The Effect of Spot Weld Failure on Dynamic Vehicle Performance

Stijn Donders[1], Marc Brughmans[1], Luc Hermans[1], Nick Tzannetakis[2]
[1] LMS International, Interleuvenlaan 68, 3001 Leuven, Belgium;
stijn.donders@lms.be, marc.brughmans@lms.be, luc.hermans@lms.be
[2] Noesis Solutions, Interleuvenlaan 68, 3001 Leuven, Belgium;
nick.tzannetakis@noesis.be

ABSTRACT

Spot welds are the dominant joining method in the automotive assembly process. As the automated assembly process is not perfect, some spot welds may be absent when the vehicle leaves the assembly line. Furthermore, spot welds are highly susceptible to fatigue, so that a substantial number may fail during the vehicle lifetime.

The scope of this paper is twofold. First, the impact of spot weld quality and design on a vehicle's functional performance is reviewed, addressing strength and stiffness, NVH and durability. The overview briefly covers both experimental tests and predictive Finite Element (FE) modeling approaches, explains the complexity of a spot weld design problem and discusses optimization strategies. Second, an industrial robustness study is presented that assesses the effect of spot weld failure on dynamic vehicle characteristics. Damaged models are generated automatically, by breaking a subset of the vehicle's spot welds, using a (weighted)-uniform selection probability. Monte Carlo simulations are then used to assess the scatter on dynamic vehicle characteristics.

INTRODUCTION

The role of Computer-Aided Engineering (CAE) in the automotive industry is rapidly increasing. Functional performances (NVH, durability, ...) are fine-tuned on the basis of numerical predictions, so that the expensive physical prototyping phase can be shortened considerably. Traditionally, optimizing a vehicle body starts with improving the fundamental torsion and bending frequency. These dynamic characteristics should be robust to failure of spot weld connections, thousands of which are present in a typical vehicle body.

The **first part** of this paper overviews the use of spot welds in the automotive industry. Section 1 deals with the resistance spot welding procedure and typical performances of spot welds. Section 2 describes small-scale experiments and real-life testing of spot weld characteristics in terms of strength and stiffness, NVH and fatigue life, and highlights the complexity of spot weld (layout) design. Section 3 describes a selection of Finite Element models that are used to predict a spot weld's functional performance with numerical simulations, and addresses the benefits and difficulties of optimization on the basis of FE models.

During the vehicle lifetime, manufacturing inaccuracies, minor accidents and fatigue failures may result in deterioration or even absence of a substantial number of spot weld connections. Also, in a CAD model transferred to a CAE department, some spot welds might be omitted or forgotten. The **second part** of this paper presents an approach to assess the robustness of dynamic vehicle characteristics to this breakage or absence. Automated procedures have been developed, to break a number of spot weld elements with highest strain energy in the nominal (undamaged) model, and also to randomly break a number of welds, either with a uniform probability or with a weighted-uniform probability. The presented application allows performing Monte Carlo simulations to assess the effect of random spot weld failure on the dynamic vehicle characteristics. Section 4 explains both the input file creation routines and the process flow of the required computations. Key results are given in Section 5.

1. SPOT WELDS IN VEHICLES

Resistance spot welding (RSW) emerged in the 1950s, and is nowadays the predominant assembly technique in the automotive industry. The vehicle components (body in white, cradle, doors, ...) are made of thin metal sheets that are connected with spot-welded joints (or simply **spot welds**); see the example [1] in Figure 1. To create a spot weld, two or more metal sheets are pressed together by electrodes, and an electric current is passed through. The resistance of the metal generates heat, and the sheets are welded together by means of local metal fusion: a spot weld has been created. No welding material is added in this process. Three regions are identified in a spot weld: a **weld nugget** with cylindrical shape, a heat-affected zone (HAZ) and the base material sheets [2]. These regions have different material properties. For example, the yield stress in the nugget is up to three times higher than in the base material [3], and the plastic properties of the HAZ are non-homogeneous [4].

Figure 1 – A spot-welded vehicle body-in-white [1].

Due to the applied pressure by the electrodes during the welding, the thickness of the nugget is often less than the thickness of the two metal sheets. This so-called nugget indentation is typically not significant for plate thickness up to 1 mm, but is more pronounced when thick plates are assembled. Stress concentration may occur at the edges where a change of thickness takes place, which may result in crack initiation [3]. Furthermore, the transient heating and cooling results in hardening of the material, and a pre-stress may remain after cooling.

A typical vehicle body-in-white is made of steel sheets and contains several thousands of spot welds. The optimal diameter and distance between two successive spot welds are determined by the sheet thickness. The diameters range from 3 to 7 mm, with a mean of 6 mm [5]. The manufacturing practice of spot welds in the vehicle assembly process poses constraints on the spot weld layout design, as not all positions can (effectively) be reached. Note also that the assembly process is not perfect: sometimes a few spot welds are even missing or broken right from the beginning of the vehicle life.

2. FUNCTIONAL PERFORMANCE TESTING

2.1 Strength and Stiffness

Two fracture modes in spot weld strength tests are distinguished [3][6]:

- **Interfacial mode** (or **nugget fracture**): fracture of the weld nugget through the plane of the weld – the dominant failure mode for small diameter spot welds.
- **Nugget pullout mode** (or **sheet fracture**): fracture of the sheet around the weld; the nugget remains intact – dominant for large diameter spot welds.

Spot welds for automotive applications should have a sufficiently large diameter, so that **nugget pullout mode** is the dominant failure mode. Interfacial mode is unacceptable due to its low load carrying and energy absorption capability. Strength tests with different static loading were performed in [3], to reveal the failure mechanisms for the lap-shear geometry and the cross-tension geometry in Figure 2.

- In the **lap-shear** geometry, a shear load is applied. The weld nugget rotates to align with the loading line. When the load is increased, localized necking occurs (see Figure 3, left, top). Fracture initiates at one of the localized necking points, when the ductility of the sheet metal is reached (see Figure 3, left, top). Although a shear load is applied, the failure mechanism is **tensile**.
- In the **cross-tensional** geometry, a normal load is applied. The failure mechanism is through-thickness **shear** around the spot weld nugget (see Figure 3, right).

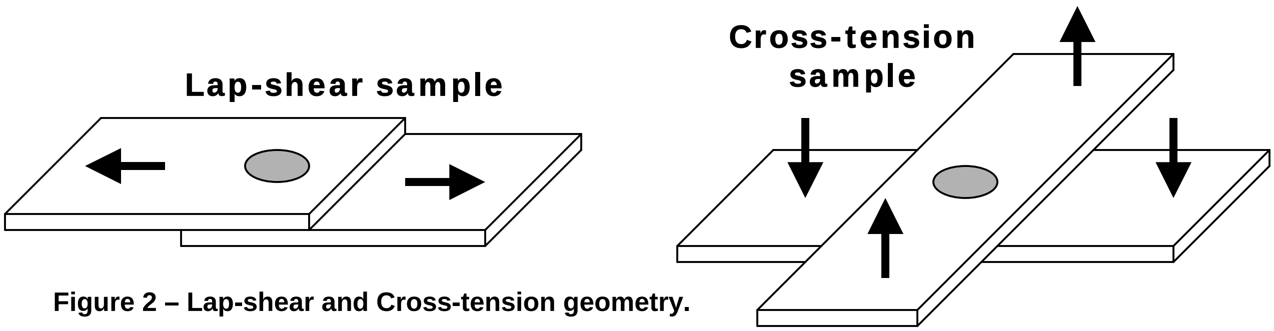


Figure 2 – Lap-shear and Cross-tension geometry.

A metal beam can endure higher tensile loads than shear loads. Given the above failure mechanisms, a spot weld connection **reverses** these properties: a spot weld under cross-tension load fails at a lower load than a spot weld subject to lap-shear load. This is well known in industry, and has been verified experimentally in [3]: the failure load of the cross-tension geometry is 74% of the failure load in the lap-shear geometry.

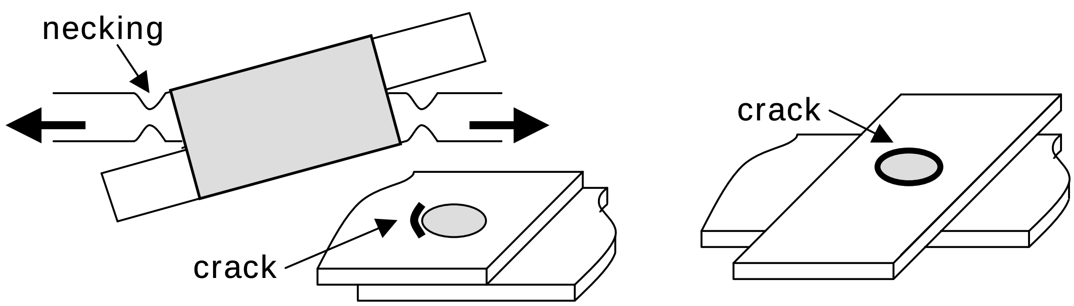


Figure 3 – Failure mechanism for lap-shear sample (left) and cross-tension sample (right)

Vehicle **stiffness** is typically assessed on a global level. The global stiffness of a vehicle is determined by the stiffness of the components and the stiffness of the spot weld connections. The global stiffness is mainly determined by the spot weld number and size; the positions of individual spot welds have a lower contribution [8].

2.2 Noise, Vibration and Harshness (NVH)

A vehicle's NVH properties are typically studied on a global level. Individual spot welds can transmit dynamic **push and pull forces, shear forces and shear moments**, see Figure 4. Size, number and position of spot welds therefore influence the vibro-acoustic performance (modal basis, frequency response functions) of the vehicle. In general, the role of mechanical joints in vibration transmission and attenuation is important and quite complex. In [7], high-frequency vibro-acoustic properties of (spot-welded) joints are derived. Joints are considered as a mechanical filter, with frequency-dependent transmissibility, reflectivity and absorption properties of the vibration energy. The method is based on a statistical approach of vibration analysis of flexurally vibrating thin plates, which requires averaging the energy in frequency bands. A minimum of 5 modes per band is required [7], which prohibits the method's applicability to the low-frequency domain. The method is demonstrated on a large plate structure that contains a spot weld (or one of 4 other types of joints).

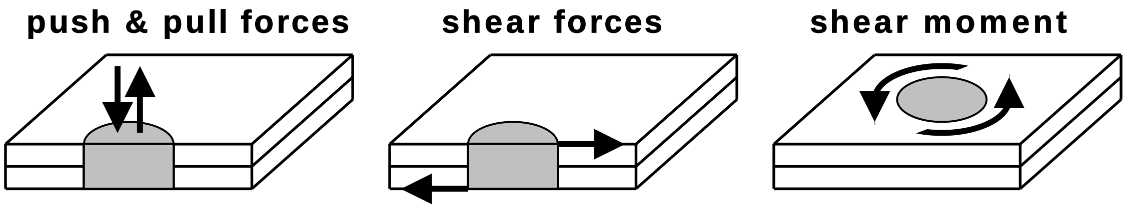
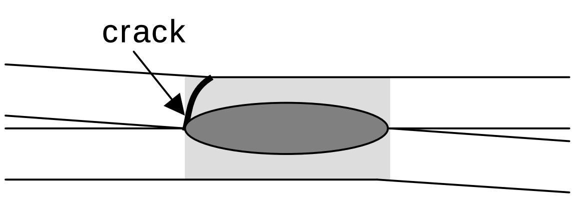


Figure 4 – Forces and moments that can be transmitted by a spot weld.

2.3 Fatigue

Durability is the functional performance that is most sensitive to spot weld quality and layout. Most fatigue failures in a vehicle body structure occur at or around the spot welds. The fracture modes are the same as explained Section 2.1 for strength analysis [6]: **sheet fracture** is the dominant fatigue failure mode in vehicles, as spot welds in vehicles have a sufficiently large diameter. As rule of thumb for fatigue failures [9], a spot weld that connects two sheets with thickness t has a large diameter d when $d \geq 5 \cdot t^{1/2}$ [mm].

The fatigue life is divided in two parts: crack **initiation** and crack **propagation**. Figure 5 shows a model and a photo of a typical sheet fracture fatigue crack, which initiates near the notch root at the interface between lower and upper sheet (where the maximum cyclic principal strain range occurs). It then propagates along the thickness direction of one sheet, and when the surface has been reached, it propagates further along the spot weld's perimeter. Fatigue failures occur at significantly **lower loads** than the critical loads attained in strength tests. Significant yielding occurs in spot welds, even under relatively low loads. Application of numerous load cycles may result in fatigue failures.



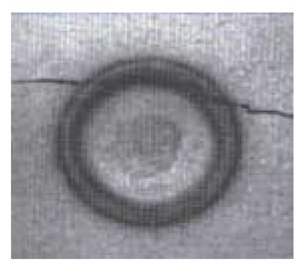


Figure 5 – A typical sheet fracture fatigue crack (left) and a photo of the crack propagation along the surface of the base sheet material (right)

While stiffness and NVH properties are characterized on the global vehicle level, fatigue is a **local** phenomenon. Fatigue life prediction is generally based on local structural stresses and strains near the spot welds. Four categories are distinguished [10][11]. The last two require calculation of equivalent stress intensity factors [4][12], taking the effect of material ductility, geometry and loading type on the fatigue failure into account.

- Stress-life method: relates the stress level S with the number of load cycles N (in an S–N curve)
- Strain-life method: relates the strain level ε with the number of load cycles N (in an ε–N curve)
- Equivalent stress intensity factor life method (ΔK_{eq}–N curve), relating fatigue life to stress intensity factors.
- Fracture mechanics (da/dN–ΔK curve), relating the crack length propagation da per load cycle to the stress intensity factor ΔK.

2.4 Complexity of spot weld design

The functional performance of a **single** spot weld is related to many variables, e.g. residual stress, material inhomogeneity, welding parameters, thickness, nugget size, material properties of heat-affected zone and base material, applied coatings and adhesives, loading ... A vehicle is assembled with a few **thousands** of spot welds, which highly increases the complexity of the design problem. How many spot weld connections should be made, and where should they be placed in a full vehicle body? Note that the spot weld layout should obey the constraints posed by the automated manufacturing process. Initially, a spot weld is placed at a location or not, which yields a discrete contribution to the global structure. During the product lifetime, intermediate connection properties become possible: a spot weld with a propagating fatigue crack is not fully connected, but surely not fully loosened. It becomes clear that even with a given number of spot welds, an almost infinite number of configurations exist to assemble the full vehicle model from all its components. The fact that the optimal number of spot welds is unknown increases the complexity even more.

Parameter **uncertainty and variability**, for instance introduced by the manufacturing process, further increases the complexity to the design problem. Two components are never connected with a series of completely identical spot welds: there will be deviations from the design locations, variations in spot weld area, thickness, stiffness, lifetime,... An optimal spot weld design should be **robust** to realistic variability in the initial layout and properties, and also to the expected deterioration of spot weld quality during the product lifetime – the breakage of a substantial number of welds.

Conventional practice in automotive industry is to perform extensive tests on spot-welded samples, and also to test full vehicle models to find (regions of) critical spot weld connections, and further evaluate variants of subsystems to establish a sufficient knowledge database for (concept) design purposes and improve the critical connections. When design directions are only derived from such experimental studies, clearly a lot of time and money is required to build physical prototypes and converge to a suitable design. As argued above, there are simply too many variables to consider, which makes it infeasible to experimentally verify all possible alternatives. It is already highly time consuming and very costly to manufacture and test a few physical assembly prototypes. Nowadays, the complexity of the design and time constraints in the development cycle are partially overcome by applying a **safety margin** in the number of spot welds: with a sufficiently high number, even the breakage of a substantial percentage during the product lifetime will not critically affect the full vehicle performance. This approach is suitable to guarantee the average lifetime of a family of identically manufactured vehicles, but there is clearly a need for **improvements**. In the mass production process, even a small reduction of the number of spot welds, enabled by an improved layout and quality, can mean a great saving in production cost and may allow reducing the assembly complexity. Spot weld (layout) optimizations in a numerical **modeling phase** are naturally much faster and cheaper. Spot welds modeling approaches are briefly reviewed in Section 3.

3. SPOT WELD MODELING

3.1 Accurate finite element models

This paper considers only **Finite Element** (FE) modeling, widely used in the automotive industry to assess the vehicle behavior in the low and medium frequency range. Accurate predictions can only be obtained when realistic spot weld connections are included in the vehicle finite element model. The detail of finite element models is a trade-off between accuracy and computation time. A balance should be sought such that **convergence** is achieved: the functional performance prediction should not significantly change when the mesh is further refined. The most straightforward spot weld modeling approach is to use coincident nodes at the boundary between two welded parts. This fails to take spot weld dimensions, stiffness and force-propagating properties into account, giving rise to (large) prediction errors. It is therefore generally agreed that the spot welds must be represented by FE models as well. As it is not feasible to model each spot-welded joint in detail, the same **simplified model** should be used to represent each spot weld connection, in such way that accurate predictions are obtained for the functional performance of interest. Stiffness and NVH predictions, determined on a full-vehicle level, can be obtained with a coarse spot weld model, while a much finer mesh should be used for spot weld fatigue analysis, as detailed local stresses must be evaluated. On an industrial point of view, the simplified spot weld models should not only yield accurate predictions in a limited amount of time. It should also be possible to **conveniently** and **automatically integrate** the spot weld models in the vehicle model, **replace components** and generate new spot weld connections. This allows performing efficient concept modeling and product refinement studies.

3.2 Spot weld models for strength and stiffness

Structural joint stiffness is predicted in [13] with a variety of spot weld models. The **single rigid bar** model, where nugget stiffness is added in the base material plane by means of nugget shell elements or with nugget spoke bars (Figure 6a), the **multiple bar** model, using rigid bars along the circumference of the weld nugget (Figure 6b), and the **solid nugget** model, where solid elements fill the entire nugget (Figure 6c). Some specimen beam structures are defined and five different load cases are applied. It is concluded that all types yield good accuracy in terms of structural stiffness under tension, out-of-plane torsion and bending loads. The single rigid bar model is highly inaccurate when in-plane torsion and shear loads are applied; the solid nugget model is then the best choice. Note that components cannot be meshed independently with these models: the spot welds affect the component meshes (both nodes and elements), which largely prohibits automated application.

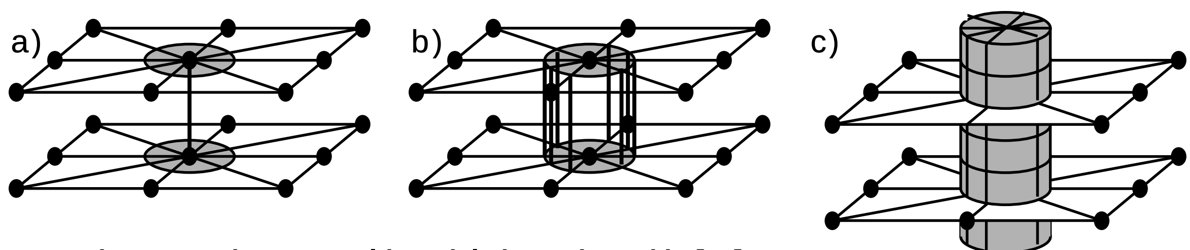


Figure 6 – The spot weld models investigated in [13].

3.3 Spot weld models for NVH

A study of four spot weld models for **NVH predictions** is given in [5]. Results with MSC.NASTRAN [14] are presented, and it is mentioned that the same trends have been observed with ABAQUS. Two **punctual** connection models use rigid bars in a geometry with non-coincident (Figure 7a) and with coincident (Figure 7b) component meshes. In addition, two **surface** connection models are investigated: the connection of coincident meshes with several rigid bars to represent the spot weld (Figure 7c), and the connection of two non-coincident meshes with the spot weld modeled as a HEXA solid element (Figure 7d, gray block). In the latter connection model – denoted as **HEXA spot weld model** [15] from here on – the link between the HEXA element and the shell nodes is performed with RBE3 interpolation elements (Figure 7d, dashed lines). The HEXA spot weld model is commercially available in LMS Virtual.Lab [16].

As a comparison study in [5], the 4 spot weld model types in Figure 7 are used to couple an academic (two plate coupling) and an industrial structure (a vehicle's front subframe). The convergence of the modal basis and frequency response predictions are investigated, and validated against experimental data. The main conclusion is that the **punctual connection models are not satisfactory**. Too flexible models are obtained, for which the eigenfrequencies do not converge due to singularities induced by the concentrated forces and moments that are generated on the connected components. Both the **surface connection models provide realistic NVH predictions**, as mesh convergence is obtained. The approach in Figure 7c is slightly more accurate when compared to experimental results. The HEXA spot weld model in Figure 7d is more sensitive to the mesh refinement – via the RBE3 interpolations, the solid element is directly linked to the shell elements and its stiffness radiates on the entire area of these shell elements; this area is mesh dependent [5]. A huge advantage of the HEXA spot weld model lies in the convenient industrial applicability. It allows coupling two components with non-coincident meshes: a component can be replaced, and new spot weld connections are easily generated without re-meshing the components.

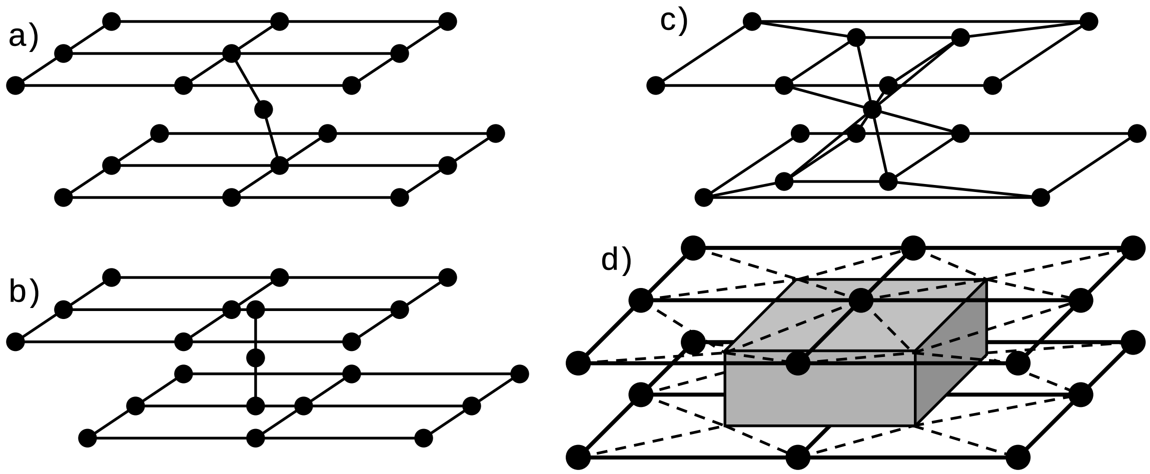
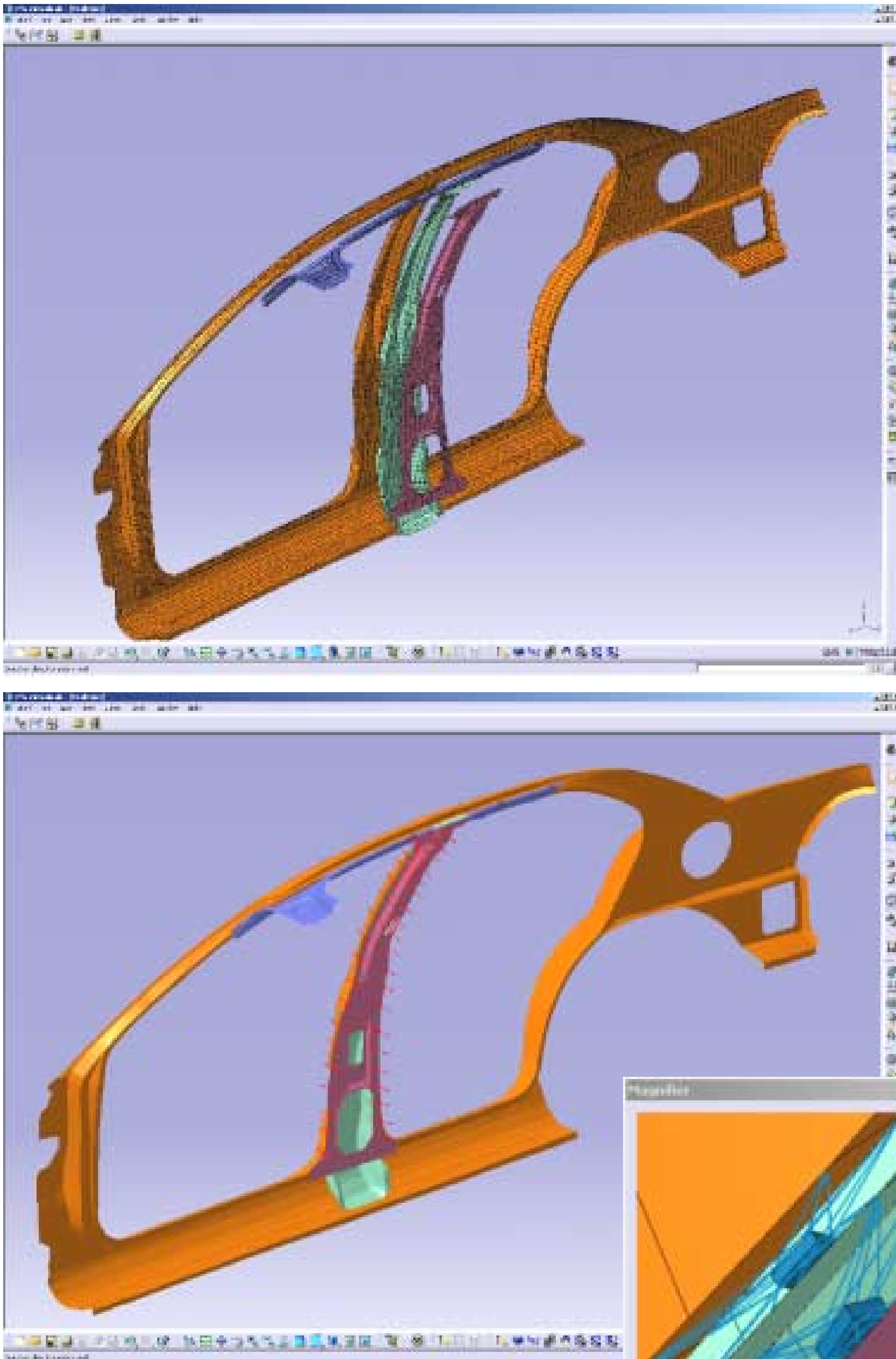


Figure 7 – The four spot weld models investigated for NVH predictions in [5].

In classical FE preprocessors, the spot weld locations are typically picked up from an ASCII file that is exported from the CAD environment that was used for designing the components. The spot welds are then created in the FE assembly of the meshed metal sheets. Over the last years, the tendency to **bring analysis to the designers** has lead to a new generation of software tightly connecting CAD and CAE in one environment with **full associativity between CAD and CAE models** [16][17]. This applies to the process of spot weld modeling as well, avoiding tedious spot weld data translation from CAD to CAE. A practical problem in the CAD-to-CAE transition is that sometimes designers do not put all spot welds in the CAD model, which means that automatic routines may fail and some components are not properly attached to each other. To alleviate this, the missing spot welds are created manually. The biggest problem with missing spot welds however is that they are often only discovered after running the entire model, and sometimes not discovered at all.

Figure 8 illustrates the spot weld assembly procedure for NVH in LMS Virtual.Lab [16]: Four metal panels (see Figure 8, top) are connected into a B-pillar. First, the spot weld locations are defined; these can be loaded from an ASCII file or manually created (see Figure 8, bottom for a visualization). Then, the spot weld model type is selected: either the HEXA model or the CWELD model [18]. The CWELD model is a dedicated spot weld element in MSC.NASTRAN: a special shear flexible beam-type element with two nodes and twelve degrees of freedom. This model has not been considered in this paper. It has been shown in [19] that the HEXA model yields more accurate NVH predictions – in line with the conclusions in [5] that a surface connection model is more accurate than a punctual connection model. In the example in Figure 8, the B-pillar has been connected with MSC.NASTRAN CHEXA elements (Figure 8, bottom, right). In LMS Virtual.Lab, the spot weld diameter can be derived from a built-in formula or a table based on the shell thickness. The height can be computed as the result of averaged thickness of the sheet panels or the distance between the sheet panels. Diameter and height can be manually altered. LMS Virtual.Lab also allows loading a full-vehicle assembly model that already contains the CHEXA or CWELD spot weld definitions. Locations and properties are automatically detected, and can easily be modified by the user.



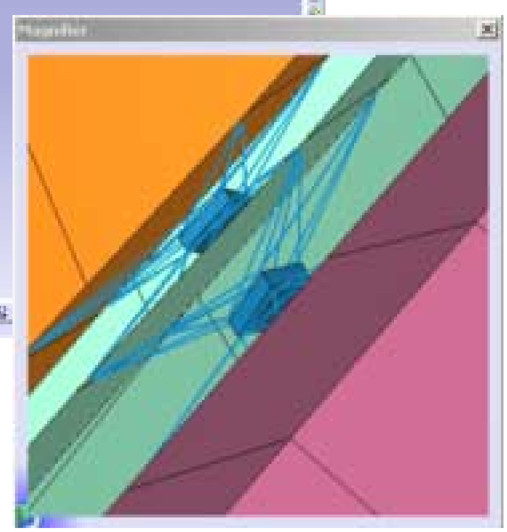


Figure 8 – LMS Virtual.Lab Spot Weld Modeling for NVH: B-pillar connection. Four panels are to be connected (top). First, the spot weld locations are defined (bottom). At these locations, HEXA spot welds are then created (bottom, right: a magnification of 2 HEXA spot welds).

3.4 Spot weld models for Durability

For NVH and stiffness predictions, the spot welds need to allow accurate predictions of the global stiffness. For a durability analysis the local behavior has to be considered. One approach to obtain accurate fatigue predictions for spot welds is to use a much finer mesh than used in models for NVH and stiffness predictions [4][6][9]. When each spot weld nugget is modeled with hundreds to thousands of solid elements, accurate local stresses can be predicted, from which accurate fatigue life predictions can be derived. When such fine spot weld models are used in a vehicle model, one should also edit the mesh in the vicinity of the spot welds, so that the radial stresses are accurately predicted. From an engineering point of view, it cannot be justified to use a very fine mesh for all spot welds in a vehicle body. Not all spot welds are critical, so that a lot of computation power is wasted to accurately predict the stresses at irrelevant locations.

From a vehicle development perspective, it would be ideal to perform the durability analysis on the same shell-based meshes as used for NVH analysis. No re-meshing is then required for durability purposes, and the coarse spot weld models allow very fast fatigue life predictions. A first alternative is the **forces-based** approach of Rupp et al. [20][22] using the forces and moments acting in the mid-plane of each connected sheet at the center line of the spot weld to calculate the local structural stress of the spot welded sheets. The structural stress is assessed by universal S/N data derived by Rupp from experiments with several multi-spot-weld test specimens [20]. Figure 9 shows an experimental study on crack initiation in a shock tower, and the Rupp model results for the FE model of the structure. Based on experience from several other BIW fatigue life analyses, the lifetime results of the original Rupp approach are considered conservative. However, the approach is considered to be capable to identify the most critical spot welds with the lowest fatigue life.

A second alternative is to use a **stress-based** approach with a more detailed spot weld representation, namely a rigid core (with the diameter of the spot weld) and a fine shell mesh close to this rigid core, see Figures 10 and 11. The first element row of this fine model consists of second order shell elements (CQUAD8) to improve the stress resolution at the notch. The radial stress detected in the first element row connected to the rigid RBE2 core is assessed with an S/N-curve calibrated to several test specimens.

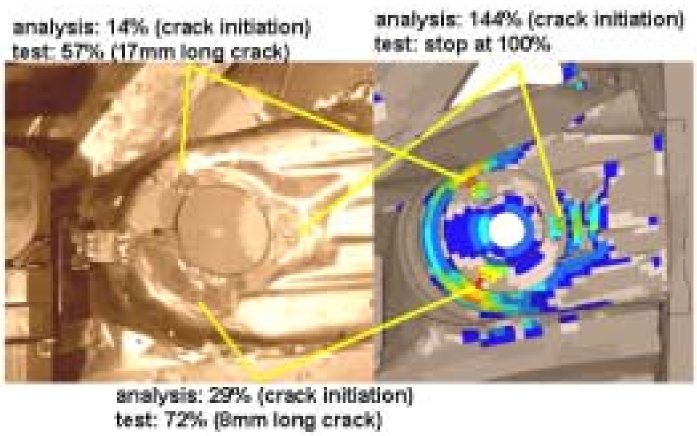


Figure 9 – Experimental crack initiation in shock tower, with the Rupp model predictions.

A **new two-step approach** has been developed at LMS International to allow fast and accurate fatigue life predictions. The procedure is commercially available in LMS Virtual.Lab [16]. A rather coarse spot weld modeling technique is used in the **first step, to predict the most critical spot welds** with respect to the vehicle durability. Three MSC.NASTRAN [14] models are supported for the first step: the HEXA model, the CWELD and the elastic bar (CBAR) model. Note here that the term “coarse model” is attribute specific: the supported models for the first step are accurate models for NVH predictions, but considered “coarse” in a durability context (as accurate local stresses must be predicted). Once a vehicle model has been assembled with such coarse spot welds, an initial durability computation is performed, to identify the most critical spot welds. **The second step then consists of replacing the critical joints with much finer meshes** to improve the fatigue life predictions. For this purpose, LMS Virtual.Lab [16] contains an automated procedure to replace a given number of critical coarse spot welds by the detailed spot weld representation. It is vital that only the most critical spot welds are replaced, since a typical vehicle body has thousands of spot welds and each implemented local fine model of a spot weld increases the number of elements. The latest version of the detailed spot weld model consists of 64 elements for each sheet, so that for a two-sheet connection the total number of elements is increased by about 128 shell elements. Within LMS Virtual.Lab [16], the LMS FALANCS [20] fatigue solver has been used for fatigue life predictions in both analysis steps. Figure 10 outlines the procedure. A graphical representation in LMS Virtual.Lab [16] can be seen in Figure 11: on the left, part of a component with coarse spot welds for the first step (namely HEXA spot welds); on the right, the same component part with refined spot welds for the second step.

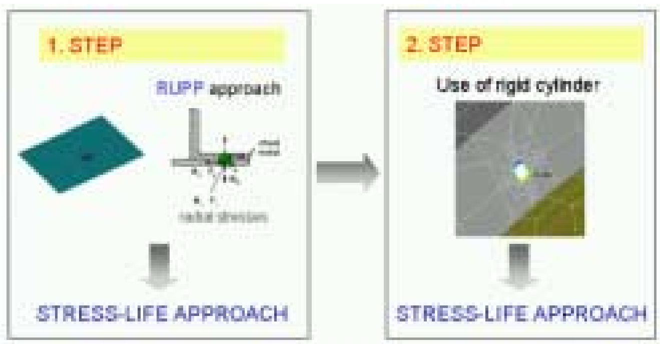


Figure 10 – Two-step approach for spot weld fatigue life prediction.

In a recent industrial study [23], the spot weld connections in 16 identically manufactured vehicle bodies have been tested for fatigue failure. The test results have been compared with FE predictions in LMS FALANCS [20] using the first analysis step, namely the approach of Rupp [20][22]. A factor of 5 between tested and predicted results has been obtained; this factor is considerably higher than the scatter found in test results. These results confirm the capability of the first analysis step to identify the most critical spot welds (a factor of 5 is sufficiently accurate for this purpose). Also, it underlines the need for the second analysis step that can now be set in LMS Virtual.Lab [16]: a more accurate estimate at the critical hot spots is desirable to improve design decisions in the vehicle development process.

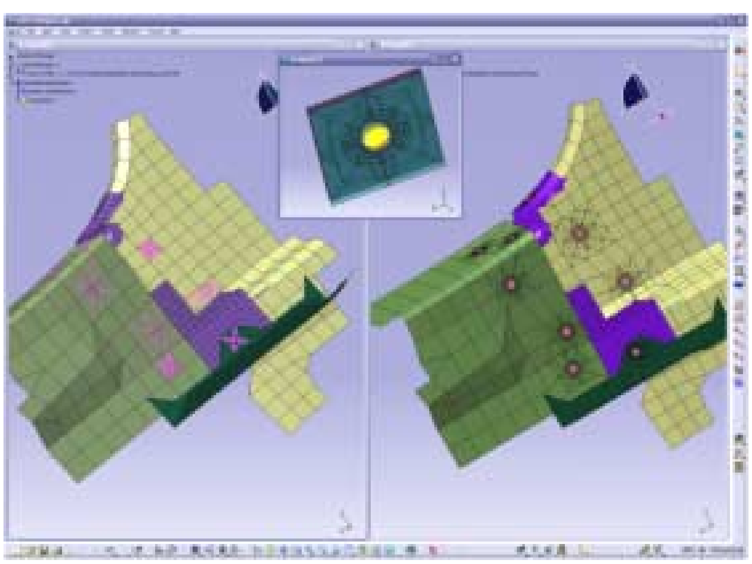


Figure 11 – LMS Virtual.Lab Spot Weld Modeling for Durability. For the first analysis step, a coarse spot weld connection model is used (the HEXA model, left) to identify the most critical spot welds. For the second analysis step, the critical spot welds are replaced by a more detailed spot weld model (right) to allow more accurate local stress predictions.

3.5 Numerical Optimization Strategies

In Section 2.4 it has been explained that it is not feasible to optimize a vehicle's spot weld layout on the basis of **experimental** studies only. In an industrial vehicle model, the huge number of design variables and the uncertainties on a level of spot weld position an quality make it infeasible to verify all design options on expensive physical prototypes that take substantial time to manufacture. Optimizations in a numerical **modeling** phase are much faster and cheaper, and also decrease the level of complexity of the problem. The linear Finite Element models are a simplified representation of the actual product, resulting from a trade-off between accurate representation and fast computation. Many local effects such as geometrical irregularities, residual stresses, material inhomogeneities and defects due to the welding process, are not taken into account by Finite Element modelling. Furthermore, whereas spot welds in a physical structure can be deteriorated or partly-broken, the spot welds in the modeling phase are discrete: a spot weld is placed at a location or not. Given the complexity of the deterministic problem itself, the parameter uncertainty and variability typically needs to be omitted from the models as well. Probabilistic fatigue life assessment studies have been performed [9], but integration of parameter variability in an optimization framework often exceeds computational time limits.

Even with these simplifications, the number of design parameters remains huge: the number of spot welds, their topology, locations, dimensions, material properties, ... remain as design parameters. Even in the numerical modeling phase, any type of industrial spot weld optimization problem therefore results in huge computation times, even when one assumes that an accurate vehicle model is available a with well-chosen global mesh density and suitable local refinements, that a representative spot weld model has been selected for each of the spot welds, and that all of the spot welds can meaningfully be coupled in the structure at all possible locations (or that sharply defined boundaries on the spatial distribution are available). The number of parameters can be highly reduced when one first analyzes the nominal model to find the critical connections, and then only considers the parameters of these spot weld connections in the optimization. A limitation here is that new critical connections may arise when one optimizes the locations and dimensions of the critical spot welds found in the nominal model, and that converge to optimal distribution is not guaranteed.

Despite the difficulties addressed above, a number of spot weld optimization applications have been reported in literature. A simple clamped plate model with two spot welds is investigated in [8], with the aim of finding the optimal fatigue life and stiffness characteristics. Viable results are obtained, but it is also shown that the problem is highly complex, despite the rather simple specifications.

The package N-hance.DOC [24] allows optimizing a vehicle's spot weld design. The vehicle FE model [14] must be available, along with selected forced response and sensitivity analysis results. One can enhance components (for better load path and energy distribution) and fastener layout (for better component connections and higher fastener fatigue-life), before finally the mass is optimized. An industrial application is presented [24]. Nominal NVH and fatigue analyses are performed; using both the frequency and time domain results, 30 critical spot welds are identified and clear design improvements are made.

An alternative method could be based on the package OPTIMUS [27] to create an integrated optimization environment to generate spot weld layouts. Critical inputs can be parameterized in the input file, and output variables of interest can be read from the output file. Distributions and constraints can be added to inputs and outputs, which are then incorporated in the optimization routines to find the most suitable design among all admissible designs. Also, one could use the integrated CAD-CAE environment of LMS Virtual.Lab [16] to select the spot weld coordinates and properties as input parameters, and directly perform the optimization.

4. ROBUSTNESS

An industrial vehicle FE model is used for a robustness study. The spot weld connections are modeled as HEXA solid elements. File management routines have been developed [26] to create the damaged models directly from the nominal MSC.NASTRAN [14] bulk model. A subset of spot welds can be broken with a uniform or weighted-uniform selection procedure. Monte Carlo simulations can then be performed to simulate the scatter of dynamic vehicle characteristics.

4.1 Preparation of Damaged Vehicle Models

Damaged vehicle FE models (with a number of broken spot welds) are generated from the nominal vehicle FE model (with all spot welds intact), following the procedure in Figure 12. The subsequent steps (corresponding to the numbers in Figure 12) are:

1) Separate the spot welds (HEXA cards) from the invariant part of the bulk data;
2) Split the spot welds into a subset to be broken and a subset to remain intact;
3) Create a damaged vehicle model by including only the intact spot welds in the bulk data file.

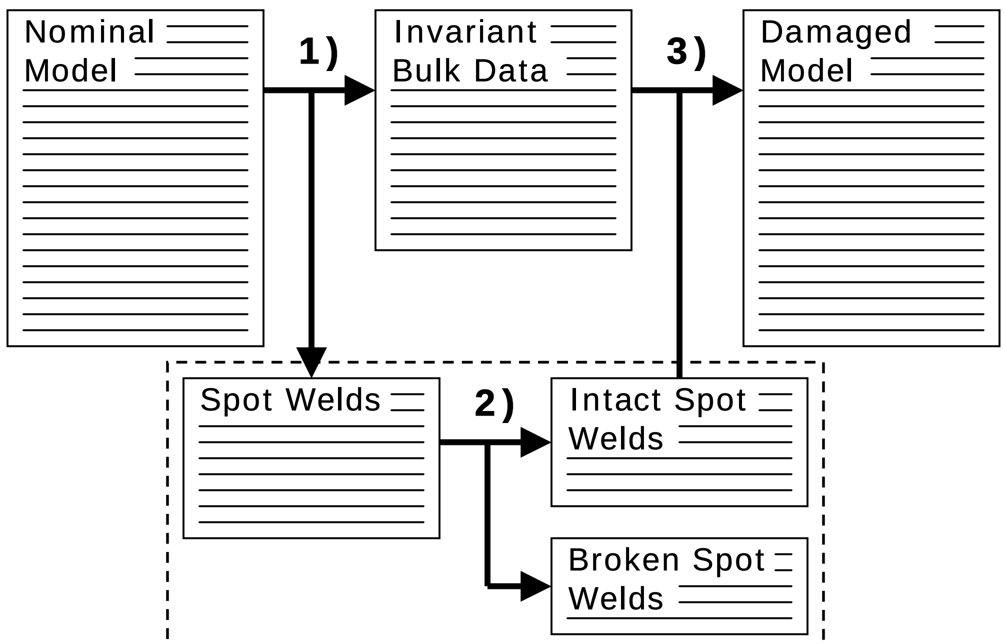


Figure 12 – File management procedures: from the nominal vehicle model, a damaged model is generated, with a number of broken spot weld connections.

Two **random selection schemes** have been used to select the subset of spot welds to be broken. Figure 13 (left) shows the first scheme: a number of spot welds are selected with **uniform probability** throughout the structure. Figure 13 (right) shows the second scheme: a number of spot welds are selected with a weighted-uniform probability; an integer weighting factor W has been assigned to all spot welds. The weighting factor could be based on all kinds of functional performances of the nominal model, such as fatigue life, stress or strain levels, ...

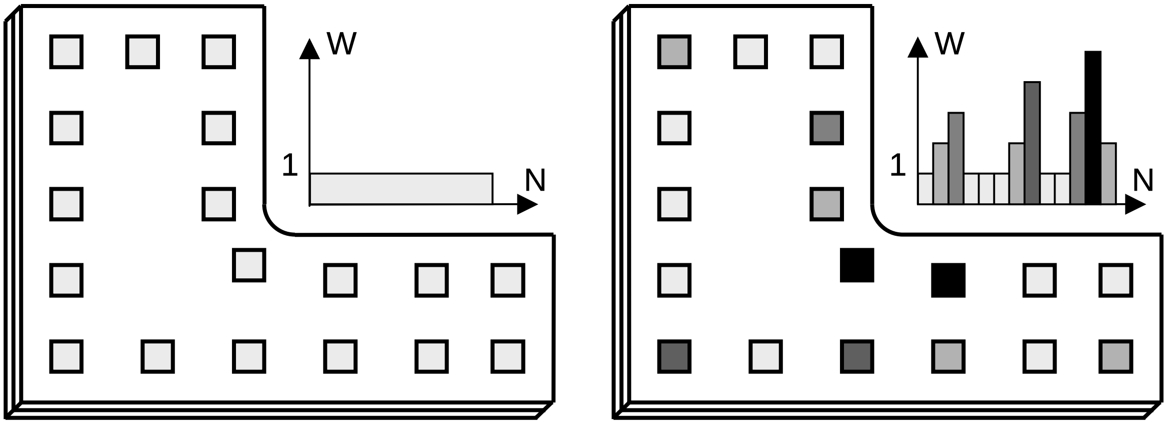


Figure 13 – Random selection schemes: uniform selection probability for all N spot welds (left), and a weighted-uniform selection probability, using an integer weighting factor W (right).

4.2 Monte Carlo Simulations

First a **modal analysis of the nominal model** must be performed, yielding the nominal eigenfrequency values $f_{n,i}$ and the nominal mode shapes; see Figure 14. The Monte Carlo robustness study then consists of two steps:

1) **Generate a number of** N_{mc} **damaged vehicle models** with the procedure in Figure 12, using one of the random selection procedures in Figure 13 to randomly break a subset N_b of the total of N spot welds.
2) **Perform a modal analysis for each of the** N_{mc} **damaged vehicle models**. Use the Modal Assurance Criterion (MAC) [25] to compare the mode shapes of the damaged model with those of the nominal model, so that the eigenfrequency values $f_{d,i}$ of the damaged vehicle model are obtained; see Figure 14.

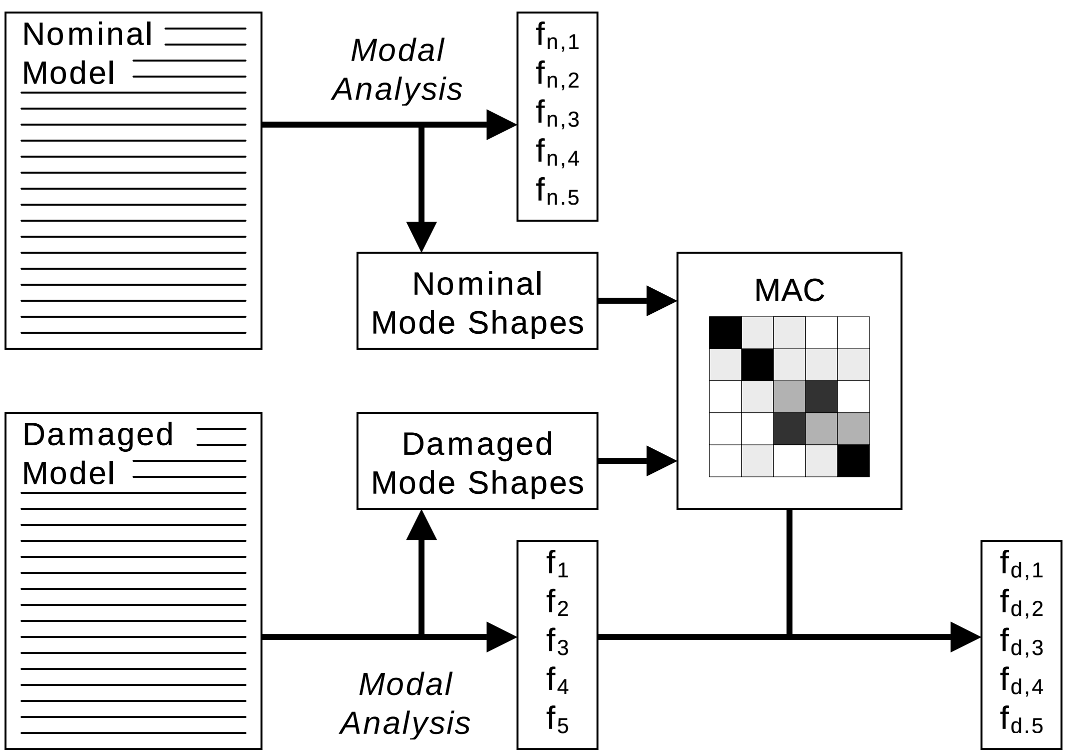


Figure 14 – Modal Analysis and MAC comparison procedure to obtain the eigenfrequency values $f_{d,i}$ of the damaged vehicle model.

5. RESULTS OF THE ROBUSTNESS ANALYSIS

5.1 Analysis of the nominal model

As described in Section 4, the robustness study is performed on an industrial vehicle model [14] with spot welds modeled as solid elements. The FE model consists of 246.336 nodes and 244.256 elements, including 5.992 spot weld CHEXA elements. Note that some “physical spot welds” connect three plates at a time, which is modeled with two CHEXA elements. In this paper, each plate-to-plate connection in a three-plate joint may fail separately.

First, the nominal (undamaged) model has been analyzed. A modal analysis has been performed to predict the nominal values of the fundamental torsion (35.26 Hz) and bending (40.89 Hz) frequency, with the nominal mode shapes for a representative subset of grid points. Also, the element strain energy (ESE) attained in the spot welds at the fundamental resonance frequencies is computed. MSC.NASTRAN outputs the ESE values for all spot welds with an ESE equal to or higher than the threshold level (the default value has been used, i.e. Y 0.001% of the total strain energy of all elements). At the fundamental torsion frequency, the threshold ESE level is exceeded in 586 elements, and for the fundamental bending frequency, the threshold level is exceeded in 693 elements. For the bulk part, these are different elements: only 32 elements have a nonzero ESE output for both fundamental resonance frequencies. In total, 1247 elements thus appear in the ESE output (that is, 586+693–32).

5.2 Breaking a subset of spot welds with highest ESE

A total ESE term is obtained by adding the ESE values attained at the fundamental torsion and bending resonance. As a first test case, a set of damaged models is generated where the N spot welds with highest total ESE are broken. Figure 15 shows the fundamental resonance frequencies for different values of N.

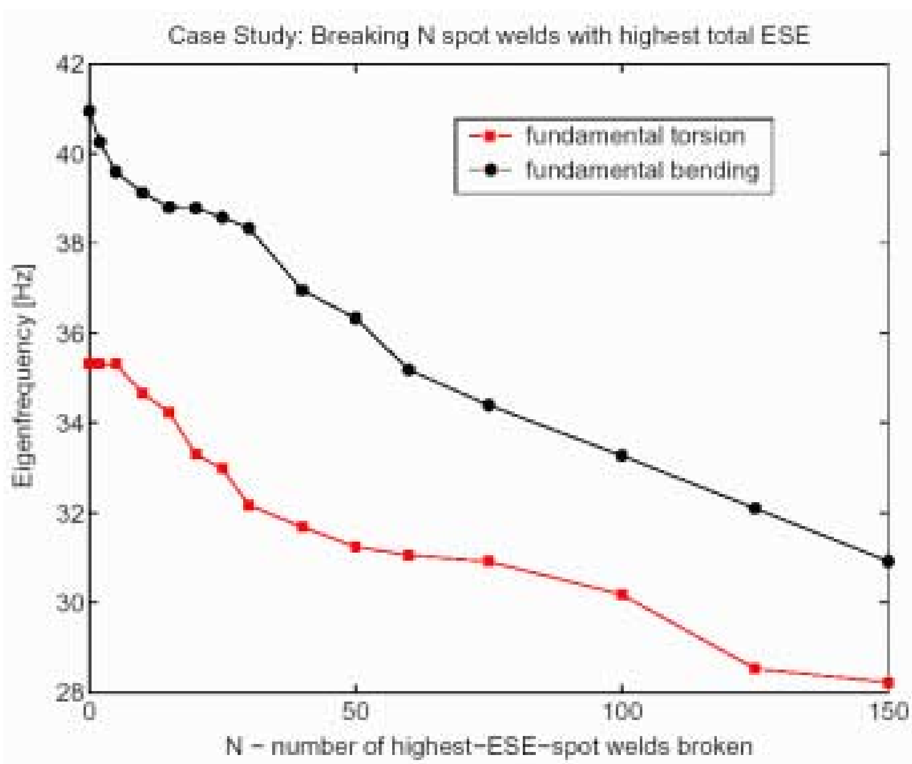


Figure 15 – The effect of breaking a subset of spot welds with highest total ESE at the fundamental torsion and bending eigenfrequency.

Clearly, breaking the spot welds affects the fundamental eigenfrequencies. Removing spot welds reduces the stiffness with respect to torsion and bending, so that the fundamental eigenfrequencies decrease when N increases. One can also recognize that the 5 spot welds with highest ESE only play a role for the fundamental bending frequency (as the fundamental torsion frequency remains constant up till N=5).

5.3 First Monte Carlo robustness study: uniform selection

For the first robustness study, a set of damaged vehicle models is created. The procedure in Section 4.1 is followed with a uniform selection (see Figure 13, left) to randomly break a subset of the spot welds in each damaged vehicle model. A Monte Carlo simulation is then performed: for each damaged vehicle model the dynamic characteristics are computed. The specification of the robustness study is as follows:

- N_{mc} = 100 Monte Carlo simulations (i.e. N_{mc} = 100 damaged vehicle models)
- N_b = 350 spot weld elements broken in each damaged vehicle (i.e. about 6% of the total of N = 5.992)

The scatter of the Monte Carlo simulation results is shown in Figure 16, together with the nominal solution. In analogy with Figure 15, the eigenfrequency values are reduced, as the stiffness characteristics with respect to torsion and bending are reduced. Some Gaussian properties have been estimated for the scatter of the Monte Carlo simulations results; see Table 1.

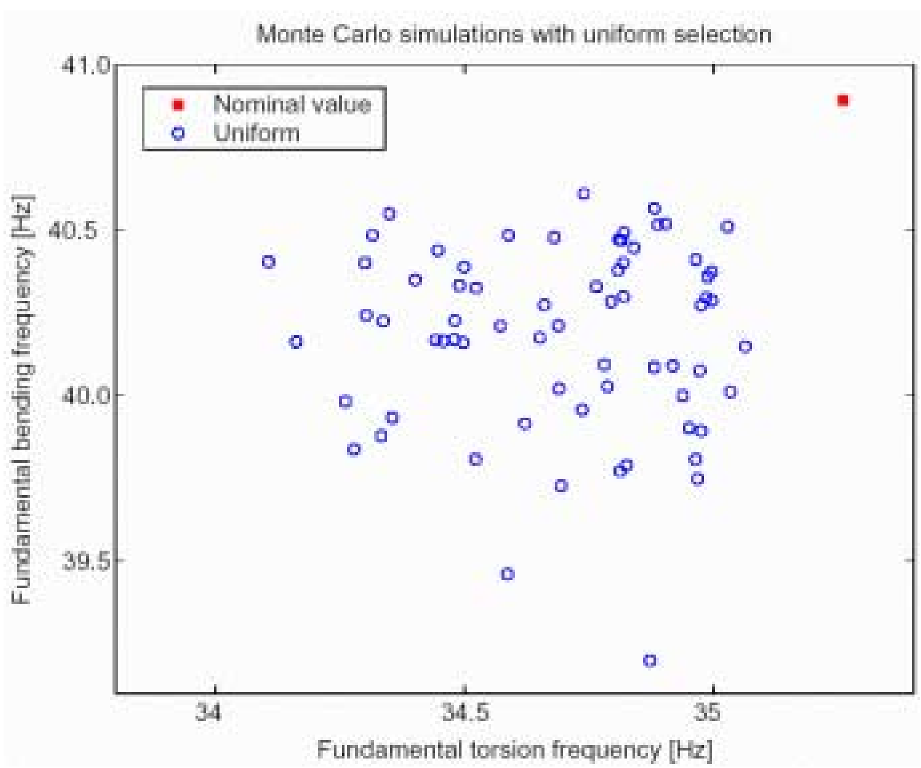


Figure 16 – First robustness study: scatter of fundamental eigenfrequency values as a result of breaking a subset of spot welds with a uniform selection probability.

5.4 Second Monte Carlo robustness study: weighted-uniform selection

For the second robustness study, again a set of N_{mc} = 100 damaged vehicle models is created, in which N_b = 350 spot weld elements are broken. The only difference with the procedure in Section 5.3 is that a weighted-uniform selection procedure is used.

The weighting factor is based on the nominal ESE levels attained in the spot weld elements at the fundamental resonance frequencies (see Section 5.1). For this second experiment, the aim has been to find a weighting factor that moderately increases the probability that spot welds are broken for which nonzero ESE levels are attained in the nominal analysis. After a few experiments for tuning purposes, the weighting factor W has been set to

$$W = \left\lceil \sqrt{\frac{E_t}{\varepsilon_t} + \frac{E_b}{\varepsilon_b}} \right\rceil \qquad \text{(eq. 1)}$$

Here, the subscripts *t* and *b* refer to torsion and bending frequency, respectively. The nominal ESE values are denoted as *E* and the ESE threshold levels as ε. The ceil operator $\lceil \bullet \rceil$ rounds a real value $\bullet$ upward to the nearest integer value. In practice, assuming that a finite ESE value exists for all elements, this implies that

- W = 1 when $E_t < \varepsilon_t$ and $E_b < \varepsilon_b$ (i.e. ESE values below threshold level)
- W follows (eq. 1) when $E_t > \varepsilon_t$ and/or $E_b > \varepsilon_b$ (i.e. MSC.NASTRAN outputs a nonzero ESE level)

Let N_{nz} =1247 denote the number of elements with nominal ESE values above threshold; see Section 5.1. Using (eq. 1), a maximum weighting factor of 13 is obtained. The average weighting factor $\hat{W}$ for the elements with ESE above threshold is equal to $\hat{W} \approx 2.74$. The following can now be derived:

- With the uniform selection procedure in section 5.3, the N_{nz} elements with nonzero ESE values constitute a fraction of 21% in the selection population that consists of *N* = 5992 elements:

$$P_u = \frac{N_{nz}}{N} = \frac{1247}{5992} \approx 0.208$$

- With weighted-uniform selection procedure, using the weighting factor W in (eq. 1), the N_{nz} elements with nonzero ESE values constitute a fraction of 42% in the selection population:

$$P_{wu} = \frac{\hat{W} \cdot N_{nz}}{\hat{W} \cdot N_{nz} + 1 \cdot (N - N_{nz})} = \frac{3417}{8162} \approx 0.419$$

The fraction of the N_{nz} elements with nonzero ESE levels is thus increased with a **factor two**. In the uniform selection procedure, one can expect that 350 · 0.21 = 74 spot welds among the N_{nz} elements with nonzero ESE levels are broken; for the weighted-uniform selection procedure, this number increases to 350 · 0.42 = 147. One obtains a moderate but substantial increase of the probability that these N_{nz} spot weld elements are broken.

For the weighted-uniform procedure, the scatter of the 100 Monte Carlo simulation results is shown in Figure 17. The nominal result and the scatter obtained in the uniform procedure (see Figure 16) are also shown for comparison purposes. The scatter of the weighted-uniform Monte Carlo simulations is much higher. Also for the weighted-uniform procedure, the estimated Gaussian properties are shown in Table 1. The mean values show that the fundamental eigenfrequency values are further reduced by the introduction of the weighting factor, thus proving the (perhaps rather trivial) relation between the ESE value attained in the spot weld and the importance of the spot weld for the considered eigenfrequency. The scatter is also increased, as can be seen by comparing the standard deviations. For the fundamental torsion frequency, the standard deviation increases with a factor two. The fundamental bending frequency increases even more, almost with a factor 3.

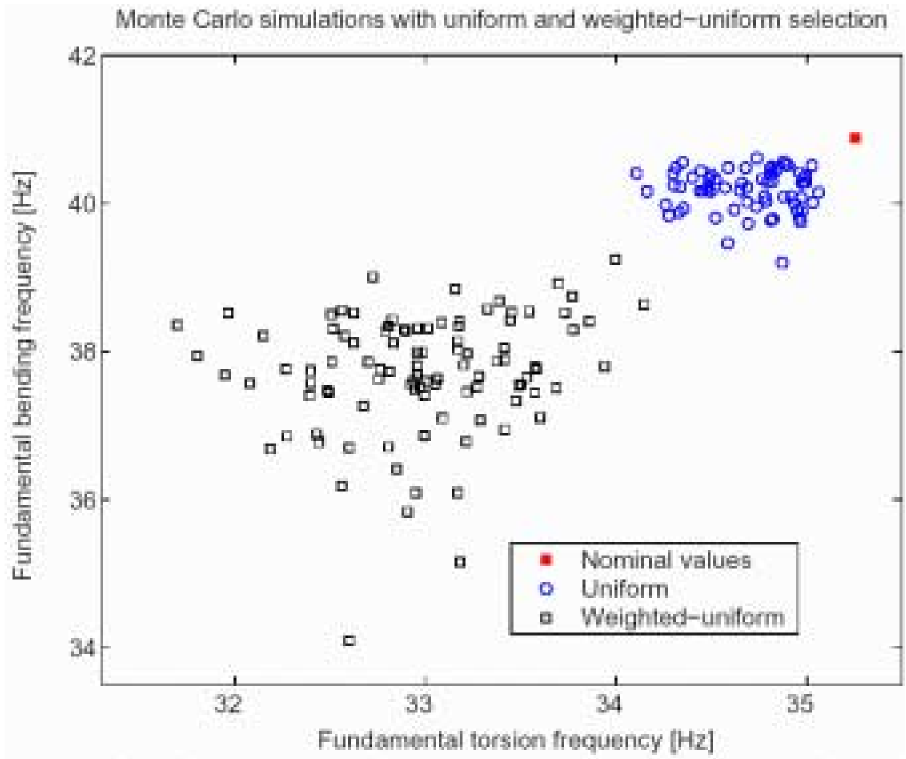


Figure 17 – Second robustness study: scatter of fundamental eigenfrequency values as a result of breaking a subset of spot welds with a weighted-uniform selection probability. For comparison, the results of the first robustness study in Figure 16 are also shown.

Table 1 – Statistical properties (mean and standard deviation of the fundamental torsion and bending eigenfrequencies, and the correlation coefficient between them) have been derived for the uniform and weighted-uniform scatter clouds in Figure 17.

	Uniform selection		**Weighted-uniform selection**	
	torsion freq	bending freq	torsion freq	bending freq
mean value (μ)	34.68 Hz	40.17 Hz	32.99 Hz	37.71 Hz
standard deviation (σ)	0.25 Hz	0.30 Hz	0.50 Hz	0.81 Hz
correlation coefficient	-0.091		0.18	

CONCLUSIONS

Several joining techniques are used in the vehicle assembly process. The dominating technique is resistance spot welding: two metal panels are pressed together, and an electric current is applied. This results in local metal fusion, so that the two plates are welded together with a spot weld. A typical vehicle body contains thousands of spot welds. The design of a spot weld layout has a high number of variables and unknowns. The complexity of the problem is further increased by the tuning parameters and inaccuracies in the welding process, and by the load history in a vehicle's lifetime. This makes it impossible to base the spot weld design only on experimental data; the spot weld layout should be designed and assessed in a numerical modeling phase.

In this paper, an overview has been given of the use of spot welds in the automotive industry. Spot weld testing and experiments, aimed at assessing the strength and stiffness, NVH and fatigue characteristics, have been reviewed. Fatigue failures occur at much lower loads than those attained in strength tests. Although fatigue failures are local phenomena, the failure of spot welds may also affect the vehicle's stiffness and NVH performance on a global level.

Numerical Finite Element models of spot welds have also been discussed. Selection of model size and complexity is a trade-off between computation time and prediction accuracy, as is the case in any Finite Element model. It is agreed that the same spot weld model should be used everywhere in the structure, as a separate design for each spot weld model takes far too much time. The desired spot weld accuracy depends on the functional performance addressed. In an optimization, the balance is more shifted to coarse models, as the computation time per iteration should be reduced. Stiffness and NVH, being global phenomena, can also be predicted with relatively course models, while spot weld models for fatigue must have a very fine mesh to accurately predict local stresses and crack initiations.

It has been argued that a substantial number of spot welds may be absent in an operational vehicle, because of manufacturing inaccuracies, fatigue failures and perhaps minor accidents. Moreover, some spot welds may be forgotten in the CAD models, so that spot welds may be absent in derived CAE models as well. To take these effects into account, file management tools have been presented that allow breaking a subset of spot welds in the nominal (undamaged) vehicle model. Selection of broken welds can be done with uniform selection probability, or with an integer-weighted uniform selection probability.

These tools have been used in two Monte Carlo robustness studies, to assess the effect of spot weld failure on a vehicle's fundamental torsion and bending frequency. Six percent of all spot weld connections have been broken, first with uniform selection, and then with a weighted-uniform selection procedure. The weighting factor has been based on the element strain energies attained in the spot welds at the fundamental resonance frequencies. Scatter plots have been shown, and statistical properties of the scatter clouds have been compared. It has thus been demonstrated that the dynamic vehicle characteristics are clearly affected by the spot weld failure.

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