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A wave-based substructuring approach for concept modeling of vehicle joints

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ABSTRACT

In highly competitive fields, such as the automotive industry, complex products must be innovated in short timeframes and at affordable costs. The need for simulation tools, able to steer and support the early design choices, pushes researchers to develop concept modeling techniques.

In this paper, an innovative procedure, based on the wave-based substructuring (WBS) technology, is proposed to achieve a database framework for joint concepts, so that joint alternatives can be easily exchanged in concept modification and optimization studies. An industrial case study is analysed, where the proposed methodology is validated through static and dynamic FE analyses.

1. Introduction

More than ever before, industries that operate in highly competitive environments have to face the complex problem of developing products of increasing quality at affordable costs. Such a challenge is typical in technological fields, where product innovation must be promoted with the highest priority in order to be successful under the competitive pressure of global markets. The considerations above fully apply to automotive manufacturers, who are forced to develop new or to renew existing products that meet conflicting demands from customers and regulatory bodies, in ever shorter timeframes. From an early design stage onwards, concept computer-aided engineering (CAE) techniques are needed to predict and optimize the vehicle performance (in terms of safety, crashworthiness, noise, vibration and harshness (NVH), environmental impact, acoustics, etc.) based on virtual optimization [1–3]. The concept phase is a highly strategic step in the vehicle development for automotive manufacturers [4–15]. It is crucial that the vehicle design engineers can rely on efficient methods for performance optimization in an early stage, when none or little geometric information is available. Still, at this stag\e, there is an imminent need for predictive methods that allow predicting the main trends in the vehicle performance, so that an initial vehicle concept design optimization can be performed that will result in an improved initial computer-aided design (CAD) model of the vehicle in a later stage of the development timeline.

In the field of NVH predictions, several classes of concept modeling approaches have been proposed, including methods based on predecessor FE models [9] and methods from scratch [10]. In this paper an innovative concept modification approach is presented, in which the early design choices are supported by a database of structural elements, consisting of variants of the original FE model that are created as to preserve the mesh-compatibility at each interface. Although the proposed approach can be used for any structural element (beams, panels, etc.), the focus of this paper is on automotive joints, since these elements can affect the NVH characteristics of vehicle bodies significantly. In 2000, Nikolaidis et al. [16] proposed the use of neural networks or response surface polynomials to rapidly predict the performance characteristics of automotive joints. In 2002, Lee et al. [17] presented a numerical approximation of the stiffness properties of thin-walled beam-jointed structures, with the aim of enabling a fast optimization of automotive joint structures.

The main goal of the research presented in this paper is to prove the feasibility of an innovative concept modeling approach. The proposed methodology is based on the idea of building a database of joint concept models, consisting of a set of structural variants of an existing automotive joint. By enabling an easy and reliable exchange of the joint alternatives that are stored in the database, vehicle concept modification and optimization studies can be performed. The concept database approach for NVH optimization of a

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vehicle body is pursued by building further on two starting blocks reported previously in the literature: the beam and joint concept modeling approach [9,10] and the wave-based substructuring (WBS) approach [18,19]. The beam and joint concept modeling approach is used to make a concept model consisting of beam elements and joint matrices. It results in efficient models for the use in the concept phase, but in the joint property calculation, some inaccuracies are introduced by "end effects", since the loose ends of isolated joints (as used in the calculation of the joint matrices) have different mechanical properties than a joint included in the assembly model. In order to take into account the local behavior of the joint's end-sections, the beam and joint concept modeling method can be combined with the WBS approach.

The wave-based substructuring (WBS) approach is a substructuring approach in which the deformation of the coupling interface is written as a combination of a set of basis deformations called "waves". Connections between substructures (normally defined in terms of interface DOFs) are replaced by connections between waves that impose the continuity of the displacements and forces. Often one can use (much) less waves than the number of physical interface DOFs, which results in a smaller-sized assembly definition. The waves can be obtained from modal displacement on subsystem or system level.

In the research presented in this paper, the WBS approach is used to enable a compact and accurate formulation of the connections between concept joints, which are statically reduced variants of the original joint and collected in a joint database, and the rest of the original vehicle FE model.

The outline of the paper is as follows. Section 2 summarizes the numerical methods used for creating the concept joint model and connecting it to the rest of the structure: the WBS technique and the Guyan reduction approach [20], and how they can be combined. In Section 3 the proposed concept modification approach is illustrated by analyzing a case study, in which one joint of a simple FE model is used to create a database of joint variants.

In Section 4 the feasibility of the database concept is investigated on an industrial automotive application case. An industrial vehicle body FE model is used, aimed at the conceptual modeling of four joints. The robustness of the nominal waves with respect to the joint modifications is analysed through static and dynamic comparisons between the variant concept and detailed models.

2. Theory: numerical methods

2.1. Guyan reduction

Guyan reduction, also known as static condensation, is a numerical method that enables the reduction of the FE model of a structure into a small-sized description, by representing it in terms of the stiffness and mass matrices condensed at a small number of boundary nodes.

For an arbitrary structure, the basic static FE matrix equation is given by

$$K \cdot x = F, \tag{1}$$

where K is the stiffness matrix, x is the nodal displacement vector and F contains the nodal loads.

Guyan reduction aims at getting a reduced representation of the model for substructuring purposes. It requires that two sets of DOFs are identified as follows:

- n_t junction (or boundary) degrees of freedom (DOFs), which are retained in the solution. This set of DOFs allows the FE connection.
- n_o internal DOFs, which are to be removed by static condensation.

For this partitioning, the matrix system Eq. (1) can be written as follows:

$$\begin{bmatrix} K_{oo} & K_{ot} \\ K_{to} & K_{tt} \end{bmatrix} \cdot \begin{bmatrix} x_o \\ x_t \end{bmatrix} = \begin{bmatrix} F_o \\ F_t \end{bmatrix}, \tag{2}$$

where subscripts t and o are used to designate the junction and the internal DOFs, respectively. From the first line of Eq. (2), the internal displacement vector can be determined as:

$$x_o = K_{oo}^{-1}(F_o - K_{ot} \cdot x_t). \tag{3}$$

By introducing the static reduction matrix $G_{ot} = -K_{oo}^{-1}K_{ot}$ and substituting Eq. (3) into the second row of Eq. (2), a matrix equation for the reduced system is obtained:

$$K_{tt,\mathrm{red}} \cdot x_t = F_{t,\mathrm{red}}, \tag{4}$$

where

$$F_{t,\mathrm{red}} = F_t + G_{ot}^T F_o, \tag{5}$$

is the reduced loading vector, while:

$$K_{tt,\mathrm{red}} = K_{to}G_{ot} + K_{tt}, \tag{6}$$

is the $n_t \times n_t$ reduced stiffness matrix.From a physical point of view, this matrix represents the stiffness values between each pair of junction DOFs.

The same transformation can be used to condense the mass matrix on the boundary DOFs, so that an efficient reduced representation of the structure, useful for dynamic analyses, is obtained. However, while exact for the stiffness matrix, the Guyan reduction is an approximation for the mass matrix. By reducing the mass matrix, it is assumed for the considered structure that inertia forces on internal DOFs are less important than elastic forces transmitted by the boundary DOFs. This is true for very stiff components or in cases where local dynamic effects can be ignored. Therefore, the accuracy of the result is case dependent. For typical automotive joints, the stiffness relations between the joint end-sections have a much stronger influence on the global behavior of the body than the exact distribution of mass along the joint. For this reason, Guyan reduction of the joint structure to its joint end-nodes is an appropriate choice to create a small-sized representation of the actual joint [10].

When using Guyan reduction in a substructuring context, each substructure model undergoes a static condensation, in which only the junction DOFs are retained in the solution. This way, a reduced FE representation of the initial substructure is created as a small superelement, consisting of equivalent stiffness and mass matrices.

2.2. Wave-based substructuring (WBS)

Substructuring methods are based on the domain decomposition of a complete (FE) structure into several substructures, which are independently solved at a sub-component level. An alternative representation of the original model is finally achieved by recombining the substructures into an assembly level model.

In substructuring, one approach is to use a reduced modal model of a structure by representing the physical DOFs of each substructure by a reduced number of interface DOFs, referred to as *generalized coordinates*. The linear combination of a set of structural modes with these generalized coordinates aims at efficiently describing the physical behavior of the substructure. Different modal-based methods have been reported [21–24], in which the generalized coordinates refer to set of vectors comprising a truncated set of natural modes (under some boundary conditions) and a set of static enrichment modes, which allow an accurate representation of the local flexibility at the connection interface. A typical static enrichment mode set consists of the so-called *attachment modes*: each deformation shape is calculated by applying a unit force on one junction DOF and a zero force on all the other junction DOFs. In the approach proposed by MacNeal [22] and Rubin [23], the enrichment modes, named *residual attachment modes*, are calculated by subtracting from the attachment modes the components that are linearly dependent on the normal modes.

Traditionally, one has to obtain an enrichment mode for each interface DOF onto the assembly-level component, which becomes prohibitively costly for complex industrial models with extensive interfaces between adjacent substructures (e.g., weather strips, spot welds between floor and body).

To address this, a wave-based substructuring approach has been developed [18,19], in which the deformation of the coupling interface is written as a linear combination of a set of basis functions called “waves”.

The calculation of the basis function requires a modal analysis on the nominal FE model, performed in free-free boundary conditions, so that 6 rigid body modes are also obtained. The mathematic treatment of a limited number of the first global modes, by means of the singular value decomposition method, enables the calculation of a set of orthonormal basis functions. In the WBS approach, connections between substructures (classically defined in terms of interface DOFs) are replaced by connections between wave DOFs that impose the continuity of displacements and forces. Often, much less waves than the number of physical interface DOFs are required for an accurate coupling of substructures, which results in a smaller-sized assembly definition. This greatly facilitates the reduction procedure for large components, since a much lower number of enrichment vectors must be calculated, now defined in terms of wave participation factors.

The wave calculation is case-dependent, but the nominal wave set is rather robust for typical applications, and can also be used to connect modified components into the assembly structure, as has been demonstrated in Donders et al. [18]. Therefore, after a nominal assembly model using WBS technology has been defined, local modifications on components in FE representation can be processed quickly to predict the updated structural dynamics performance.

2.3. Combining Guyan reduction and WBS

The innovation in this paper is the combination of joint concept modeling and wave-based substructuring technology, with the aim of limiting the loss of accuracy due to the ‘end effects’ that arise when the joint FE model is extracted from the full vehicle body model. A typical automotive joint is a thin-walled structure, formed by two or more panels that are assembled through spot-weld connections. When external loads and boundary reactions are applied to the end sections of the isolated joint to create the reduced model, the increased local flexibility causes an underestimation of the joint stiffness properties.

To overcome the ‘end effects’ in joint concept modeling, one can define a wave-based substructuring (WBS) layer to connect the joint into the concept assembly model. Since the waves already include assembly-level dynamics information, the end effects are thus reduced. In addition, the WBS layer allows connecting also modified joints into the vehicle concept model, so that a database concept approach for vehicle body design modification and optimization in an early design stage becomes possible.

As reported in the previous section, classical Guyan reduction allows describing the stiffness and mass relationships between each pair of junction DOFs. The reduced stiffness representation is the one formulated in Eqs. (4)–(6).

By using the wave functions, the physical displacements and forces can be written as:

$$x_t = W \cdot p, \tag{7}$$

$$F_{t,\mathrm{red}} = W \cdot l, \tag{8}$$

where t denotes, in this specific case, the junction DOFs, $\boldsymbol{W}$ is the transformation matrix from the physical to the wave coordinates, p and l are the wave-displacements and wave-forces respectively, i.e., displacements and forces at junction DOFs in waves coordinates [18]. Thus, Eq. (4) can be rewritten in terms of wave-displacements and wave-forces as:

$$W \cdot l = K_{tt,\mathrm{red}} \cdot W \cdot p, \tag{9}$$

By pre-multiplying both members in Eq. (9) by l and considering that the wave matrix contains orthonormal vectors, the following relationship between wave-displacements and wave-forces can be obtained:

$$l = K_{\mathrm{WBS\text{-}red}} \cdot p, \tag{10}$$

where the stiffness matrix for the reduced system $K_{\mathrm{WBS\text{-}red}}$ is given by

$$K_{\mathrm{WBS\text{-}red}} = W^T \cdot K_{tt,\mathrm{red}} \cdot W. \tag{11}$$

By introducing wave relations along all joint interfaces in a body concept model, and subsequently obtaining the reduced joint models through a Guyan reduction for the set of wave degrees of freedom at the boundary, an efficient body concept model for joint modification and optimization is achieved. The procedure for the WBS-reduced formulation can be then summarized as:

- Define a beam and joint layout for a structure.
- Calculate a limited number of global modes at the physical end nodes of the joints.
- From these global modes, calculate the wave functions at the joint end sections. Effectively, one writes the physical interface DOFs of the joint as a linear combination of a few wave DOFs.
- Perform a Guyan reduction on the wave DOFs. This not only yields a very small matrix model of the joints, but (since the waves contain global dynamics information) also increases the accuracy of the joint matrix to be used on a global vehicle body modification level.
- The reduced joints are then connected to the remainder of the vehicle along these wave DOFs connections.

In a next step, a database of joint alternatives can be obtained, by defining a series of variants of the original joint, in terms of either geometry or material properties, while the mesh compatibility at each interface is preserved. By re-using the nominal wave formulation layer, one can then connect these alternative joints into the vehicle concept model, to evaluate the structural performance. One thus obtains a database concept approach for vehicle body design modification and optimization in an early design stage.

3. Academic case

In this section an academic case study is analysed to illustrate the proposed concept modification approach. A joint of a simple FE model is isolated from the rest of the structure and a series of variants is generated. A concept joint database is created and tested by connecting each joint variant, statically reduced, to the surrounding FE elements through a number of waves that are previously computed on the original model.

3.1. Model description

Fig. 1 shows the FE model of an open structure, consisting of eight thin-walled beam-members, modeled by 4-node shell elements that have a uniform thickness of 1 mm and typical steel material properties.

The central joint, i.e., joint B in Fig. 1, is isolated from the surrounding four beams by duplicating the interface nodes (white

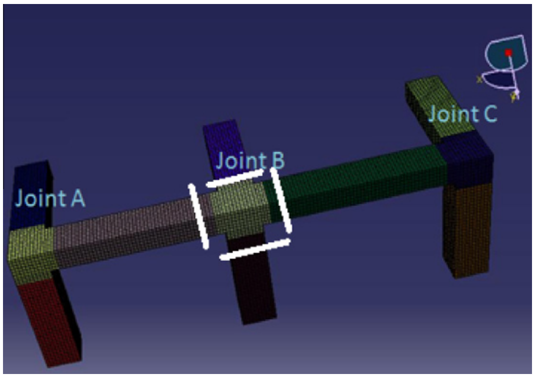

Fig. 1. FE model of the structure.

lines in the same figure). Twelve waves, computed through a modal analysis of the entire structure, connect each end-section of the joint to twelve scalar points that are used for the static reduction of the joint. A number of waves equal to 12 has been selected through a sensitivity analysis, which showed that increasing further the number of waves does not improve significantly the accuracy of the concept model in the frequency range of interest, i.e., 0–50 Hz in the example illustrated here. With the wave layer, one thus obtains a concept model, schematically represented in Fig. 2, where the equivalent stiffness and mass matrices of the joint are wave-connected to the end-nodes of the surrounding beams.

3.2. Concept database for the central joint B

A set of model variants is created by replacing the original FE model of joint B with a joint variant.

A number of joint variants have been created, in such way that for each joint variant, the mesh compatibility at the interface with the beams has been guaranteed. Two types of structural modifications have been applied:

- Uniform increase of the original thickness of joint's walls by 100% and 500%. This wide range of variation of the shell thickness aims at testing the robustness of the nominal waves with respect to substantial modifications; nevertheless, it is recognized that the thickness increment by 500% should be considered as an upper limit aimed at assessing the limitations of the methodology, and should not be seen as a recommended thickness increment in a vehicle design optimization process (given the high cost involved in adding design material, and hence weight to the vehicle).
- Insertion of a horizontal plate inside the central joint. The reinforcement plate has the same material and geometry characteristics of the other panels that form the joint's walls, and is located in the median horizontal plane of the joint, as illustrated in Fig. 3. Such a kind of structural modification is common in vehicle body manufacturing and is expected to significantly stiffen the joint and to modify the modal shapes at the joint-beam interfaces.

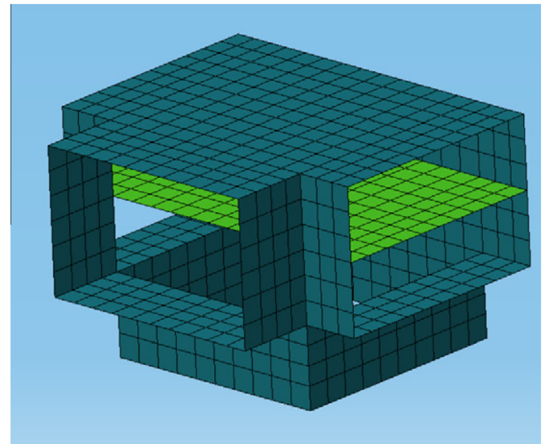

Fig. 3. Structural modification. Stiffening plate inserted into the central joint.

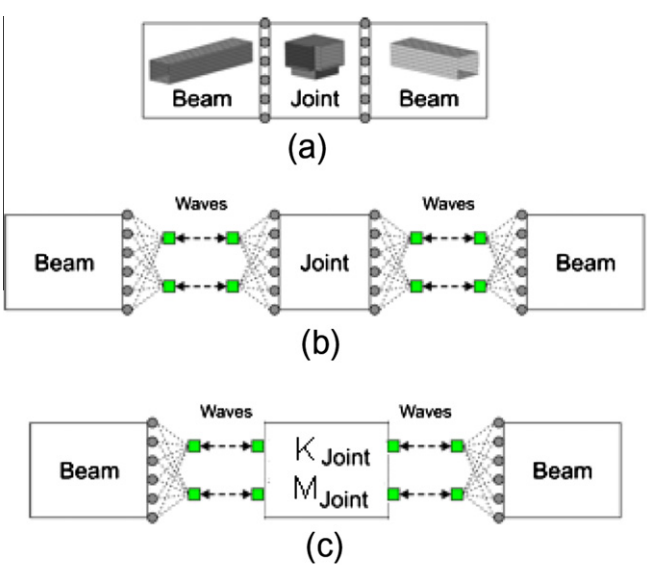

Fig. 2. Schematic representation of the original (a), wave-connected (b) and concept (c) model. In the last case, the equivalent mass and stiffness matrices of the joint are wave-connected to the rest of the structure.

In order to assess the impact of the joint modifications on the global modes of the structure, the variant models are compared with the nominal model in terms of natural frequencies. The results are summarized in Fig. 4. Among others, it can be seen that a 500% increase of the thickness is the modification with the highest impact.

As a next step, concept modifications are performed. For each variant, a concept model is created where the modified joint is statically reduced and connected to the rest of the structure by using the original waves.

The concept models are compared with the corresponding detailed models, i.e. the models in which the nominal joint is replaced by the detailed FE model of the joint variant, with the aim of analyzing the approximations involved by the use of the waves computed on the nominal model. The results are illustrated in Fig. 5, where the percentage errors on the estimation of the first

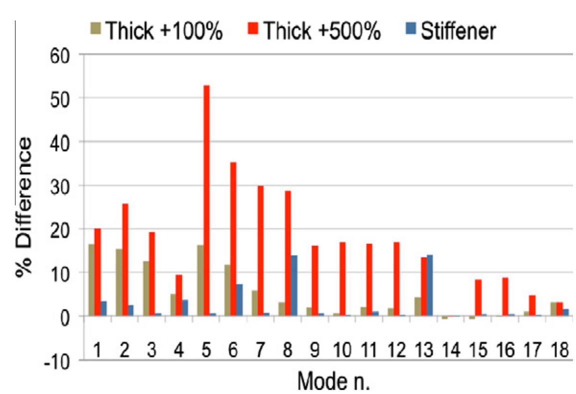

Fig. 4. Variations of the first 18 natural frequencies of the entire structure due to the joint modifications.

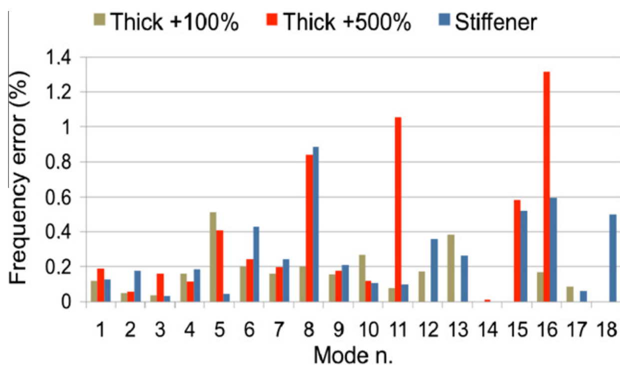

Fig. 5. Dynamic comparison between variant concept and detailed models.

18 natural frequencies are shown. It can be seen that the concept model slightly overestimates the eigenfrequency values of the full FE reference model in a systematic manner (since the introduction of the wave layer with a low number of waves introduces a stiffening effect), but the dynamic predictions are sufficiently accurate for concept design studies.

The dynamic comparison between the detailed and the concept models does not show significant errors, especially in the frequency range in which the waves have been calculated: for the first six natural frequencies, the maximum error is lower than 0.5%. In the model with a +500% variation of the thickness and in the model with a stiffener inside the joint the maximum errors are lower than 1.4% and 1.0%, respectively.

4. Industrial case study

In this section, the feasibility of the database concept is investigated on an industrial automotive application case. An industrial vehicle body FE model is used, aimed at the conceptual modeling of four joints.

4.1. Model description

Fig. 6 shows the FE model of a vehicle Body in White (BIW), predominantly formed by appropriately shaped panels that are modeled with linear shell elements and assembled with each other by means of spot weld connections. In an industrial BIW vehicle model, a joint is a part of the structure where beam-like structures (generally with different material and geometrical properties) converge.

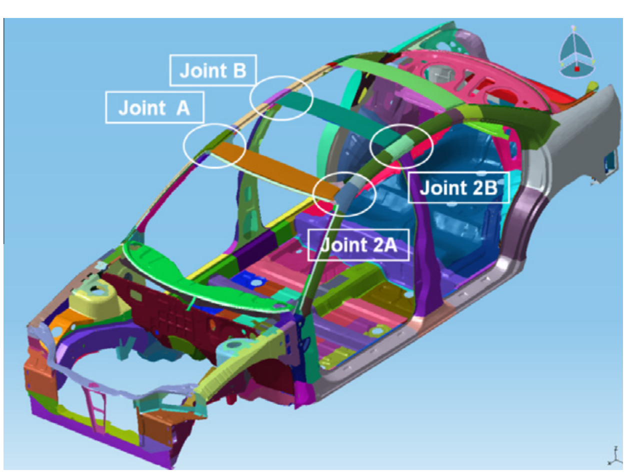

Fig. 6. FE model of a Chrysler Neon BIW.

In order to prove the feasibility of a concept joint database in an industrial context, four joints, labeled in Fig. 6 as Joint A, Joint B, Joint 2A and Joint 2B, are analyzed in this work. The four joints are the right and left A-pillar and B-pillar to roof joints and are symmetrical with respect to the longitudinal direction of the vehicle body. These are key joints in automotive engineering; moreover the choice of joints on both sides is motivated by the notion that realistic modifications in the vehicle design engineering process are typically made in a symmetric manner, since the body performance on left/right side in terms of static strength, dynamic vibrations and mode shapes, as well as crashworthiness, is typically desired to be equal on both sides of the longitudinal symmetry plane.

A total of fourteen cross sections are identified at the interface between each joint and the surrounding beam-members. In each of them, twelve waves are calculated through a modal analysis of the vehicle body.

4.2. Static and dynamic load cases

In line with the standards used by automotive manufacturers, two indicators of the full-vehicle static behavior are defined: the bending and torsion static stiffness. These indicators are evaluated for both the original and the concept body model by using the two load cases shown in Fig. 7.

In both cases, the vehicle body is clamped at the rear suspension locations, while two vertical forces are applied at the front suspensions (denoted as points A and B in Fig. 7). Based on the estimation of the vertical displacements v_A and v_B at the excitation points, the bending and torsion stiffness K_b and K_t of the vehicle body are determined as:

$$K_b = \frac{2FL}{\alpha_b}, \tag{12}$$

$$K_t = \frac{F \cdot W}{\alpha_t}. \tag{13}$$

Here, F denotes the amplitude of the vertical forces, L is the wheelbase and W denotes the width of the car, while α_b and α_t are the deflection angles in the bending and torsion load case respectively, given by

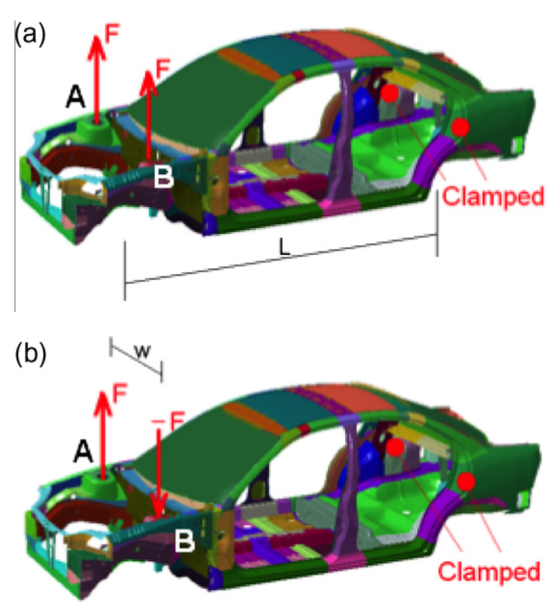

Fig. 7. Static load cases used to compare the original and the concept models in terms of bending (a) and torsion (b) stiffness.

$$\alpha_b = \arctan\left(\frac{v_A + v_B}{2L}\right), \tag{14}$$

$$\alpha_t = \arctan\left(\frac{v_A - v_B}{W}\right). \tag{15}$$

In addition to the static performance, also the dynamic performance will be considered, in terms of both natural frequencies and mode shapes. To compare the mode shapes, the modal assurance criterion (MAC) is used. The MAC correlation between two generic modal matrices V_1 and V_2, is given by

$$\mathrm{MAC}_{i,j} = \frac{\left(\{V_1\}_i^H \cdot \{V_2\}_j\right)^2}{\left(\{V_1\}_i^H \{V_1\}_i\right) \cdot \left(\{V_2\}_j^H \{V_2\}_j\right)}, \tag{16}$$

where the subscripts i and j, with $i, j = 1, \ldots, N$ and N the number of modes stored in V_1 and V_2, identify a generic mode shape of model 1 and model 2 respectively, while the superscript H denotes the conjugate transpose of a vector.

4.3. Model modification

The proposed method is based on a modification approach that consists of modifying an existing FE model (predecessor model) with the aim of achieving a variant version, which presents the wished structural behavior.

Thus, the first step consists in defining a series of geometric modification to be applied to the joints chosen as target for the concept modeling, while the mesh compatibility at each interface must be preserved.

A structural modification of the four joints is made by increasing the thickness of all panels by 50%.

First, the influence of this variation is assessed through both static and dynamic comparisons between the variant model (full FE model with the four joint variants) and the nominal vehicle FE model.

The influence of the applied modifications on the global static behavior is summarized in Table 1, where a stiffening effect is shown for both load cases. The applied modification on the joints strongly affects the torsion stiffness (+8.07%) while results in a (much) smaller effect for the bending stiffness (+0.18%).

With the aim of assessing the influence of the proposed joint modification on the full vehicle dynamic behavior, FE modal analyses are performed on both the original and the modified model. A comparison between the two models can be made in terms of natural frequencies of the full vehicle body and MAC.

The results of the dynamic comparison between the original and the modified model are summarized below.

Fig. 8 shows the difference between the two models in terms of global frequencies in the range 0–50 Hz, where six global modes are observed. The maximum percentage difference between the two models is +3.29%.

The effects of joints' thickness increase on the full vehicle modal shapes are also investigated through a MAC correlation between the original and the modified model. An average MAC value of 97% has been calculated by considering the modal correlation factors for the six global modes of the vehicle in free-free conditions

Table 1
Influence of the joints modification on global statics: bending and torsion stiffness.

	Bending		Torsion	
Stiffness	Original model	Modified model	Original model	Modified model
Nm/rad	4.994E4	5.002E4	1.457E5	1.572E5
Δ %	–	**+0.18**	–	**+8.07**

Bold values indicates percentage difference.

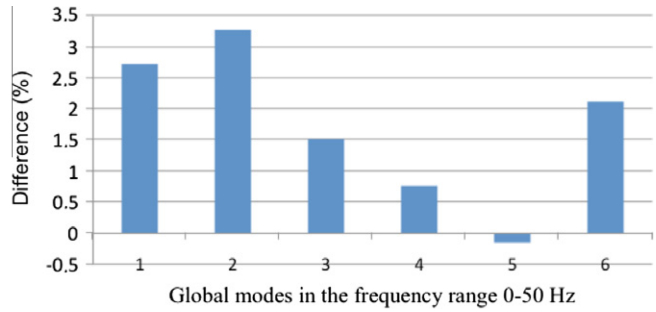

Fig. 8. Variations of the global frequencies of the vehicle body in the range 0–50 Hz due to the modification of the four joints.

in the frequency range 0–50 Hz (i.e. the first 6 natural modes). It is noted here that the 6 rigid body modes have been excluded from the MAC comparison.

The observed changes prove that the proposed model modification, even if local, has a significant impact on the static and dynamic behavior of the full vehicle.

4.4. Concept model

Now that the static and dynamic impact of the modification has been assessed, the next step is set with the creation of a WBS-concept representation for the joints. A concept model of the vehicle BIW is created, with the four joint variants (with an increment of thickness by 50%) statically reduced and connected to the rest of the body by using 12 waves (6 from rigid body modes and 6 from

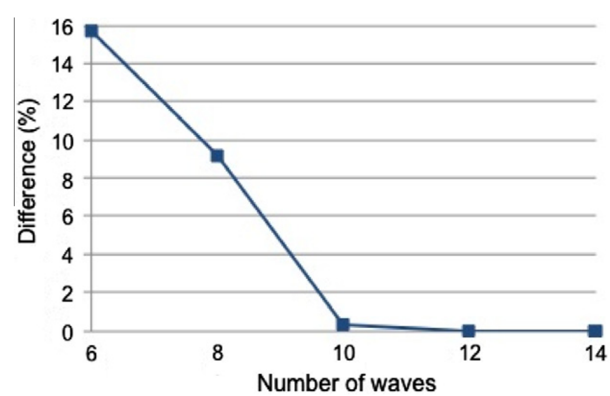

Fig. 9. Comparison between the original and the wave-connected models in terms of eigenfrequency for the highest order mode for different number of waves.

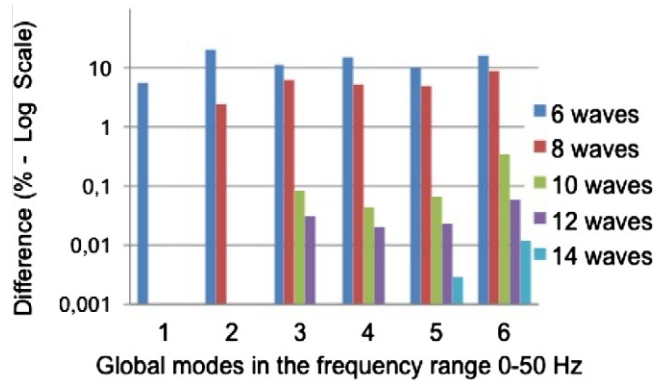

Fig. 10. Comparison between the original and the wave-connected models in terms of global mode frequencies in the range 0–50 Hz for different number of waves.

Table 2
Comparison between the variant detailed and concept models in terms of bending and torsion stiffness.

	Bending		Torsion	
Stiffness	Variant model	Concept model	Variant model	Concept model
Nm/rad	5.0023E4	5.0026E4	1.5719E5	1.5742E5
Δ %	–	**+0.006**	–	**+0.146**

Bold values indicates percentage difference.

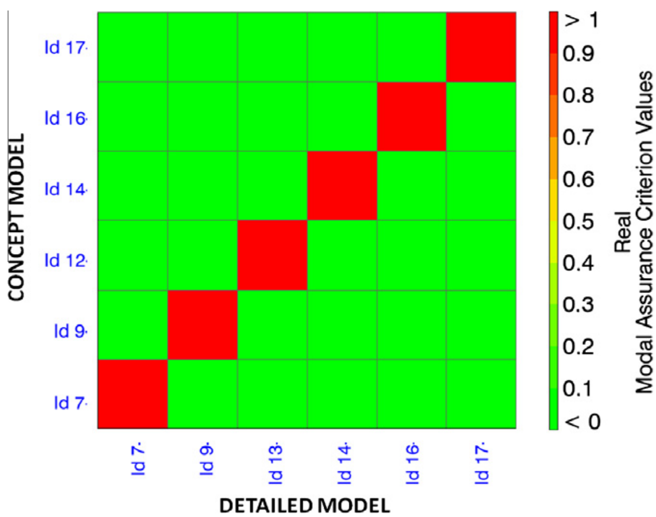

Fig. 11. MAC correlation between the variant concept model and the variant detailed model.

natural modes), calculated by using the nominal model. Since the number of waves needed to achieve an accurate representation of connection interface is case-dependent, also in this case the number of waves has been chosen by means of a sensitivity analysis. The results are illustrated in Fig. 9, where the difference between the original and the wave-connected models in terms of eigenfrequency for the highest order mode, i.e. the 6th global mode of the vehicle body, is shown as a function of the number of waves. Fig. 10, where the same comparison is made by taking into account all global modes in the frequency range 0–50 Hz, proves the reliability of the wave-connected model with 12 waves, being the maximum difference between this model and the original one lower than 0.1%. It is noted that the wave-connected model with 14 waves allows predicting the frequencies of the first 4 global modes, which are the normal modes no. 7, 9, 13 and 14 of the model, exactly. This is consistent with the theory of WBS technique, which provides exact solutions for the modes of order lower or equal to the number of waves.

The concept model of the vehicle body, containing the reduced models of the four joint variants, is validated through a comparison with the corresponding detailed model, in which the modified joints are included in the detailed FE model. For static validation purposes, the two global load cases shown in Fig. 7 are analysed. The numerical results are reported in Table 2.

In both cases, the concept model shows an excellent accuracy, especially under bending where an error of 0.006% has been calculated. Both load cases show a very small stiffening effect, mainly due to the use of a limited number of waves, twelve in this specific application, for substructure connections. However the prediction accuracy is very high, by all means accurate enough to allow concept modification and optimization of the joints.

The dynamic performances of the concept model have been assessed by means of a modal analysis in free-free boundary conditions.

The correlation between the modal shapes of the detailed and the concept model is reported in Fig 11, where the MAC values for the global modes in the frequency range 0–50 Hz are shown.

The average diagonal MAC value is 0.98, which proves that a good MAC correlation is obtained in the frequency range of interest. The minimum correlation values are achieved for the two global modes, shown in Fig. 12, that have significant modal displacements in the region of the roof rails.

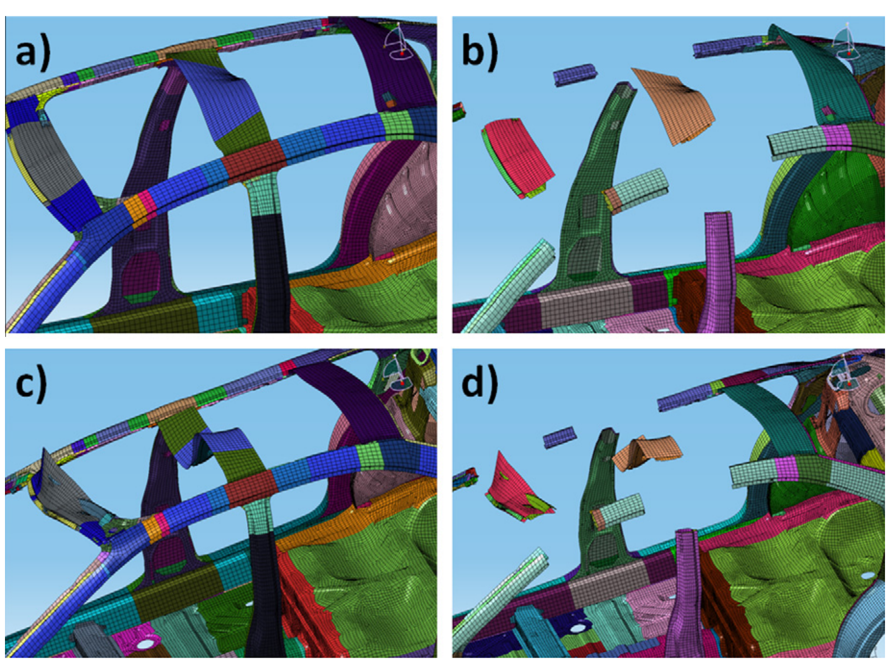

Fig. 12. Global modes with significant modal displacement in the region where concept joints are connected to the roof beams. Detailed model (a and c) vs concept model (b and d).

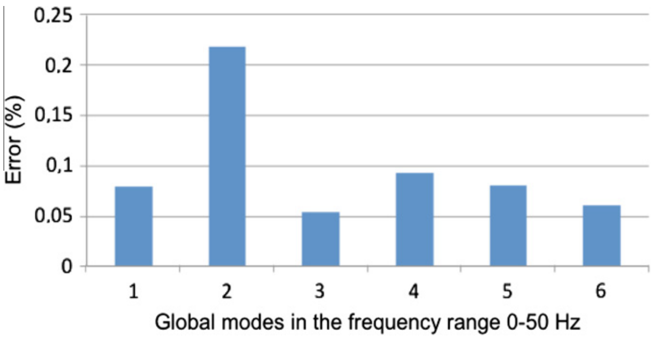

Fig. 13. Comparison between the variant concept and the variant detailed models in terms of global frequencies in the range 0–50 Hz.

Fig. 13 shows the comparison between the two models in terms of natural frequencies. A maximum error of +0.22% can be observed for the second frequency of the full vehicle body.

5. Conclusion

A methodology for the concept design of vehicle joints has been presented in this paper. The main innovation is the proof that a 'wave connection' can be used to deal with the 'end effects' in the reduced beam and joint modeling approach. Subsequently, the feasibility of a 'database concept' has been demonstrated, which can be used for the concept modeling of automotive bodies. The approach can be used for both static and dynamic concept modeling and simulations.

The feasibility of the database concept has been verified on a simple model with only three joints and eight beams. An industrial case study has been analyzed as well to demonstrate the proposed approach. It can be concluded that excellent static and dynamic concept predictions of joint modifications can be obtained with the proposed approach. Accordingly, it can be concluded that the approach can be used in the concept design phase of a new vehicle, starting from an existing one.

The proposed approach has been tested only for joints replacement, but it can be applied also for the replacement of other substructures, such as beams and panels. Future work could be oriented to the identification of a set of joint physical parameters useful for the automatic generation of model variants, and to the identification of a range of variability of these parameters.

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