friction between the tyres and carrier, each tyre of the tested vehicle is placed on two PTEF sheets. In order to avoid vehicle movement before impact it can be fixed to the carrier. The vehicle will be released 5 m before the impact [31].

The rigid pole is made of metal. It should be installed vertically no more than 102 mm above the lowest point of the tyres and 100 mm above the highest point of the vehicle's roof. The pole is attached to the barrier or other surface so that the tested vehicle will not hit the barrier. The pole diameter is 254 ±3mm. The carrier with the vehicle is pushed into the pole with a speed equal to 29±0.5km/h. The speed will be measured as near as possible to the point of impact; however, the carrier will achieve its appropriate speed of at least 10 m before the pole.

The side pole impact is shown in Figure 10.8. The elaboration of the results consists only on determination of HIC.

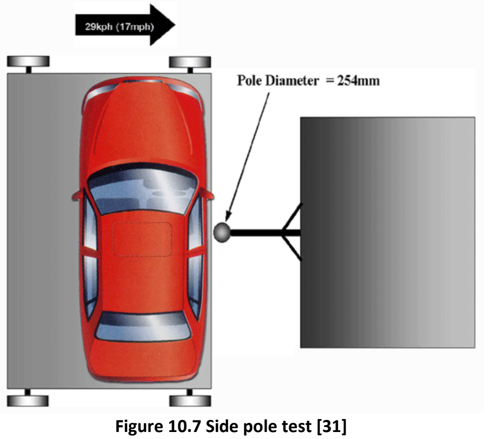

Figure 10.7 Side pole test [31]

<u>Pedestrian impact</u>

Pedestrian impact has been established in order to determine the possibility of injuries to pedestrians as well as the vehicle occupants which occur in these types of collisions.

Pedestrian impact includes both adult as well as children's impact against the exterior of the vehicle. Different parts of the dummies' legs and heads are thrown against the tested car. The measurement devices installed in the dummy parts measures the severity of the impact. Results are used to rate the car. In order to provide the test with maximum relativity the tested parts of the dummy are made of aluminium and covered by polypropylene which simulates the skin.

The pedestrian tests include eighteen points of impact due to the fact that each of the heads is tested in six different locations and each limb is tested in three locations as it is impossible to forecast human behaviour during the impact. This is done in order to provide maximum information about injuries which occur after impact in different parts of the car.

This test consists of launching various child and adult dummy parts at the front of the vehicle. This simulates an accident in which the car runs over the pedestrian with an approximate speed of 40kph (25mph).

The Leg Form Test

The legs shall be aimed toward the bumper with the lower bumper less than 500 mm above the ground at the impact point. All impact points will be a minimum of 66 mm inside the bumper corners and will be a minimum of 132 mm apart. The measurement equipment as well as the testing conditions are shown in the *Figure 10.9 a, b* [32].

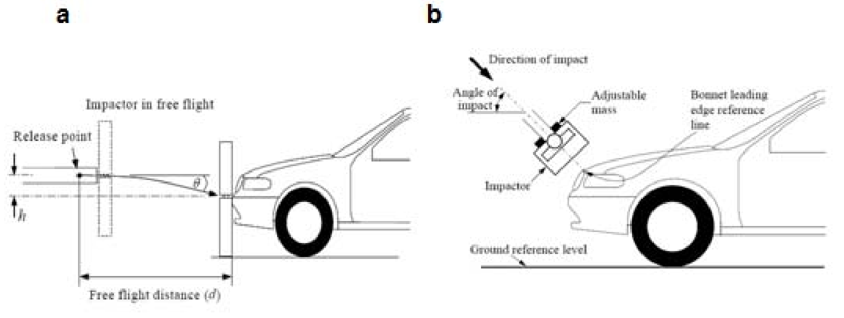

Figure 10.8 Legform test, a) lower legform, b) upper legform [32]

At the point of first contact:

v = impactor velocity (11.1m/s ±0.2m/s)
θ = direction of impact (0° ±2°)
d = free flight distance
h = height increase

At the time of the first contact, the centre line of the legform impactor will be within ±10 mm of the selected impact point. The legform impactor must not contact the ground or any object not part of the vehicle during the test [32].

Upper Legform Aimed Toward the Bonnet Landing Edge Test

The total weight of the upper leg form should be 9.5kg±0.1kg. The legform is thrown against the vehicle's bonnet. Velocity of impact as well as the impact angle and total kinetic energy of the impactor will be calculated from the bonnet landing edge height and bumper leader [32].

Head Form Testing

The projected points for the adult headform will be at least 82.5 mm inside the Side Reference Lines and at least 165 mm apart. The projected points for the child headform will be at least 65 mm inside the Side Reference Lines and at least 130 mm apart. These are minimum distances which will be set with flexible tape held tautly along the outer surface of the vehicle.

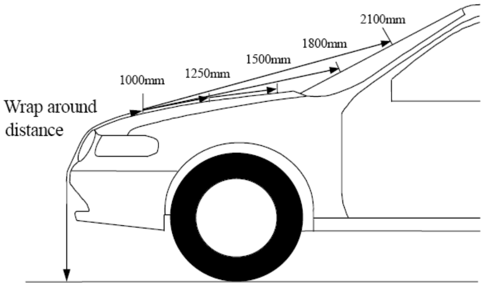

Figure 10.9 Headform Test [32]

Figure 10.10 shows the boundary lines in which the dummy's head should hit. The adult dummy's head should hit between wraps 1500 mm and 2100 mm. The child headform should hit between 1000 mm and 1500 mm. The headforms should be thrown against the tested vehicle at an angle of 65°±2° to the Ground Reference Level for adult headforms. For child headforms, the angle is 50° ±2° to the Ground Reference Level. The impactor will be shot with the speed equal to11.1 ±0.2m/s The gravity effect will be considered when the angle is obtained from the measurement taken from the time of first contact.

Table 10.1 Headform values

Headforms		
Value	**Adult headform**	**Child headform**
V	11.1 ±0.2m/s	11.1 ±0.2m/s
θ	65° ±2	50° ±2°

The distance of the pedestrian rebound with respect to velocity of collision is described by means of many empirical equations. Base to those equations is shown below *(Equation 10.20)* [25]

$$s = a \cdot v^2 + b \cdot v + c \qquad (10.20)$$

where:
s is a distance of pedestrian rebound
v is a collision velocity

It is obvious that equation will have different constants a, b, c for different speed, cars body type and the age of pedestrian (child or adult). All equations can replaced by the average (*equation 10.21)*

$$s = 0{,}0491 \cdot v^2 + 0{,}2915 \cdot v + 0{,}2386 \qquad (10.21)$$

According to the Euro NCAP, the lower limit of dummy injury in pedestrian impact should not exceed following values:

Headform injury criteria HIC = 1350
Upper legform
Bending moment = 380 Nm
Sum of all forces = 6 kN
Legform
Tibia deceleration = 200g
Knee shear displacement 7mm
Knee bending angle 20º

Rollover crash testing

The rollover is a type of accident which endangers mostly the vehicles with high gravity centre i.e. LTVs (light truck vehicle). During the testing the most problematical meter is lack of appropriate ATD dummy. The dummy has to be calibrated specifically to each type of test. Rollover causes mainly head injuries (see *Figure 10.10*), therefore the Euro SID dummy is used due to the fact that this model is capable of measuring HIC (head injury criteria) with very high accuracy. The exception is test curried out according to FMVSS 208 (occupant protection) where the male Hybrid III 50% ATD is used [19], [34].

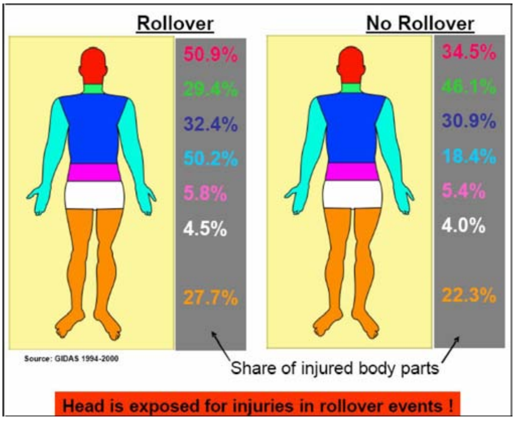

Figure 10.10 Injuries in roll / no roll events [37]

Apart from stiff passenger compartment construction, occupants protection in case of rollover event is ensured by seat belts and modified airbag curtain. The modification of the airbag curtain consist on ensuring that airbag stays inflated for up to 6 seconds, which is crucial in terms of head protection during long-lasting rollover event, while regular side airbags inflates and deflates very rapidly.

Moreover, the airbag curtain prevents occupant from ejecting form the car during rollover event [15].

The rollover accident can be separated on several scenarios shown on *Figure 10.12*.

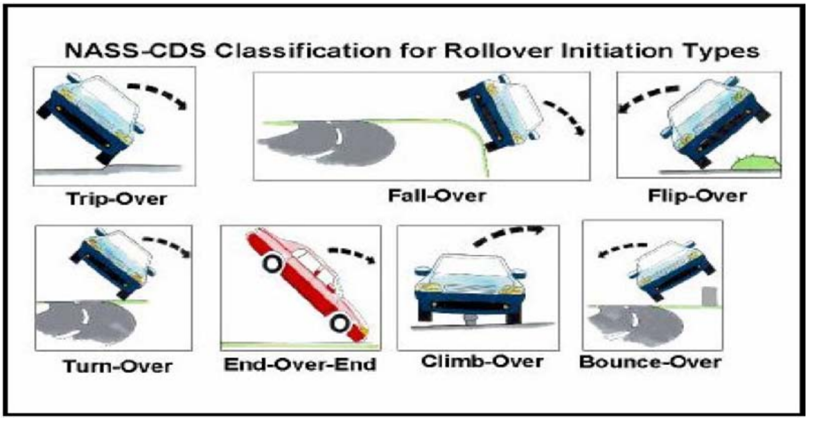

Figure 10.11 NASS classification for rollover initiation types [33]

According to **GIDAS** (**G**erman **I**n-Depth **D**ata **A**ccident **S**tudy) the most common type of rollover is the soil trip , which consist on lateral approaching to the surface with lower hardness (see *figure 10.12*). During the soil trip rollover test, the tested vehicle is placed on the flying floor. The flying floor is accelerated thwarts the soil area on which is rapidly decelerated. In consequence, rollover event occurs. This kind of rollover is exhibited in *Figure 10.13*

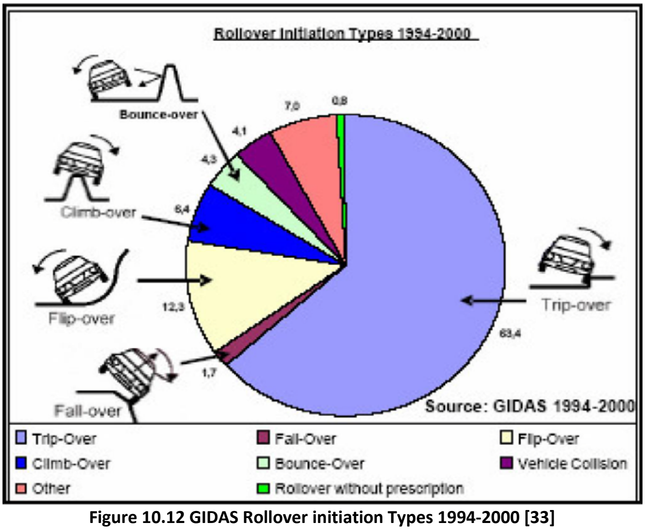

Figure 10.12 GIDAS Rollover initiation Types 1994-2000 [33]

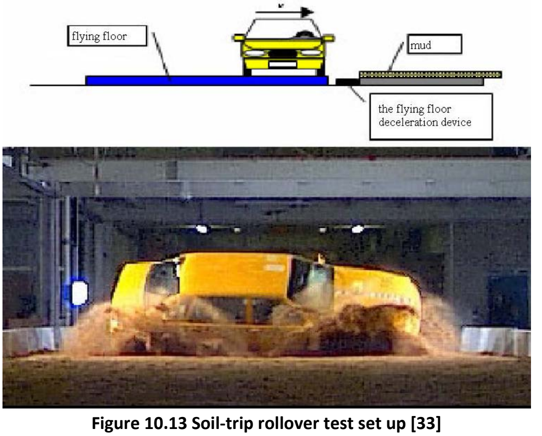

Figure 10.13 Soil-trip rollover test set up [33]

The **embankment** (see *Figure 10.14*) rollover has been classified as second in terms of frequency of occurrence. The embankment rollover test consist on pushing the tested car thwarts inclined planed. In consequence the rollover event is provoked. This test include [33]

- A slow occupant movement, which requested a relatively late deployment time.
- The lateral acceleration is in a lower range.
- The roll conditions of the test car vary with the adjusted velocity, approach angle, slope angle and steering actuation.

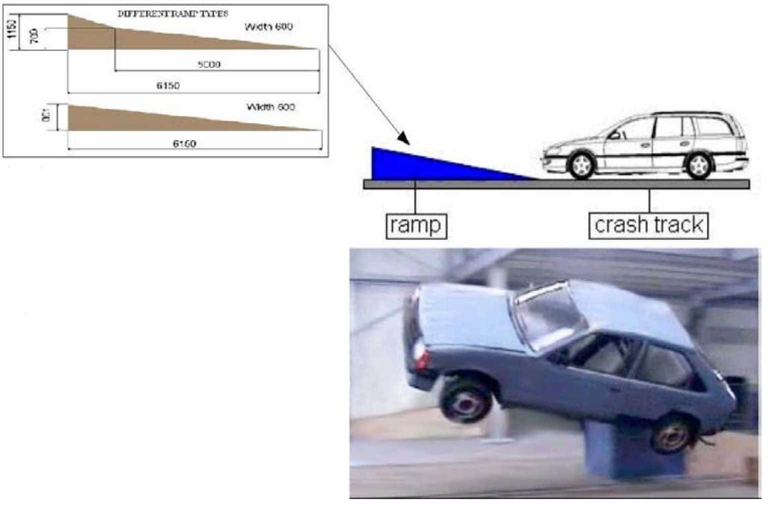

Figure 10.14 Ramp rollover test set up and different ramp types [33]

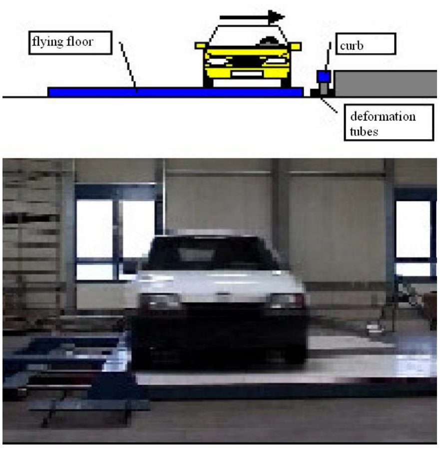

Figure 10.15 Curb-trip rollover test set up [33]

The **Curb – Trip** rollover (see *Figure 10.15*) include a lateral movement only. During this test, the car is installed on the flying floor and impacts laterally against a curb whit the wheel rims. The high of the curb is depended on the wheel size. The flying floor is decelerated by means of decelerated by deformation tubes without affecting the car movement. This test includes [34]:

- A rapid occupant movement, which requests an early deployment time.
- The lateral acceleration is in a higher range.
- The roll conditions of the test car vary with the adjusted velocity and used curb high.

The rollover test curried out according to FMVSS 208 directive is used in order to analyze the behaviour of the vehicle construction under rollover loads. A great emphasis is given on the passenger compartment intrusion.

During FMVSS 208 rollover test the car is placed on the platform inclined under 23 degrees capable of moving laterally. The platform is accelerated to the velocity of 48,4 km/h and rapidly decelerated by means of deformable tubes which not affect the vehicle movement. [33]

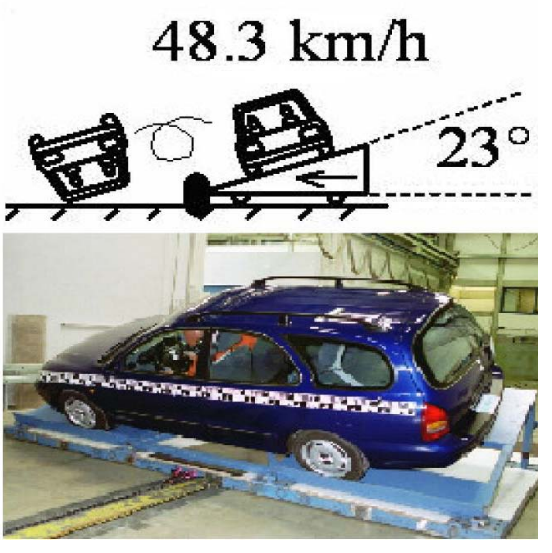

Figure 10.16 FMVSS 208 rollover test set up [33]

10.3.5 Crash test reliability

The stresses acting upon particular member of the body design will be expressed with aid of numerical simulation. The discrete model was created based on Audi A3 and is depicted in *figure 10.18*. the markers (pink areas on the body) ire introduced in order to visualize deformation of the construction. At this point the PAM-CRASH software is used due to the fact that this software represents very accurate results. Moreover the results obtained in this software clearly indicates the behavior of energy consuming elements. First, the model was tested in terms of nonsymmetrical frontal impact with deformable barrier with speed respecting the EuroNCAP standard (64 km/h).

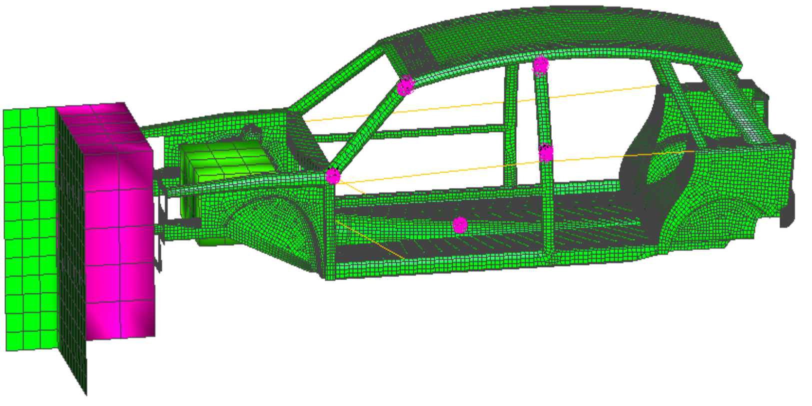

Figure 10.17 The discrete model was created based on Audi A3

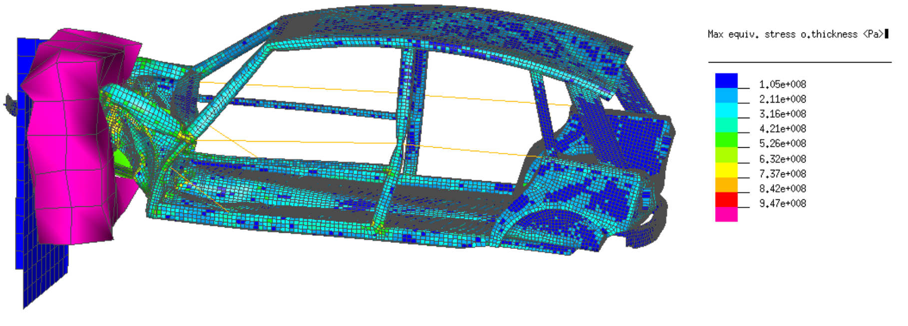

Figure 10.18 The intensity of load transfer resulting from frontal impact with deformable barrier

The results obtained during this simulation indicates the maximum deceleration acting within a time of 80ms inside the passenger compartment is 57g. The A pillar was dislocated for 180 mm. the results are shown in *figure 7.19.*

10.3.6 Real accidents vs. crash tests

Comparing the crash rest of a VW Golf II conducted in DEKRA in Neumünster Germany with this same model of car after a frontal nonsymmetrical crash the similarity of the results is clearly visible. As it can be seen in *figure 10.20.* In both cases the passenger compartment significantly intruded. The A pillar of a car tested by DEKRA was displaced for 400 mm. In case of real accident the displacement of A pillar reached 300 mm. In contras the displacement in modern cars is 20 – 30 mm. This condition was farther confirm with aid of a numerical simulation. This allows to exhibit the dependence between displacements of particular member of a design within particular duration. COSMOS software was used for the simulation of accident. As it can be seen the numerical model and results of the crash tests shows satisfactory resembles

Figure 10.19 Comparison of crash test with real accident

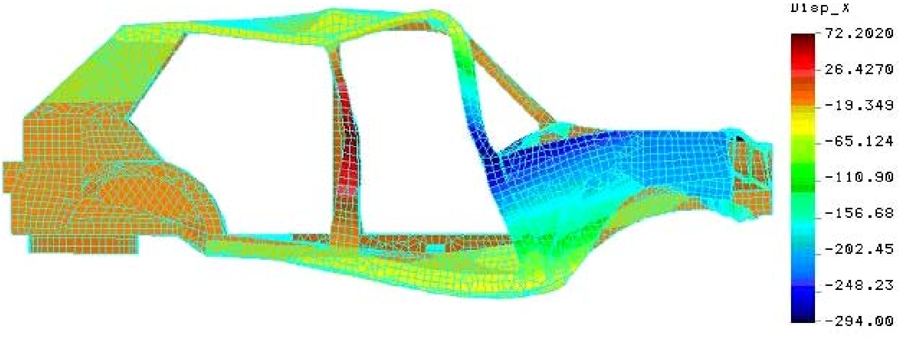

Figure 10.20 longitudinal displacement resulting from the frontal impact with deformable barrier

In the case of numerical simulation the displacement of A pillar reached 290 mm. this occurred as a result of maximum load of 600 – 750 MPa. It is depicted in *figure 10.22*

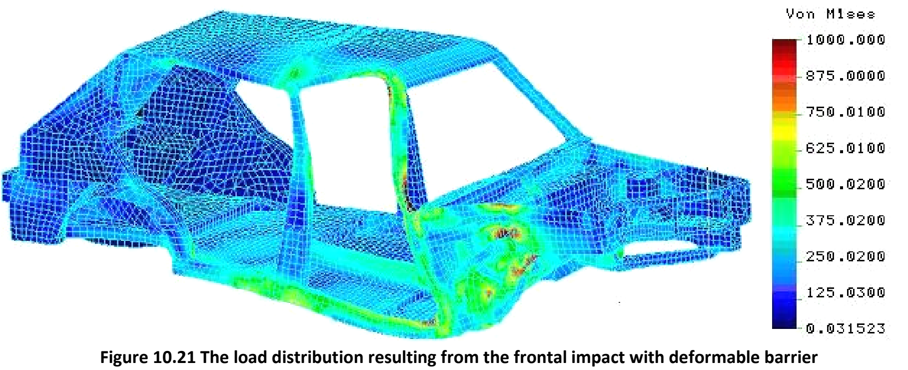

Figure 10.21 The load distribution resulting from the frontal impact with deformable barrier

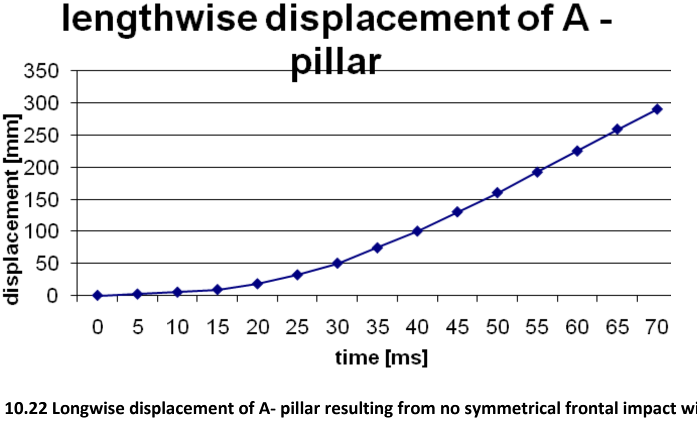

Figure 10.22 Longwise displacement of A- pillar resulting from no symmetrical frontal impact with deformable barrier

10.4 Vehicle repair

During vehicle life state of its elements is constantly changing due to wear or damage. One of the main factors influencing element state are:

- Road conditions – road quality dependent;
- Climate conditions – rain intensity, wind power and temperature dependent;
- Usage intensity – driver dependent.

Because of the possible wear of vehicle elements each car has to go through manufacturer specified checkup. Nowadays the mileage between checkup is rising due to more advanced technology of part manufacturing and better suited element strength to its real working conditions – as an example new Fiat Bravo should be serviced every 30 000 km. Standard service is conducted by checking all mechanical elements by sight and by reading reports from the OBD system (On-board-diagnostics). The OBD is very useful as some failures may not be seen but thanks to the various sensors placed in the car – all signs of wear or damage can be identified.

Apart from the auto maker recommended service there is yearly checkup required by the law – these may vary depending on the country. Main elements of this checkup are listed below:

- Tyre condition,
- Functioning and positioning of all external lighting,
- Brake system condition, efficiency and even brake force actuation,

- Steering system condition and slack in the system,
- Suspension system condition,
- Wheels alignment – all angles connected with wheels: toe-in, toe-out, camber, etc.,
- Bodywork and chassis condition,
- Exhaust system condition,
- Exhaust gas emissions checkup,

After all necessary test were passed by the vehicle it is allowed to enter the public roads.

Special inspection is conducted with a vehicle which was repaired after a crash. The most important systems that are thoroughly inspected are: suspension geometry, steering system geometry, shock absorbers, brake system, engine. A special stand is used for controlling the geometry points of the chassis which was deformed in the crash. Sensors are scanning the vehicle and comparing reference points positions with database provided by car manufacturer.

10.5 Recycling

From the moment it was invented automobile gained in popularity. Nowadays annual world production exceeds 60 million of cars and commercial vehicles. A significant part of them will replace old machines which are not suitable for operation anymore. In 2002 in 15 European Union countries it were 12 million of vehicles, in USA 10 million and in Japan 9 million. In mentioned situation necessary is proper segregation and secondary usage of the materials of which cars comprise. This process prevent the planet from being covered with enormous amount of waste, sometimes very aggressive and toxic like acids from batteries or lubricants, hydraulic fluids and fuel. Moreover recycling can provide lower costs for production. Retrieved materials are already processed so there is no or little need of further treatment. The processes of mining and ore foundry are limited or eliminated. Recycling of some plastics requires only cleaning and shredding for being again treated as production materials. However during the recycle process inevitable is precise segregation and purification of many materials. [37, 38]

10.5.1 Materials

The composition of a typical car has changed substantially in recent years. Ferrous metal content has decreased significantly as lighter, more fuel-efficient materials such aluminum or plastics are incorporated into vehicle design. [35, 38, 39]

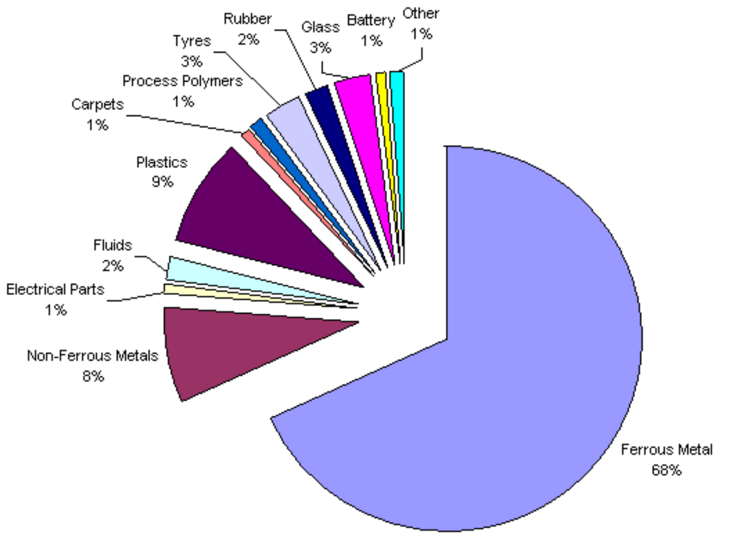

Figure 10.23 Content of different material in a car (by weight)

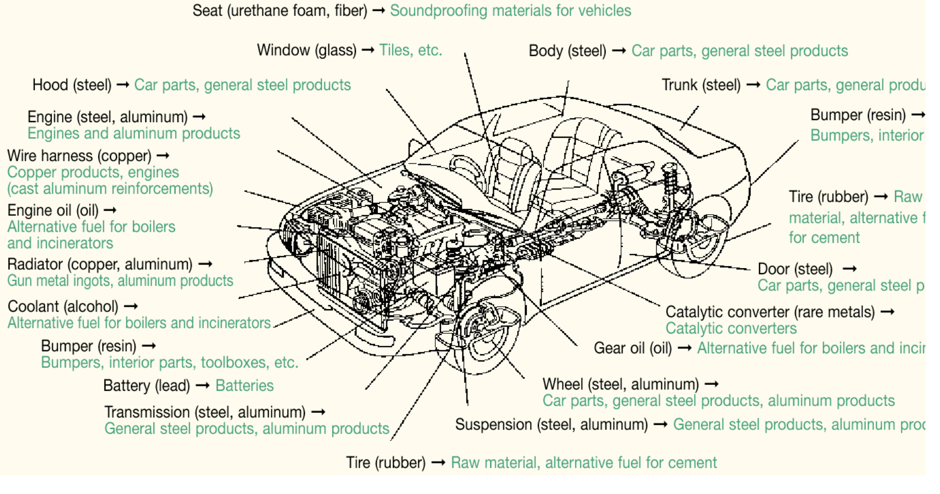

Figure 10.24 Example of parts being recycled from ELV

Metals

Approximately 68% by weight of the average car is metal, most of which is comprised of sheet steel. The overall metal content of cars has declined rapidly during the past 20 years accompanied by an increase in the proportion of non-ferrous metals used in their manufacture, such as aluminum and magnesium. Currently about 98% of the metals in a car are recycled. These metals are recovered by the vehicle shredding industry and subsequently utilized by the steel industry and ironworks plants. [35, 38, 39]

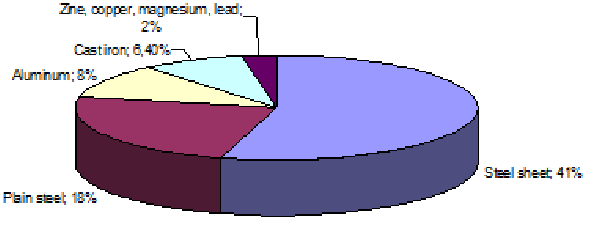

Figure 10.25 Types of metals used in an automobile (by weight)

Iron from used vehicles is processed either in ironworks where steel is produced in convector stoves or in plants manufacturing steel in electric arc furnaces. In 1995 annual amount of recycled steel in European Union exceeded 400 000 tons. Recycling of steel requires only half of the energy needed for ore smelting because all mining processes are omitted. [35, 38, 39]

Highest amount in nonferrous metals recycled from ELVs has aluminum. The material is very vulnerable to contaminants and needs to be precisely segregated. It is melted in convection furnaces with presence of fluxes due to high oxidation ability of the metal. Recycled material is used for manufacturing of less responsible products like household equipment. [35, 38, 39]

Plastics

Plastics used in the car industry have risen considerably, where an average new car in 1984 contained 8.5% by weight of plastics a similar car today contains around 11%. Content of the polymer materials in different parts of vehicle: [35, 38]

- Interior – 65%
- Car body – 15%
- Powertrain – 10%

- Electronics – 5%
- Chassis – 5%

Plastics are used for their distinctive qualities, such as impact and corrosion resistance, in addition to low weight and cost. Due to its lightweight properties, the use of plastics can lead to considerable energy savings, with a car weighing 1.3 tons without plastics consuming approximately an extra 1000 liters of fuel during its life compared to a car weighing 1.1 tons with plastic . Despite the relatively high recycling rate for ELVs, the proportion of plastics from ELVs being recycled is extremely low. One reason for this is the wide variety of polymer types used. Thermoplastics are only plastics that can be melted for the second time and used again. Recycling process of thermosets and thermoplastics is very complicated and difficult therefore not entirely profitable. Identification, by marking components at production or by improved sorting technologies, will be vital if the practice of recovering plastic parts is to become viable. [35, 38, 39]

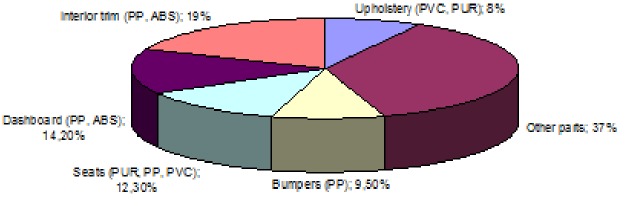

Figure 10.26 Types of plastics used in automobile (by weight)

The most common automotive plastics types are polypropylene (PP), polyethylene (PE), polyurethane (PU) and polyvinylchloride (PVC). PP accounts for approximately 41% of all car plastics (common in bumpers, wheel arch liners and dashboards), and like PE and PU (most common in seat foam) it is easily recycled. Viable markets for PP, PE and PU from non-automotive sources already exist. [35, 38, 39]

PVC makes up about 12% of the plastics content of an average 1990s European car. PVC, by contrast, is relatively difficult to recycle, and there are currently no large-scale recycling schemes operating for post-consumer PVC. Alternative disposal methods such as incineration have raised a number of environmental concerns including dioxin emission during incineration and the use of phthalate plasticizers, which are thought to be disrupters of hormone systems. Nevertheless, this is likely to change due to proposals for a European Directive on the disposal of PVC. Car manufacturers are currently looking for alternatives to PVC. [35, 38, 39]

Typical process of plastics recycling is shredding the parts into granulate. Next the material needs to be sorted. For faster and more precise effects companies employ methods that base on differences of physical properties between types of plastics: [35, 38, 39]

- Floating – the process takes place in water with addition of surfactant substances
- Centrifugal segregation
- Pneumatic segregation – here, except density, important are shape and size of the elements
- Electrostatic
- With usage of organic solvents which melt the sorted material

Achieved materials have properties lower than newly produced so before manufacturing of new parts ennobling of the material is necessary. Possible is also further, "cascadous", usage of the same material but in less responsible parts. For example in FIAT plastics from bumpers are reused for manufacturing interior carpets. [35, 38, 39]

When remanufacturing of plastics is not possible or not profitable enough an option is burning the material in furnaces for energy recovery. Calorific value of plastics is comparable to typical hydrocarbons which makes the method suitable for use. [35, 38, 39]

Glass

Glass constitutes approximately 3% of a vehicles weight. There are two types of glass used in the auto industry, toughened and laminated. Toughened glass is easy to remove from vehicles after shattering. Laminated glass, however, doesn't shatter and will need to be removed manually, which is time-consuming. Some windshields contain wiring and ceramic or galvanic coating which impede recycling of the parts. Unwanted additives are difficult to remove which yields to usage of the recycled glass only for production of glass products with lower transparency. [35, 38, 39]

10.5.2 Other parts/agents

Vehicle operating fluids

This is one of the areas of greatest concern regarding motor vehicles. Although the disposal of fluids from ELVs is a major issue, the effects of inappropriate treatment of fluids removed during servicing are also significant. Basic operating fluids necessary to proper disposal and recycling are lubricant oils, coolants, brake fluids and cooling agents in air conditioning systems. Increasing amounts of engine oil are being recovered and recycled however less than a third of waste oil is recycled. Lubricating oil has the greatest pollution potential. Much of the waste oil collected for recovery is processed (by removing excess water and filtering out particulates) and used as a fuel burnt in heavy industry and power stations. However, stricter emission limits and fuel quality controls resulting from environmental legislation could mean a reduction in the amount of waste oil used in this way. There are multiple technological methods of used oils regeneration. After the process they often have properties similar to new agents. First step during oil regeneration is removal of particulate matter and water. Further actions depend on the used recycling method i.e. distillation, refining, adsorption. [35, 38, 39]

When removed, oil filters can retain large amounts of oil and this may be discarded with the filter leading to further pollution. Vehicle dismantlers leave oil filters on the engines and they are recycled along with them. Oil can be recovered using special oil filter presses which squeeze out the oil and the remaining flattened metal filter can be recycled with other steel. Oil filter crushers are available for use on site at garages, although this is currently not common practice. Nevertheless, it is hoped that oil filter crushers will be increasingly introduced into civic amenity sites. [35, 38, 39]

It is estimated that nowadays up to 50% of the 20,000 tons of oil removed from vehicles by motorists is handled improperly. If oil finds its way into sewers and water courses it can cause significant contamination. One liter of waste oil is sufficient to contaminate one million liters of water and oil poured onto the ground will affect soil fertility. [35, 38, 39]

In case of coolants recycling there is no need to segregate them into different types. First oils and other unwanted particles are filtered, then in process of distillation water and glycol fractions are removed. In the end the glycol fraction is rectificated. This substance is again used in coolant production and other fractions take part in manufacturing of antifreeze employed in aggregate transportation. [35, 38, 39]

Basic contents of brake fluids are solvents, lubricants, antioxidants and corrosion protection additives. In the beginning particulate matter is removed and then distillation process is employed. Water and ether fraction are extracted in temperature of 110 – 160°C, glycol in 160 - 190°C. Ethers and glycol are used for brake fluids production, whereas the rest for antifreeze agents for coal storing and transportation.

Catalytic Converters

Catalytic converters are widely used in both diesel and gasoline fueled engines. In the US, there is a well-established network of agents who collect the catalytic converters. The steel from the exhaust and the precious metals can be recovered when the catalytic converter is replaced.

Platinum, rhodium and palladium are suitable for reuse, either in new automobiles or for some other purpose, and as 68% of platinum and 90% of rhodium used in Western Europe go into the production of catalysts. The ceramic porous insert is also recovered as a powder for refining. [35, 38, 39]

Inserts are melted with presence of a base metal which binds with the noble metals during the process. Later the alloy is heated and the base metal oxidizes leaving pure noble metals. In second method the noble metals are extracted from the inserts and later separated using the different physical properties of the substances. [35, 38, 39]

Batteries

Batteries are especially dangerous to natural environment. They contain significant amounts of lead and other heavy metals, sulfuric acid and plastics like polypropylene and ebonite. It is assumed that 60-70% of worlds lead production is for battery manufacturing which makes recycling of the parts very justifying. EC Directive 91/157/EEC requires the separate collection of certain batteries, including those containing more than 0.4% lead by weight, which includes vehicle lead acid batteries. There is a well-established system for the recovery of lead acid car batteries with many local authorities and garages having collection points. The recycling rate for car batteries is estimated to exceed 90%. However, a significant number of batteries is still not recovered and recycled (for example, many scrap cars still contain batteries when they are shredded). A revision of the existing battery legislation is currently being undertaken. EU proposals include a 70 - 100 % collection target for automotive lead acid batteries with a recycling target of 50 - 80%. [35, 38, 39]

In the past the basic recycling method was used. Whole old batteries were melted. Effectiveness of the process was very low and today popular is shredding the batteries after removal of the electrolyte. Then the material is divided in water with special additives on two fractions: organic and metallic. Polypropylene is extracted from the organic fraction and reused while lead is melted in furnaces. The method provides 95% recycling of the lead from batteries. [35, 38, 39]

Secondary Restraint Systems

Secondary restraint systems used in vehicles consist of airbags and seat belt pre-tensioners. Some air bags are only activated as a result of certain types of collisions, so occasionally the bag is undetonated and in the absence of manufacturers' deployment instructions, a strict procedure should be followed in order to disarm the bag safely. Air bags do not contain high value materials, so reclamation is not a viable option. In addition, because of the high product specifications and specialist installation procedures required to fulfill their safety purpose, reuse is not currently an option either. [35, 38, 39]

Tires and other rubber elements

Rubber accounts for about 6.7% of the vehicles mass. Elements made of rubber are sealing, hoses, dampers but the most important are tires, since they need to be periodically replaced. In typical lifetime of a car tires get worn down at least few times. Annual number of used tires in European Union exceeds 2.5 million tons and 7.5 million in the world. Unfortunately only around 23% of the parts are recycled due to lack of the processing facilities limited receivers. [35, 38, 39]

Tire disposal methods

Waste prevention is a primary objective when looking for future developments in scrap tire options. Ongoing research into improvements in tire design and construction has resulted in the life expectancy of tires continuing to lengthen. [35]

- Reuse of part-worn tires
- Reuse through landfill engineering

Whole tires can be used in the preparation/construction of landfill sites, where they are used as leachate draining systems. [35]

Recycling through rethreading

Rethreading involves either replacing only the tread section or replacing rubber over the whole outer surface of the tire. Manufacturing a retread tire for an average car takes 4.5 gallons less oil than the equivalent new tire and for commercial vehicle tires the saving is estimated to be about 15 gallons per tire. Car tires can only be rethreaded once while truck tires can be rethreaded up to three times. [35]

Recycling through grinding

Grinding is the most widespread materials recovery process. In 1999 it is estimated that 83,000 tons of tire were granulated. This process produces a range of crumb sizes through the progressive size reduction process with the energy used to break up tires increasing as the particle size decreases. Crumb is used in sports and play surfaces, brake linings, landscaping mulch, carpet underlay, absorbents for wastes and shoe soles. Crumb can also be recycled in road asphalt. Rubberized asphalt can increase road elasticity, temperature range and resistance to oxidation, which can result in fewer ruts, potholes and cracks in the surface. Some crumb can be used in formulations with virgin rubber, but this is less than 5% of the total. [35]

Recycling through cryogenic fragmentation

During cryogenic fragmentation, tires are shredded and cooled to below minus 80 degrees C. A hammer mill then pounds the chips to separate the components. The resultant rubber granules can be used for athletics tracks, carpet underlay, playground surfaces and rubberized asphalt for road surfaces. The energy input required for such low temperatures is relatively high. [35]

Recycling through de-vulcanisation

Treating vulcanized rubber with heat or chemicals can produce devocalized rubber, which can be used to replace part of the virgin material in automotive and cycle tires, conveyor belts and footwear. The variety of uses for this rubber has been limited due to its unreactive nature leading to poor bonding and strength. Possible uses are for automotive components, building products, coatings, sealants and containers for hazardous waste. The developers believe it provides a valuable option for waste tires. [35]

Recycling through microwave technology

Advance Molecular Agitation Technology (AMAT) have developed a prototype using microwave technology. This breaks the tires into their original components. The steel is of grade A quality and can therefore be sold for recovery, the carbon and oil are also reusable. The amount of emissions produced are minimal. [35]

Energy recovery

The process of energy recovery from tires bases on burning the rubber material in industrial furnaces. Due to rubbers higher than coal calorific value the method has profitable justification. Noticeable excess of air in industrial heat installations enables significant reduction of toxic products of incomplete combustion emission. Steel insert of tires is later employed as construction material. [35, 38]

10.5.3 The European Union End-of Life Vehicles (ELV) Directive

The End-of-Life Vehicles Directive (2000/53/EC) came into force on 21 October 2000 and Member Countries should have enacted legislation to comply with the Directive by 21 April 2002. This act was aimed strictly to regulate the disposal of used automobiles. It considered wide range of actions and processes from design of a vehicle, through usage, to proper utilization and reusage of the materials from which a vessel comprised. The Directive will require EU Member Countries to: [37, 40]

- Ensure that all ELVs are only treated by authorized dismantlers
- Provide free take-back of all ELVs for new vehicles put on the market after 2002; from 2007 provide free take-back for all vehicles including those put on market before 2002
- Restrict the use of heavy metals in vehicles from July 2003
- Ensure that a minimum of 85% of vehicles are reused or recovered (including energy recovery) and at least 80% must be reused or recycled from 2006, increasing to a 95% reused or recovered (including energy recovery) and 85% reused or recycled by 2015. For vehicles produced before 1980 the minimal reuse and recovery rate was set for 75%, while reuse and recycling rate was set to 70%.

It also requires the 'de-pollution' of vehicles before being recycled. This involves extracting petrol, diesel, brake fluid, engine oil, antifreeze, batteries, airbags, mercury-bearing components and catalysts. [37, 40]

10.5.4 Trends in automobile development influencing vehicle recycling

Automobile manufacturers constantly introduce new improvements in their companies to increase efficiency of the products and lower the costs. International regulations and change in consumer approach to natural environment cause inventing of methods that help in vehicle recycling processes. Those actions involve both design of cars and employment of certain materials: [38, 41]

- Parts adapted for dismantling – avoiding elements combined from different materials, introducing easier to disconnect links (snaps instead of bolts), precise and clear dismantling instruction elaboration for every vehicle
- Adapting to material selection – distinguishing certain materials by color for easier identification, parts heavier than 100 g are signed with a symbol of the material
- Usage of materials easier to recycle – replacing thermosets with thermoplastics, more intense employment of biodegradable materials
- Lowering the number of used materials – Toyota invented a new material Toyota Super Olefin Polymer (TSOP-5), which replaces 10 previously used materials and is suitable for recycling

- Limitations in usage of significantly dangerous materials – restrictions in employment of heavy metals like lead, mercury, cadmium and hexavalent chromium as ingredient of automobile parts

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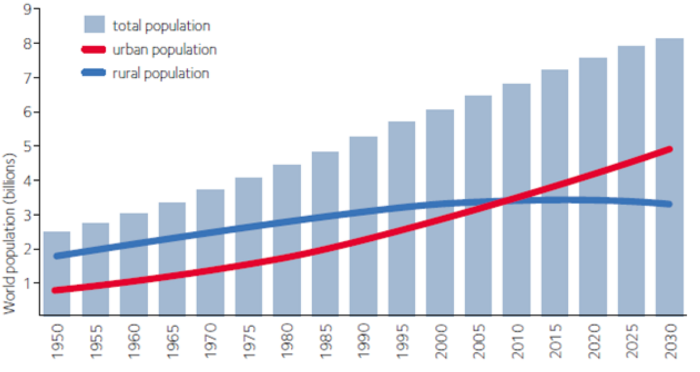
11 MODERN TRENDS IN VEHICLE DEVELOPMENTS

This chapter will cover modern trends in vehicle developments. Modern vehicle design cannot be done without considering social, ethical and environmental factors. Therefore human living environment will be taken into consideration in this chapter.

In Europe, the percentage of the population living in urban areas is expected to rise from 73 % in 2000 to approximately 80 % by 2030, meaning that a large and increasing proportion of the population will limit their daily travel to short / medium distances of less than 100 kilometres, often within entirely the urban environment.



Urban and rural population of the world, 1950-2030
Source: UN Population Division.

Figure 11.1 Urban and rural population of the world

This research field focuses on identifying and analyzing all options to increase safety within the urban environment, integrating vehicle and infrastructure issues, and addressing the need to improve protection of the vulnerable road users. Systems like:

- Automatic brake in case of danger of a crash
- Traffic management – to improve vehicle flow through the city

11.1 Alternative fuels

An alternative fuel vehicle is a vehicle that runs on a fuel other than "traditional" petroleum fuels (petrol or diesel); and also refers to any technology of powering an engine that does not involve solely petroleum (e.g. electric car, hybrid electric vehicles, solar powered). Hybrid electric vehicles such as the Toyota Prius are not actually alternative fuel vehicles, but through advanced technologies in the electric battery and motor/generator, they make a more efficient use of petroleum fuel. Other research and development efforts in alternative forms of power focus on developing all-electric and fuel cell vehicles, and even the stored energy of compressed air.

Figure 11.2 Hybrid Toyota Prius 1

11.2 Alternative fuel numbers

As of July 2010 more than 40 million alternative fuel and advanced technology vehicles have been sold worldwide, compared to around 900 million cars and light trucks in use in the world in 2009.This alternative fuel fleet is made up mainly of:

- 20.7 million flexible-fuel vehicles by mid 2010, led by Brazil with 10.6 million, followed by the United States with 9.3 million, Canada (600,000), and Europe, led by Sweden (199,004).Additionally, 183,375 flexible-fuel motorcycles were sold in Brazil in 2009.
- 11.2 million natural gas vehicles by 2009, led by Pakistan with 2.4 million, Argentina (1.8 million), Iran (1.7 million), Brazil (1.6 million)
- Between 2.4 to 3.0 million neat-ethanol vehicles still in use in Brazil, out of 5.7 million ethanol only light-vehicles produced since 1979.

More than 3.1 million hybrid electric vehicles sold by mid 2010, led by the United States with almost 1.8 million units, followed by Japan with more than 1.1 million and Europe with around 250 thousand. Worldwide, Toyota Motor Company is the leader with 2.68 million hybrids sold by July 2010, followed by Honda Motor Co., Ltd. with more than 300 thousand hybrids sold by January 2009, and Ford Motor Corporation with more than 140 thousand hybrids sold by June 2010.

11.3 Air engine

The air engine is an emission-free piston engine that uses compressed air as a source of energy. The first compressed air car was invented by a French engineer named Guy Nègre. The expansion of compressed air may be used to drive the pistons in a modified piston engine.

Efficiency of operation is gained through the use of environmental heat at normal temperature to warm the otherwise cold expanded air from the storage tank. This non-adiabatic expansion has the potential to greatly increase the efficiency of the machine. The only exhaust is cold air–(15 °C), which could also be used to air condition the car. The source for air is a pressurized carbon-fiber tank. Air is delivered to the engine via a rather conventional injection system. Unique crank design within the engine increases the time during which the air charge is warmed from ambient sources and a two stage process allows improved heat transfer rates.

11.4 Battery electric vehicles

Battery electric vehicles (BEVs), also known as all-electric vehicles (AEVs), are electric vehicles whose main energy storage is in the chemical energy of batteries. BEVs are the most common form of what is defined by the California Air Resources Board (CARB) as zero emission (ZEV) passenger automobiles, because they produce no tailpipe emissions while being driven. The electrical energy carried onboard a BEV to power the motors is obtained from a variety of battery chemistries arranged into battery packs. For additional range genset trailers or pusher trailers are sometimes used, forming a type of hybrid vehicle. Batteries used in electric vehicles include "flooded" lead-acid, absorbed glass mat, NiCd, nickel metal hydride, Li-ion, Li-poly and zinc-air batteries.

Figure 11.3 The Henney Kilowatt, the first modern (transistor-controlled) electric car. Based on a Renault Dauphine

Figure 11.4 The 2011 Nissan Leaf, introduced in Japan and the U.S. in December 2010, is the world's first mass production 100% electric car for sale from a major manufacturer.

Nowadays better and better batteries are being build.

The corresponding quantified targets that need to be met for passenger car Lithium-ion battery systems are:

- Performance: Energy density has to be improved at least to 180 Wh/kg in 2020 (130 Wh/kg in 2015). Current technologies achieve below 100 Wh/kg.
- Durability: Calendar and cycle life targets have to meet the expected lifetime for the vehicle. Batteries must last at least 15 years lifetime or 5500 deep charge/discharge cycles by 2020 (4000 charge cycles in 2015 with 10 years life-time) in order to operate the vehicle without replacement
- Costs: A target of less than 140 €/kWh has to be achieved in 2020 (less than 215 €/kWh in 2015) for a widespread dissemination of electric vehicles.

Correspondingly, in order to avoid over-dimensioning the battery capacity and added vehicle costs, engineers are required for:

- Cost reduction and efficiency improvement for the electric powertrain concerning main drive components e.g. electric machines, power electronics and those specific to the electric vehicle such as range extenders and charging devices.
- Efficient solutions for electrification of vehicle auxiliaries, for example for heating (which today uses waste heat from the combustion process), cooling, steering and integrated passive and regenerative braking,
- System architecture (e.g. redundant concepts in order to ensure the function of safety critical systems) and integration, including novel range extender concepts. Simulation methodologies covering all electrical aspects are requested for the optimisation of the system architecture.

11.5 Solar solution

A solar car is an electric vehicle powered by solar energy obtained from solar panels on the car. Solar panels cannot currently be used to directly supply a car with a suitable amount of power at this time, but they can be used to extend the range of electric vehicles. They are raced in competitions such as the World Solar Challenge and the North American Solar Challenge. These events are often sponsored by Government agencies such as the United States Department of Energy keen to promote the development of alternative energy technology such as solar cells and electric vehicles. Such challenges are often entered by universities to develop their students engineering and technological skills as well as motor vehicle manufacturers such as GM and Honda.

Figure 11.5 Nuna solar powered car, which has travelled up to 140km/h (84mph).

For the optimization of future diesel engines, activities should focus on renewable diesel fuels including biodiesel, Hydro treated vegetable oil (HVO), Dimethylether (DME), as well as on the potential to run diesel engines using gasoline type fuels such as E95 (95 % ethanol).

11.6 Accident prevention

Research is required in the fields of integrated braking and steering for collision mitigation or potentially avoidance (including avoidance of pedestrian collision) and integration of all actions from driver warning to system-initiated intervention, the aim being improved safety while increasing vehicle design freedom.

The challenge consists in defining and combining the relevant driving and driver parameters in order to adapt feed-back to all driving conditions. The amount of feedback and information given to the driver will depend on the traffic and environment conditions. All data that is delivered from internal and external sources to the vehicle needs to be assessed in terms of immediate necessity for the driver, integrated and processed. The feedback will be adapted to the driver in real time, whereas information will be provided to improve the driving style successively when the trip has been completed. The aim of driver feedback is to improve the safety and efficiency, hence reducing CO2 emissions.

A lack of detailed information still exists regarding the pre-collision phase of accidents which is required for the development of primary safety systems. Only limited information derived from reconstruction of accidents and event data recorders is available, and the processes and factors that have an influence on the change from a normal driving situation to a critical situation remain largely unknown. Correspondingly naturalistic driving studies are needed to deliver the necessary data in order to fill this knowledge gap.

11.7 Materials

The affordability and competitiveness of European vehicles is of paramount importance which in the future will depend on the application of light, smart, innovative materials in addition to the new technical solutions for propulsion and safety. Consequently the development of design criteria for weight reduction plays a major role for further improvements in efficiency and lower energy consumption. Increasingly innovative, sustainable solutions can only be developed by following in an integrated approach which takes into account the entire lifecycle of the vehicle when developing intelligent design concepts and new material and process technologies. In the future, series production models will require significantly lighter solutions which offer the best balance in terms of performance, cost, weight, volume and functionality criteria.

Enhanced interior comfort and further improvements in perceived quality will be possible by performing collaborative research into integrating new functions such as:

- scratch resistance,
- self-healing,
- smell reduction,
- haptic quality,
- optical effects (e.g. interior lighting) and
- thermal properties (e.g. Infrared absorption and self-cleaning, reflection, thermal capacity etc.)

11.8 Life-cycle

New approach in both production and recycling of the vehicle is required.The modelling of transformation processes concerns many aspects of relevance including

- material extraction,
- forming, treating,
- finishing,
- assembly and disassembly,
- scrapping,
- Recycling,
- heat generation, neutralisation,
- graving and logistics,
- foundry or processing,

This approach encompasses a series of challenging research topics of high relevance and common interest to the vehicle manufacturers and the supply chain:

- New open, flexible architectures to integrate product and process representation, data and process management, design and evaluation, refinement tools, factory and service feedback.
- Holistic, evolutionary product representations based on semantics, ontology and embedded rules,
- Co-located and remote collaboration technologies and systems, focused on low cost, open solutions preserving security and IPR management,
- Cost-effective, unobtrusive, high immersion rate of virtual reality and augmented reality systems for virtual product engineering and user-in-the-loop testing.
- Real time architectures and middleware are able to seamlessly link product representations and simulation and virtual reality environments.

11.9 Tendencies in vehicle engineering

In this chapter the main topics of the future discussions are presented.

11.9.1 Vehicle dynamics

- Improving of theoretical models describing behavior of vehicles in different conditions.
- Computational models efficiency improvement by the use of bigger and faster computers.
- Improving of road irregularities investigations (for instance by optical laser methods).
- Implementation of the new ideas in designing of different systems having influence on vehicle dynamics.
- Implementation of the new materials in every mechanism having influence on vehicle dynamics.

11.9.2 Powertrain system

- Clutch
  - Mass reduction,
  - Reliability improvement,
  - Better materials,
  - Size reduction,
  - Vibration reduction,
- Shafts and joints
  - Mass reduction,
  - Reliability improvement,
  - Reinforced materials,
  - Very good balancing,
  - Better geometry in joints,
  - Better surface in joining elements,
- Gearbox
  - Mass reduction,
  - Very compact solutions,
  - Quiet work,
  - More gears,

- Automatic also for smaller cars,
  - Better lubricants,
  - Greater reliability,
  - Mechatronics in gearboxes,
- Differential + final drive
  - Mass reduction,
  - Better materials,
  - Compact design,

11.9.3 Tyres

- Mass reduction;
- Smaller rolling resistances;
- Better materials (synthetics);
- Better tread structure;
- Low noise;
- Better collaboration with the road;
- High specialization;
- Better balancing (static, dynamic, optimization);
- Monomaterials;
- Pressure monitoring;
- Safe solutions.

11.9.4 Suspension systems

- Mass reduction;
- More comfort in every class of vehicles;
- Better materials used for suspension elements;
- Reinforced materials;
- Faster reaction for road irregularities;
- Intelligent materials;
- Magneto rheological fluids;

- Permanent improvements of car geometry;
- Computer controlled car movements;
- New solutions not only in high class vehicles;
- Mechatronics in suspension system.

11.9.5 Steering systems

- Mass reduction;
- Size reduction;
- Materials replacements;
- Angle optimization;
- Electro-mechanical solutions;
- 4-wheel steering;
- Force on steering wheel optimization;
- Mechatronics in steering systems.

11.9.6 Brake systems

- Mass reduction;
- Better materials for lining elements (reinforced, high friction coefficient);
- Better thermal conductivity;
- Better braking efficiency;
- Electric solutions;
- Brake system monitoring;
- Mechatronics in brake systems (ABS, ESP, ASR, etc.).

11.9.7 CAD/CAM/CAE software

- Use of advanced algorithms for computer aided optimization
- Wider use of computers in vehicle design
- Higher degree of relations between parts which are being designed
- More complex calculations – thanks to more powerfull computers

11.9.8 Noise reduction

- Better understanding of noise phenomena;
- Better insulating materials;
- Layered solutions;
- Silencers in main systems;
- External and internal noise reduction.

11.9.9 Heat protection

- More efficient cooling systems;
- Better insulating materials;
- Better understanding of heat transfer phenomena;
- Reduction of fuel consumption (less heat loss);
- Computer control over all heat-producing units;
- Mechatronics in every heat-producing system.

11.9.10 Materials

- Replacement of contemporary materials by non-conventional materials which are coming to practice
- Use of high strenght steels
- Introducing low-cost materials
- Implementing of more plastics in every vehicle system
- Application of reinforced and layered elements
- Elimination of non-recyclable materials
- Reducing the variaty of plastics

11.9.11 Car body optimization

- Use of advanced algorithms for computer aided optimization
- More complex calculations – thanks to more powerfull computers

11.9.12 Recycling

- Parts adapted for dismantling
- Adapting to material selection
- Usage of materials easier to recycle
- Lowering the number of used materials
- Limitations in usage of significantly dangerous materials

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