STRUCTURAL DESIGN AND PERFORMANCE OF CAR BODIES

Pierre Duysinx Research Center in Sustainable Automotive Technologies of University of Liege Academic Year 2021-2022

Goals of the lesson

- Introduction to:
 - Estimation of load cases and stress level applied to the chassis
 - Different types and technologies of chassis and their major application domains
 - Introduction to simplified structural analysis (SSA) approach of chassis
 - Detailed structural analysis of chassis using numerical simulation tools (e.g. Finite Element Method) and structural optimization
 - Different kinds of analysis carried out during design process of automobile structures

Design load cases

DESIGN LOAD CASES

- We consider
 - Passengers cars and light duty vehicles
 - Loads come from road roughness and from maneuvers
- Five major load cases:
 - Bending: the chassis is loaded in the vertical plan xz due to the weight of the components along the chassis
 - Torsion: The vehicle structure is subject to a torque by applying a two opposite vertical forces at both end of the axles (at the knuckle)
 - Combined bending / torsion: actually the gravity loads always act during the torsion and it is reasonable to consider simultaneous application of bending and torsion loads.

DESIGN LOAD CASES

- Five major load cases (cont'd):
 - Lateral load cases: creation of loads in lateral direction y when the vehicle is turning or when it touches a curb
 - Longitudinal loads (forward or rearward) : inertia loads of components onto the chassis when developing large acceleration / deceleration (braking)
- The most severe ones are: bending, torsion, and combined bending / torsion
- The lateral loads and severe acceleration / deceleration load cases are important for the design of suspension elements and for the sizing of the connection points between the chassis and the suspension parts







Happian Smith: Bending load case

Torsion load case





Happian Smith: Combined bending and torsion load case







Happian Smith: Lateral force load case

Braking / Acceleration load case

DESIGN LOAD CASES

- Other local load cases (not considered hereunder):
 - Efforts in hinges when opening doors and hood
 - Load cases created by attachment of seats and seat belt during emergency braking and brutal decelerations due to crash

- Principle:
 - Considering the vehicle chassis as a beam in the car midplan xz, the vehicle being assumed to be symmetric about its mid-plane
 - Determine the static distribution of the weight loads from the vehicle components along x axis.
 - State the list of the main (heaviest) vehicle components
 - Compute the vertical forces per unit distance along x
 - Calculate the reaction forces under the wheel axles (still assuming the vehicle structure as a beam)



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- Principle:
 - Establish the diagram of bending moments and shear efforts along the x axis of the vehicle
 - Calculate the stress state and deformations of the car body and in the vehicle longitudinal beams



From Happian Smith : The diagram of bending moments and shear efforts along the x axis of the vehicle

- Dynamics loading should be better considered when the vehicle faces uneven road surface.
 - For instance when the car overtakes a road bump, the front wheels may leave contact with the ground. When touching the road again, the load under the wheels will increase severely.
- Following the expertise accumulated by car manufacturers, the static load should be replaced by an equivalent static load with a magnification factor:
 - 2,5 to 3 for road vehicles
 - Up to 4 for all terrain operations.

LOAD CASE 2: TORSION

- One considers now the pure torsion load case applied under one the axle while a reaction moment is developed at the other one.
- The base torque is calculated with the vertical forces under the lightest axle. Its value is given by the vertical force under the lightest axle multiplied by the axle track length:

$$M_x = R_F \frac{t_F}{2} = R_R \frac{t_R}{2}$$

 Generally t_F and t_R are different and the lightest axle is often the rear axle R_R<R_F.

LOAD CASE 2: TORSION



From Happian Smith: Computation of torsion moment

LOAD CASE 2: TORSION

- Again one should consider dynamic loading instead of a static load.
- From car manufacturer experiment feedback, the quasi static loads must be multiplied by a dynamics factor to estimate the equivalent static loads used for structural analysis:
 - 1.3 for road vehicles,
 - 1.8 for all terrain vehicles.

LOAD CASE 3: BENDING + TORSION



From Happian Smith: Combined bending – torsion load case

- Combination of bending and torsion loads gives rise to the most critical situation in which one wheel of the lightest axle lifts off the ground and the vertical force under this wheel vanishes.
- The vertical force of the axle is then totally withstood by a single wheel.
- In addition it is usual to recommend to limit the lift off by 200 mm, which corresponds to the maximal rebound of usual suspension mechanism

LOAD CASE 3: BENDING + TORSION



- Load under front right wheel
- Load under the front left wheel

• Example:

- t_F = 1450 mm and t_R = 1400 mm
- Vertical load under the lightest axle: $R_R = 6184 \text{ N} (R_F = 7196 \text{ N})$
- Torsion moment

$$M_x = R_R \frac{t_R}{2} = 4328Nm$$

Loads under the front wheels

$$R'_{F} = M_{x} \frac{2}{t_{F}} = 5971N$$

$$R_{FR}^{tot} = \frac{R_{F}}{2} - \frac{R'_{F}}{2} = \frac{7196}{2} - \frac{5971}{2} = 613N$$

$$F_{FL}^{tot} = \frac{R_{F}}{2} + \frac{R'_{F}}{2} = \frac{7196}{2} + \frac{5971}{2} = 6583N$$

 When taking a turn, tires develop lateral forces that counterbalance the centrifugal acceleration

 MV^2/R

- The most critical situation happens when the vehicle is close to roll over, because the whole vertical load is transferred on the external wheel.
- The structure is modelled as a beam in the horizontal plan xy



 The lateral acceleration leading to the critical situation is obtained by writing the equilibrium of the vehicle

$$\frac{MV^2}{R}h = Mg\frac{t}{2}$$

- $\frac{V^2}{R} = g \frac{t}{2h}$
- The critical acceleration is given by:
- The centrifugal force is MV²

$$\frac{MV^2}{R} = Mg \frac{t}{2h}$$

The lateral forces under the front and rear axle writes

$$Y_F = Mg \frac{t}{2h} \frac{c}{L} \qquad \qquad Y_R = Mg \frac{t}{2h} \frac{b}{L}$$

- We can model the structure as a beam on two supports and loaded in the plane xy at the center of mass.
- A more detailed model can be carried out using a mass distribution along the x axis as it is made for body bending under the self weight.
- In practice, the critical situation never occurs because the usual values of elevation of the CG and because of the limitation of the coefficient of adhesion
 - h=0,51 m and t=1,45 m
 - $g t/2h = g 1,45/(2*0,51) = 1,42 g > \mu g \sim 0,8 g$
- So no safety margin is generally accounted in this load case, the design being already very conservative.

- Conversely impacts on curbs or obstacles induce large loads and results sometimes in the vehicle roll over.
- The lateral loads are generally not a critical design case study because of the large bending moment inertia of the body in the x-y plane.
- Nonetheless it is a severe design case for the attachment points of the suspension mechanisms, which have to be able to withstand the high load levels happening during the shocks.
- For safety reasons, one generally considers dynamic loads twice bigger than lateral and vertical static loads under the wheels.

LOAD CASE: LONGITUDINAL

- The longitudinal loads comes from the inertia forces during high acceleration and braking maneuvers.
- High acceleration and braking maneuvers lead to a large weight transfer between the front and rear axles and vice-versa.
- To be fully rigorous, one should determine the horizontal and vertical position of each components along the axis of the vehicle and then calculate the acceleration loads.
- However this is general difficult because of the lack of information at the preliminary design stages.
- So in the preliminary design phases, one uses a simplified model in which the full mass is lumped at the Center of Gravity.

LOAD CASE: LONGITUDINAL

 <u>In case of acceleration</u>, one has the following weight on the front wheels:

$$R_F = \frac{Mgc}{L} - \frac{Mh\dot{v}}{L}$$

While for the rear wheels it comes:

$$R_{R} = \frac{Mgb}{L} + \frac{Mh\dot{v}}{L}$$

In case of braking, one gets:

$$R_F = \frac{Mgc}{L} + \frac{Mh\dot{v}}{L} \qquad R_R = \frac{Mgb}{L} - \frac{Mh\dot{v}}{L}$$







LOAD CASE: LONGITUDINAL

- The braking and acceleration forces are limited by the saturation of the friction coefficient between the tire and the road.
- Other bending moments
 - The traction and braking forces developed at the tire contact patches are transferred to the body through the suspension mechanisms and thus create torques, which must be sustained by the body structure.
 - Another effect is related to the vertical position of the CoG. It comes that the acceleration loads produce an additional bending moment onto the structural members.



- One experiences an asymmetric loading onto the body when one wheel hits an object with a certain height at the ground level or when one of the wheels falls into a pothole.
- This case leads to a vertical and a horizontal load along one single side of the car. It results into a complex loading of the structure.
- The applied load depends on the initial vehicle speed, the suspension equivalent stiffness, the mass of the wheel and of the non suspended mass, etc.
- As the shock has a short duration, one can reasonably assume that the wheel continues along its trajectory with the same speed and that the force is directly transmitted to the wheel center.

Longitudinal component of the load:

$$R_{ux}=R_u\cos\alpha$$

Vertical component of the load:

$$R_{uz}=R_u\sin\alpha$$

Angle:

$$\alpha = \arcsin\left(\frac{r_d - h_u}{r_d}\right)$$

• α increases for large diameter wheels



- The vertical component implies an additional load onto the axle, an inertia load applied on the CoG, and a torsion moment on the body structure.
- The horizontal load creates a bending moment in the xz plane and a moment about vertical direction 'z' on the body structure.
- The unsymmetrical load case can be decomposed into the superposition of for simple load cases.



From Happian Smith: Asymmetric load case

Strength of the Body: Limit stresses

- The previously discussed load cases leads to an estimation of the stress level all over the structure.
- For critical load cases, stress level, or the equivalent stress constraints must remain below the allowable stress limits, which are determined according to the usual rules.
- One also considers safety factors in connection with standards in the field
 - Dynamic factor
 - Safety margin: which is typically 1.5
- A similar approach is accounting for the fatigue resistance, even if the fatigue resistance is mostly investigated in the points where there is a high stress concentration, for instance in the connection joints of the suspensions.

Stiffness requirements in bending

- Up to now, we have investigated the loads and the related stress levels in the structural members, which assesses the strength of the structure and prevents the failure of the material.
- However, an important criterion, may be sometimes the most important one, is the stiffness of the car body. A structure resists but it can be too compliant and so does not fulfill its mission, for instance for vehicle dynamics.
- The required stiffness can be determined by various specifications from tolerant gap for the assemblage, the vibration and noise transmission, road holding, or some empirical considerations elaborated by the car manufacturer.

Stiffness requirements in bending

- For a passenger car, the bending stiffness must be sufficiently high because of multiple reasons.
 - Door opening: if the body is too compliant, the doors when opened can not be closed again a good manner.
 - The stiffness of the floor is necessary for passenger acceptance. One uses for instance some omega stiffeners which are hot pressed during the sheet metal forming or by using sandwich panels.
 - Reduction of the vibration generation and transmission in the various panels.
 - Stiffness of wings, doors, and car bonnet etc.
 - One has to reinforce the members around the connection points with the seats, the seat belt, etc.

Stiffness requirements in torsion

- The torsional stiffness is an essential criteria that can be evaluated in a quantitative way.
 - A good car body will exhibit a torsion stiffness of at least 8.000 10.000 Nm/degree during a torsion test.
- A too low stiffness is detrimental:
 - Difficulties for the doors opening and closure
 - Difficulties for the vehicle dynamic performance.
- The torsion stiffness is strongly influenced by
 - The wind shield contributes a lot to the global torsion stiffness (around 40%)
 - The presence and the stiffness carried out by the roof is a major contributor to the global torsion stiffness (counter examples see the loss of torsion stiffness of car bodies in convertibles)

Types of vehicle chassis

Types of vehicle chassis

- Review of different technologies of vehicle body construction
- Assessment criteria
 - Ability to sustain the different types of load cases
 - Manufacturing technologies and cost for mass or series production
- Different types of chassis:
 - Ladder frame chassis
 - Cruciform frame
 - Torque tube backbone frame
 - Space frame chassis
 - Integral structure and semi monocoque structure


- Historically, horse carriages were built from a ladder frame structure on which a superstructure and a body were attached.
 - Resistance of the superstructure is weak (made of wood) and it offered a poor protection of passengers in case of crash.
 - The ladder frame is in charge of resisting to both bending and torsion loads.
- Major advantage:
 - Its great adaptability to accommodate to a wide variety of shapes and body types or superstructures.
 - It is still currently used widely for utility vehicles, from pick-ups to trucks.





- Ladder frame includes
 - Two longitudinal rails running along the structure
 - Transversal or cross members maintain a prescribed distance between the longitudinal beams.
- Longitudinal rails are made of beams with open or closed cross section profiles (the later have better torsional stiffness)
- To achieve a high bending stiffness to mass ratio, the rails have large height to increase their bending inertia for a given cross section area.



- The flanges contribute to the bending inertia and are able to sustain high stresses.
- Open cross sections provides an easy access to attaching brackets and component supports.
- Attaching the bracket to the web allows not to reduce the effective cross sections of flanges which are subject to high stresses.
- The shear centre of open cross sections is located outside of the profile and of the web. This means that loads applied through the brackets can be applied at the shear centre to generate purely bending moments and negligible torsion torques.





- However cross sections exhibit poor torsional stiffness. Because of the nature of ladder frame structure, the bending of the longitudinal rails results in torsion of transverse beams and vice-versa. Thus as ladder frame structures globally exhibit a poor torsion stiffness.
- To increase the torsion stiffness, one must resort to longitudinal and cross beams with closed cross sections.
- With closed box sections, the design and the strength of the joint become a critical issue.
- On the one the side, the maximum stresses are concentrated in these regions.
- On the other hand, the attachment brackets could deform the box profiles and the profile must be reinforced

Cruciform frame



- To increase the torsional stiffness, one can design a frame in the shape of a cross, in which no element will be subject in torsion.
- The two beams are only loaded by bending moments.
- The overall torsional stiffness depends on the transmission of the bending moment at the central connection. There, the bending moment is maximum and the joint must be reinforced to present a great bending stiffness.

Cruciform frame

- The real frames generally combine the technologies of ladder frame structure and cruciform structures by mixing the longitudinal beams and the cross frame reinforcement pattern.
- One obtains a structural frame that exhibit simultaneously a good overall bending and torsional stiffness.
- The cross beams under the front and rear axles participate efficiently to resist lateral loads and torsion moments



Torque tube backbone frame





- Closed wall boxes have a high torsional stiffness. This property is exploited in torque tube backbone chassis initially proposed by Lotus (e.g. Lotus Elan).
- The backbone of the chassis is made of a large tube with a closed box cross section.
- The transmission shaft runs through the tube from engine compartment to the rear live axle.
- Columns are used at the front and the rear to extend the suspension connection points while transversal beams increase the resistance capability against lateral forces.

Torque tube backbone frame



- The backbone is subject to both flexural and torsional loadings.
- The columns under the suspension connection points are loaded in bending.
- The transverse beams work in traction and compression against the cornering forces developed in the tyres.

Space frame chassis





- Previous chassis structures were mostly2D, apart from the height of the beams.
- Using 3D structural designs, bending and torsional stiffness properties are greatly enhanced because the inertia of the structure is increased due to the presence of material far away from the neutral axis.
- 3D structures are usually used for race cars, sport cars or small series vehicles.
- The 3D structure can be combined with lightweight bodies made for instance of composite materials.

Space frame chassis



- Structural elements of truss structures <u>must work in tension and compression</u> otherwise the overall stiffness drops drastically because it is dominated by the stiffness of the connection joints which is weak (pin joints).
- It is mandatory to create triangular patterns to remove all kinematic deformation modes.
- However weak points are generally related to windows panels and from doors and opening which decrease the overall stiffness of the structure.





- Modern vehicles are mostly based on integral chassis made of stamped steels joint using spot welding.
- The structural components have double functions: structural and body parts with aerodynamic and aesthetic)
- The structural stiffness (in bending and torsion) takes benefit of all components, especially the ones which are far from the neutral axis such a roof panels.



- Structural analysis of mass production car body is obviously more complex and resorting to FEM is generally necessary.
- The system is massively indeterminate and the load path may be redistributed over different components.

The integral chassis presents one of the highest stiffness / mass ratio if compared to solutions based on separate chassis and body.

- The integral body can be produced in a mass production scale for a mastered cost.
- The comfort (in particular noise and vibration performance) is superior to any other ones



The integral chassis includes three compartments:

- The <u>central compartment</u>, the largest one, is located between the two axles provides a volume for the passengers
- The <u>front compartment</u> is generally devoted to the powertrain, engine / e-motor and transmission units
- The <u>rear compartment</u> is the space devoted to the freight and the bags.

- Each compartment is made of structural members which work:
 - In compression (c)
 - In traction (T)
 - In flexion
 - In torsion



(a) Lengthwise body loading



(b) Transverse body loading

- The structural members are manufactured from metallic parts or stamped steel profiles
- Their shape is adapted to the loads to sustain and to their position in the vehicle





(a) Small central tunnel, sills, front valance, rear wheel arches and all round spring to



(b) Large central tunnel, sills, front valance and spring tower with rear box-section reinforced seat par



^{:)} Longitudinal front members, sills, rear seat pan with bulkhead and spring tower

- The rigidity of modern integral chassis rely heavily on the stiffness of the platform to reduce the size of the pillars and of the roof to enlarge the windows surfaces and to increase the driver's visibility.
- The bending stiffness derives mostly from the inertia of the transmission tunnel and from the longitudinal beams. If necessary additional longitudinal rails more in the inner part of the body.
- The torsional stiffness mainly comes from the dashboard panel, from the side skirts, and from the rear bulkhead. The spring beams also contribute to the stiffness if present.

- To increase the bending stiffness of the platform, one adds integrated stiffeners with open or closed cross sections either in longitudinal or transverse directions. The cross sections are manufactured in the panels of the floor or of the walls.
- It is also possible to stiffen the shells by pressing grooves in the body panels.
- Thus one increases also the local buckling loads of these panels and increases the natural frequencies of the thin wall shells so to improve the NVH comfort.



- It is also necessary to use chassis subframes at the front and rear axles to withstand the suspension loads.
- The subframes provides an adequate support to the suspension lower arms. They also maintain a prescribe geometry between the two wheels of the axle, preventing skew deformation of the chassis.
- It is also usual to fix the engine and the gear box mountings on the subframe so that the chassis, which is made of thin metal shells, have not to be reinforced as a whole in order to take the engine reaction torques.







- Another advantage of subframes is to isolate the chassis from the chocs coming from the rolling motion and from the vibrations of the engine if one uses rubber bushings. This is highly positive for NVH performance.
- For longitudinally mounted engines and rear propulsion, one often adopts subframes with beam shape (See figure bottom right).
- One can use a similar design to transfer the suspension loads from rear trailing or semi training suspensions and to insure a better load distribution of the point loads over the integral chassis.



D) Horseboe subframe



- For longitudinal mounted engines and front driven wheels, a subframe with the shape of a horse shoe is general preferred to distribute the high loads related to the powertrain weight (See central picture in the right).
- This shape combines two functions: to serve as the engine frame, and to host the connection hinges/ balls of the front suspension lower arm and of the anti roll bar.





- Finally for the transversely mounted engines, the subframe design has generally quadrangular shape (see top figure). This design is well appropriate to take and disseminate the propulsion efforts and the weight of the powertrain components.
- The quadrangular shape of the subframe allows to increase the torsional rigidity without introducing severe point loads in the main chassis.







Integral chassis: Analysis using the SSS method

- Despite the power of Finite Element Method (FEM) it is interesting to have simple models to understand the load bearing mechanism in action in the structure.
- The <u>Simple Structural Surface (SSS</u>) method introduced by Pawlowski (1964).
 - Is able to describe and the estimate the loads and the stresses in the main structural elements of the integral chassis, even if they are redundant.
 - Main assumptions:
 - The panels only resist to in-plane loads (shear panels).
 - The beams (pillars, rails, etc.) take only axial efforts (mostly) (traction/compression) and sometimes bending loads.

Integral chassis: Application of the SSS method



From Happian Smith

Structural Analysis of light duty vehicle using the method of Simple Structural Surfaces



- A Simple Structural Surface is a structural component
 - Resisting only to in-plane efforts: tension, compression, shearing, and in-plane bending,
 - Offering negligible stiffness to out-of-plane loads and moments (flexion, torsion).

From Happian Smith





- Structural analysis of a stiffened beam with a shear panel.
 - Thin panel framed with bars
 - The truss is unstable in shear without the panel
 - With a bar along the diagonal, the structure is statically determinate (fully determinate by equilibrium equations) while it is indeterminate with the shear panel.
- We assume that the panel does sustain shear loads only
 - No contribution of the panel to the bending of the cross section



- Equilibrium equations:
 - In the left vertical bar:

$$F_z - Q_2 = 0 \qquad \qquad F_z = Q_2$$

• Rotation of the panel:

 $Q_1 b - Q_2 a = 0$

$$Q_1 = Q_2 \frac{a}{b} = F_z \frac{a}{b}$$

• Upper horizontal bar: $Q_1 - K_1 = 0$

$$-K_1 = 0 \qquad \qquad Q_1 = K_1$$

Lower horizontal bar:

$$Q_1 - K_2 = 0 \qquad \qquad Q_1 = K_2$$



Equilibrium equations:

$$F_{z} = Q_{2}$$
$$Q_{1} = Q_{2} \frac{a}{b} = F_{z} \frac{a}{b}$$
$$Q_{1} = K_{1}$$
$$Q_{1} = K_{2}$$

Bending moment diagram diag





0

 K_4

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- Modelling of a floor panel:
 - Require introducing an auxiliary beam to sustain the vertical load.
 - Equilibrium
 - Panel

 $Q_3 w_1 - Q_4 l = 0$

Beam

$$K_3 + K_4 - F_z = 0$$

 $K_{3}w_{1} - F_{z}(w_{1} - w_{2}) = 0$





Modelling of floor panel: point load

- Introduction of a longitudinal and a cross beam.
- Decomposition by superposition into two load cases
 - Displacements (to render equal \rightarrow compatibility equation)

$$\delta_{c} = \frac{F_{zC}w^{3}}{48EI_{c}} \qquad \qquad \delta_{L} = \frac{F_{zL}ab}{6EI_{L}(a+b)} (L^{2} - b^{2} - a^{2})$$



 $K_2 a = K_3 b$

- Equilibrium + compatibility!
 - Compatibility

 $\delta_c = \delta_L$ $2K_1 + K_2 + K_3 = F_{zL}$ Equilibrium

$$K_{1} = \frac{8a^{2}b^{2}I_{c}F_{zL}}{I_{l}(a+b)w^{3} + 16a^{2}b^{2}I_{c}}$$

$$K_{2} = \frac{F_{zL}w^{3}I_{l}b}{I_{l}(a+b)w^{3} + 16a^{2}b^{2}I_{c}}$$

$$K_{2} = \frac{F_{zL}w^{3}I_{l}a}{I_{l}(a+b)w^{3} + 16a^{2}b^{2}I_{c}}$$

SSS analysis of a box structure

3 4 K_3 K_{l} K_3 Bending K_3 K_2 h K_2 $\downarrow K_2$ K_3 K_{I} $R_f R_f$ K_{2} Κ, K_{I} **Equilibrium equations** $2K_1 - F_z = 0$ (1)Note : (4) $K_2 + K_3 - K_1 = 0$ Roof panel (No.3) carries no loads. $K_1 a - K_3 l = 0$ $2R_f - 2K_2 = 0$ 5

 $2R_r - 2K_3 = 0$

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SSS analysis of a box structure



SSS analysis of a box structure

Pure torsion: without the roof




From Happian Smith

Different models of the chassis structure of vehicles using SSS method.





- SSS 1, 2, 4: Supports the seat loads, support of SSS 7
- SSS 6: Supports the bags weight, the load of rear suspension
- SSS 7: Supports the weight of the engine, of the transmission, of the front suspensions
- SSS 8: Sustains SSS 7
- SSS 9: Transfer of the load towards SSS 10
- SSS 3, 4, 5, 8, 9, 11 12-16: Are loaded in shear



- SSS 7: Supports the rear suspension loads
- SSS 10 : Collects the front suspension loads
- SSS 8, 9 : Support of SSS 10
- SSS 2, 11: Support of SSS 8
- SSS 4, 6, 7, 11-15 : Subject to shear loads



- SSS 1-6: Sustain in-plane bending loads
- SSS 5-10: Sustain torsion loads



Advanced Numerical Solution using the Finite Element Method (FEM) – Case study

- Goal of the study is to estimate the global behavior of the car body made of composites
- Materials:
 - Carbone fibers
 - Aluminum
 - (Honeycomb)





Courtesy of Samtech France and PSA



Material properties

- Face sheets: Carbone fibers
 - MTM49-3/CF1103 for 'hot' parts
 - VTM264FRB/CF1103 for `cold' parts
- Core materials
 - Aluminum honeycomb

Considered load cases

- Natural vibration frequencies
- Body torsion
- Opening parts under their self-weight
- Emergency braking impact of stress level in attachment points



Finite Element mesh

- FE mesh with more than 95% of quadrangular elements
- Front part: 30 000 nodes and 27 000 finite elements
- Body: 217 000 nodes and 220 000 finite elements







First eigen mode of vibration (torsion)



Displacement restriction



Overall torsion stiffness



 Reinforcement of connection points with (front) suspensions⁸¹



- Design of car body is based on various design specifications:
 - Maximize / bound the body stiffness in torsion / flexion
 - Maximize / increase the natural frequencies
 - Satisfy the strength restrictions in various critical points
 - Minimize or limit the mass
 - Minimize or limit the cost
 - Account for the manufacturing constraints

Application of structural optimization to vehicle body design

Formulating car body design as an optimization problem

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- Constraints of car body design:
 - Stiffness against several load cases
 - Stress criteria under various global or local load cases
 - Natural frequencies in order to give good road holding properties as well as the ride and comfort properties
 - Weight and cost
 - Manufacturing and assembly constraints

 How to cast design problem into the standard mathematical statement?



- How to select design variables?
- How to solve it efficiently?

Formulating car body design as an optimization problem

- Car body design lends itself naturally to illustrate the problem of well posed optimization problems:
- Design problem can be cast into the standard mathematical statement:

$$\min_{\substack{x_i \\ \text{s.t.}}} f(x) \\ \text{s.t.} \quad c_j(x) \le \overline{c}_j \qquad (j = 1 \dots m) \\ \\ \underline{x}_i \le x_i \le \overline{x}_i \quad (i = 1 \dots n)$$

• Choice of the design variables, constraints, etc. requires some training.

TOPOL, a topology optimization software tool

- Optimal material distribution approach
- Based on Samcef linear analysis codes (Asef, Dynam, Stabi)
 - Material effective model based on the SIMP model

$$E = \mu^p E_0 \qquad \rho = \mu \rho_0$$

 Implemented at the stiffness and mass matrix level so that 2D, 3D and shell elements can be used

$$\mathbf{K}_e = \mu^p \ \mathbf{K}_e^0 \qquad \mathbf{M}_e = \mu \ \mathbf{M}_e^0$$

- Solver: CONLIN optimizer
- Constraints:
 - Compliance (multiple load cases)
 - Displacements
 - Eigenvalues (vibration, stability)
- Additional features
 - Filtering techniques based on Sigmund's filter
 - Symmetry planes
 - Prescribed density regions

Boss Quattro, a parametric multidisciplinary optimization tool

Complete open object oriented MDO environment



NUMERICAL APPLICATIONS

Design of the car body of a new prototype



Design of the structure of the urban concept vehicle





- Replace car bodies of 2004 and 2006
- Design criteria:
 - Stiffness under 2 major load cases:
 - Bending + roll over
 - Torsion + bending = curb impact
 - Failure criteria in composite material
 - Minimum weight = maximum fuel economy
 - Room available for pilot and propulsion system
 - Pilot visibility



Car body 2004



- At first a CAD model was built in agreement
 - With Eco Marathon regulations,
 - With aerodynamics considerations
 - With space requirements for the pilot and the propulsion system



Aerodynamic shape

Structural shape: wheel covers are removed for maintenance ⁹⁰

- Load case 1: bending
 - Self weight
 - Components (20 kg)
 - Pilot (50 kg)
 - Roll-over load (70 kg on top of roll cage)



(Figures from Happian-Smith)

- Load case 2: torsion + bending = curb impact
 - Rear axle clamped
 - Right front wheel free supported
 - Left front wheel



Finite element model: 24113 shell elements



- Material: black metal [0°/90°/45°/-45°]_s
- Topology optimization
 - SIMP with p=3
 - Filtering
 - Minimum density 0.01

- Topology optimization
 - Minimum compliance
 - Minimum density 0.01

- SIMP with p=3
- Filtering
- Symmetry left/right



Load case 2: curb impact

• Load case 1: bending

- Optimal material distributions suggests clearly the following layout of the shell
 - Low density regions \rightarrow windows for the visibility
 - High density regions \rightarrow panels and stiffened regions



- CATIA digital modeler is used to check
 - The packaging feasibility for the fuel cell and motors
 - The visibility and the lying position of the pilot



 Strength verification was conducted in Samcef Field and Boss Quattro

- Laminate shell
 - [0°/90°/45°/-45°]_s
 - t_i=0,25 mm; t=2 mm
 - Carbon epoxy



- Strength verification was conducted in Samcef Field
- Laminate shell
 - [0°/90°/45°/-45°]_s
 - t_i=0,25 mm; t=2 mm
 - Carbon epoxy
 - Mass: 9,1 kg (-10%)







- For sake of simplicity for this first vehicle: separate function
 - Structure: aluminum truss reinforced with composite panels
 - Aerodynamics: nonstructural shell
- Composite panels
 - Web separating engine (hydrogen!) and pilot compartments
 - Floor
- Truss layout and panel positions were determined with topology optimization
- Composite panels and beams cross sections were verified using Samcef Field







Truss structure for stiffness

- Load case 1: bending
 - Structure + component + pilot weights
 - Roll-over load (FIA): 3*weight





- Load case 2: torsion + bending = curb impact
 - Left front wheel withstanding 3 times the weight of the axle 99

- Intuitive designs
 - Target mass of 20 kg
 - Stiffness and stress level mostly determined by load case 2 (torsion)



Saloon car rather stiff but overstressed





Best intuitive design: 21,62 kg¹⁰⁰

- Topology optimization of the truss structure
 - Target mass of 15 kg
 - Minimum compliance
 - Mostly determined by load case 2 (torsion)
 - SIMP material with p=3
 - Left / right symmetry of material distribution
 - Filtering



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Convergence history



 Discretizatiton of the car using SSS (simple structural surface method) (Pawlowski, 1964)

- Topology optimization of the truss structure
 - Minimum max compliance of load case 1 & 2
 - Volume constraint



Volume = 60%



- SIMP material with p=3
- Left / right symmetry of material distribution
- Filtering sensitivities

Detailed design

- Verification of von Mises stress and Tsai Wu criteria in the two load cases
- Samcef Field / composites















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 The structure has been fabricated and is quite successful: the vehicle weight is 95 kg, while its closest competitors are over 140 kg!





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