High Performance, Process Oriented, Weld Spot Approach

Dipl.-Phys. **Daniel Heiserer**[1], **Mladen Chargin**[2], Dipl.-Ing. **Juergen Sielaff**[3]

1 Abstract

Simulation of automotive vehicles has become a more important step in car development in the last few years. Rapid prototyping, which offers a close-to-market product, with less hardware, requires a fast and accurate computer simulation to guarantee a quality product.

Generation of body-in-white finite element models for full vehicle simulation of *Linear Statics*, *Normal Modes*, *NVH*, and *Crash* is one of the most critical items. This is due to the need to connect various body-in-white components which requires:

- a significant amount of manpower because of difficult automation possibilities
- a "physically correct" representation of part-connectors such as weld spots, screws, etc., which cannot be easily automated and requires more manpower resulting in delays and a potentially large source of error

In the past, a number of methods were developed for joining these parts together, such as *homogenous models* and various other *weld spot approaches*. In the generation of FE-models, each part depended on its connected parts, which made it difficult to shorten the modeling process. Benefits of faster hardware, and some improvements in preprocessors, were diminished by the need for bigger models.

Despite the above improvements, a significant amount of time was still required to generate FE models. **CDH** and **BMW** developed a spot welding approach, **linchweld**, which simultaneously reduced the modeling time, invested manpower, and increased the quality of simulation (compared to test).

The program containing the linchweld approach is called *CDH/SPOT*.

2 Requirements for Spot Weld Modeling

2.1 Quality requirements

One of the most important reasons for modeling spot welds is that without a physically meaningful representation of such connectors, it is probably not possible to generate finite element

[1]BMW AG Muenchen
[2]CDH GmbH
[3]BMW AG Muenchen

models that have sufficient correlation to test in order to build virtual vehicles. All experiences in this field showed us that in models which do not consider spot weld information, it is possible only to propose relative changes, not to propose absolute values. This is important if the design process is to be shortened dramatically and design decisions made based on virtual studies only.

Based on comparison of Modal Analysis up to 50 hertz in experimental and numerical testing of at least five body-in-white models, it is probably not possible to build homogenous finite element models which have a mean relative error in eigenfrequencies less than 4-8% and a mean MAC (Modal Assurance Criterion) greater then 0.85.

The targets of a spot weld approach have to reach mean values of eigenfrequency errors less than 2% and mean MAC values of 0.95.

2.2 Modeling time requirements

Looking at well known modeling techniques for spot welds [2] using 1-dimensional elements like bars, beams, springs, rigid bars or even coincident nodes, it is obvious that modeling of body-in-white structures does not become easier. In modeling one part, all the connected parts have to be considered to get matching flanges and also to get matching nodes with spot weld data coming from CAD databases. The modeling of body-in-white structures, which is still a very time consuming and therefore expensive procedure, becomes tremendously more time consuming when spot welds are considered. Evaluations of different spot weld locations are nearly impossible, because the whole mesh of all attached parts has to be changed.

A spot weld approach has to shorten the modeling time to make the whole analysis procedure, ranging from deriving CAD data up to postprocessing the analysis results, much faster. A reduction in modeling time of 50% is desired.

2.3 FEM-data management requirements

Changing an existing finite element model; for example, exchanging an old part, e.g., a sideframe, results in more work if the model is built based on the philosophy of dependent meshes, such as homogenous models, or other spot weld approaches mentioned above. All parts attached to the new part either have to be changed or the new part has to be modeled with boundary conditions given by the attached parts, making life very difficult for mesh generators.

Changing meshes only to make them congruent to new parts also raises questions regarding the influence of the mesh change compared to that of a new part.

The third requirement for a spot weld approach was that parts, represented by meshes which do not change, should not change their mesh if the connected parts change.

3 Physical Representation of the Spot Weld

A new spot weld approach for *linear* models was developed and tested by CDH and BMW.

3.1 Problems and mechanical considerations of 1D representations

Experiences with different spot weld representations using 1 dimensional elements demonstrated the following problems:

- Shell element rotational stiffness degrees of freedom are not *strong* enough to carry the rotations introduced by elements such as beams or springs. A singularity is introduced in the shell area. The model does not run without *PARAM, K6ROT* and *PARAM, SNORM.*

- Beam, or even worse, bar elements are used whose diameter is approximately 5 times bigger than their length. The main problem, besides the ill conditioned matrices, is the abuse of finite elements whose formulation is based on the assumption that their axial dimension is much *bigger* than their radial dimension.

Some type of a **surface contact**, representing the area of the real spot weld, had to be developed. We should look at a typical spot weld to find a reasonable solution for this requirement.

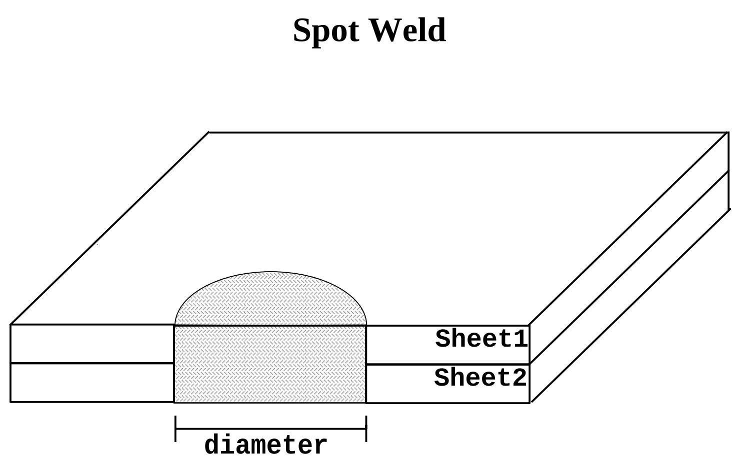

Figure 1: spot weld scheme

Approximately 4000 spot welds are used in a typical body-in-white. The diameters range from 3 to 7 millimeters, while typically connecting sheets with a thickness of around 1 millimeter. Normally only two or three sheet connections are used. Note that the ratio of width to height is about 2. The forces that can possibly be transmitted by the spot weld are:

- pull and push forces (blue)

- shear forces (green)

- shear moment (red)

Assume that 1-dimensional elements are used to directly connect two nodes on each side of the part. Consequently, the shear forces would generate bending and the shear moments would generate rotational forces in the plate which can only be resisted by the artificial stiffness created by *PARAM, K6ROT*, if specified. Whenever 1-dimensional elements are interpolated on the surface, the interpolation suffers from this problem if the interpolating grid gets close to a grid of the shell.

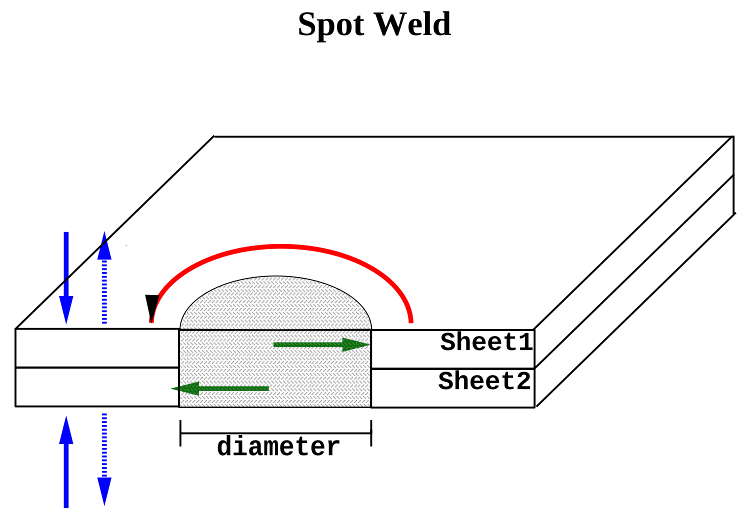

Figure 2: relevant forces for the spot weld

To support relevant weld spot forces, not only a single point of $sheet_1$ has to be connected with $sheet_2$, but the full area shown in Fig. 3.

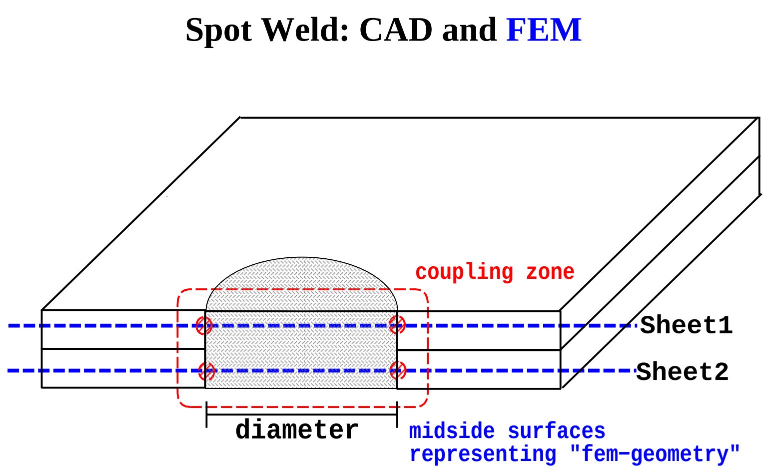

Figure 3: finite element sheet location and relevant coupling zone for the spot weld

3.2 Interpolation scheme for independent meshes

An interpolation scheme was developed to support

- the request for absolutely independent meshes of $sheet_1$ and $sheet_2$ and
- the exact location of the spot weld.

The boundary marked in Fig. 3 is the exact location for the connection between the *spot weld* and the sheets. In general, this location is not coincident with a grid location on either one of the sheets. The only solution that provides an exact coupling of the spot weld points with the sheets is to generate and interpolate new grids by using the displacement function for the underlying element. Assuming the underlying element is a linear quadrilateral we get from [3]:

$$\vec{u}_{G_5} = \alpha_1\vec{u}_{G_1} + \alpha_2\vec{u}_{G_2} + \alpha_3\vec{u}_{G_3} + \alpha_4\vec{u}_{G_4} \qquad (1)$$

$$\sum_{i=1}^{4}\alpha_i = 1 \qquad (2)$$

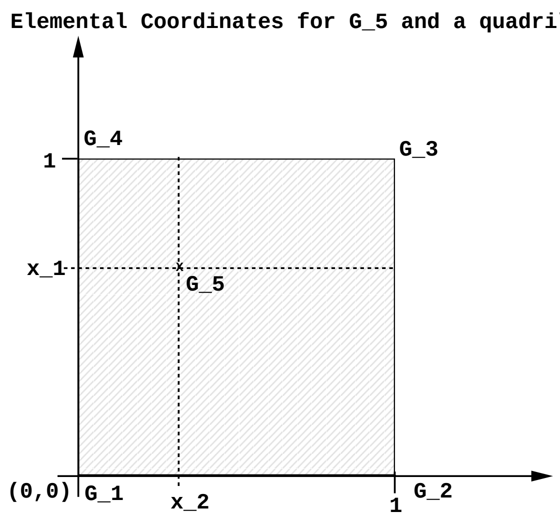

Figure 4: elemental coordinates for a quadrilateral element

In Fig. 4 the coordinates of the new grid G_5 are shown in the elemental coordinate system of a quadrilateral element.

The coefficients for equation 1 are

$$\begin{aligned} \alpha_1 &= (1-\xi_1)(1-\xi_2) \\ \alpha_2 &= (1-\xi_1)\xi_2 \\ \alpha_3 &= \xi_1\xi_2 \\ \alpha_4 &= \xi_1(1-\xi_2) \end{aligned} \qquad (3)$$

In the case G_5 falls into a linear triangular plate element the shape displacement functions (1) and coefficients (3) are similar [3]. The interpolation can easily be done using MSC/NASTRAN RBE3 elements.

3.3 The spot weld element

Figure 3 illustrates that there are not that many obvious possibilities to generate a reasonable element which can provide for:

- transmission of forces considered in 3.1
- a geometric *coupling zone* as shown in Fig. 3

There are two possible approaches to satisfy the above requirements. One approach is to use a rigid (RBE2) element, whose master node is defined by the center of the spot weld. However, the RBE2 element has some disadvantages:

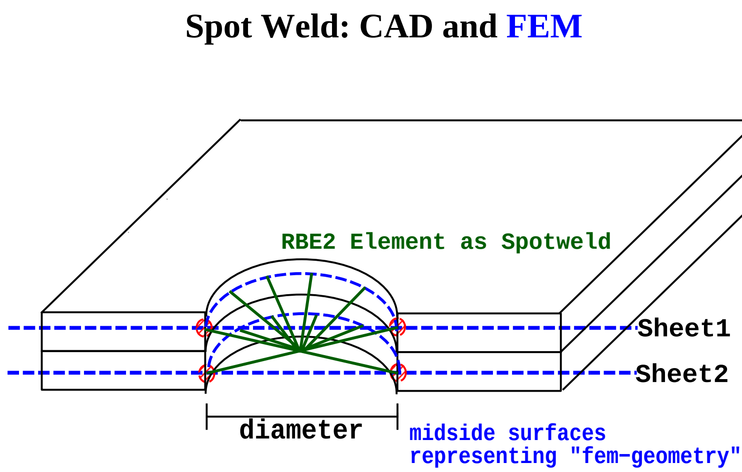

Figure 5: scheme for a spot weld element using a rigid (RBE2) element

- It can be difficult to visualize (looks bad).
- It is hard to use for subsequent analysis, such as optimization, which may require either material sensitivity or internal stress.
- In general, it is not possible for *any* discretization to resolve the dependency conflict introduced by the rigid and constraint elements. Even with RBE3's *UM* specification it cannot be used if more than three nodes fall into one triangle.

The second possibility is to use a HEXA element. Compared to the RBE2, HEXA has the following characteristics.

- It can easily be visualized (looks good).
- It can be used for subsequent analysis, because it has material and stress sensitivity.
- It does not cause conflicts with the interpolation scheme

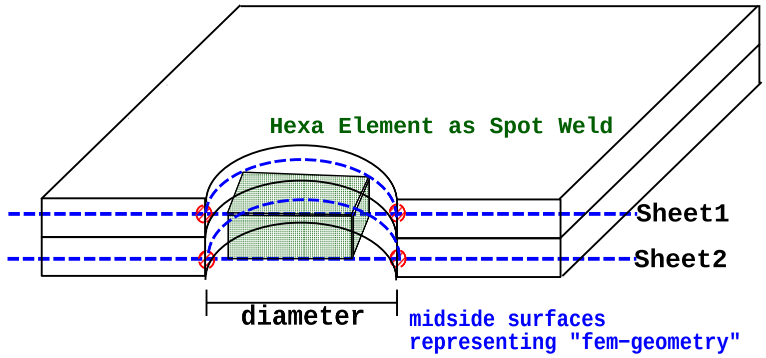

Figure 6: scheme for a HEXA element as a spot weld

- it is able to represent the required surface coupling in the desired way.

The use of HEXA element is the most reasonable for our approach. Its usage results in almost no problems, assuming the underlying mesh can meaningfully be connected, i.e., no bad geometry.

The basic input for *CDH/SPOT* is a simple list of spot welds containing all the necessary information:

- id
- diameter
- point (x,y,z), coordinates as geometrical information
- part-identifier
  - Part-identifier 1 (can be blank)
  - Part-identifier 2 (can be blank)
  - Part-identifier 3 (optional)(can be blank)

The input required by *CDH/SPOT* for the spot welds has the following format:

```
$ The '$' indicates a comment
$...1.-><-..2.-><-..3.-><-..4.-><-..5.-><-..6.-><-..7.-><-..8.-><-..9.-><-....->
$Ident        ID     Dia        x        y        z    pid-1    pid-2    pid-3
GRID      904210           174.11  -789.60   598.33    11011    11061    11281
GRID      904211     8.0  -213.70  -789.60   673.06    11011    11061    11241
GRID      904212     3.6    52.87  -483.83   578.79    11011    11021
GRID      904213     3.6     9.87  -483.69   584.46    11011    11021
GRID      904214           -33.15  -483.63   590.16    11011    11021    11031
```

```
GRID      904215          -76.17 -483.62  595.86   11011   11021
GRID      904216      5.0 -119.05 -483.62  601.57   11011   11021
GRID      904217      5.0  174.11  789.60  598.33   11012   11062   11282
GRID      904218         -213.70  789.60  673.06   11012   11062   11242
GRID      904219           52.87  483.84  578.79   11012   11022   11032
```

A data format similar to NASTRAN's *GRID* format makes it very easy, at least for those able to use an editor, to integrate the procedure into current preprocessors. The feature of blank fields for the part identifiers allows, if spot weld data are not available, the use of simple grid points only.

3.4 Examples of sheet coupling with the *linchweld* approach

Figures 7 to 10 give an impression of how the program works and how the generated elements look.

In Fig. 7 the generated HEXA for a 2 sheet spot weld is given. One can also clearly see the mesh independence of $sheet_1$ and $sheet_2$. Figure (8) shows how one of HEXA's new grids is interpolated on the upper sheet using *RBE3* elements. Also, a three part connection can easily be made using the *linchweld* approach as seen in Fig. 10. A view of an assembly of some parts welded together by the algorithm is given in Fig. 9.

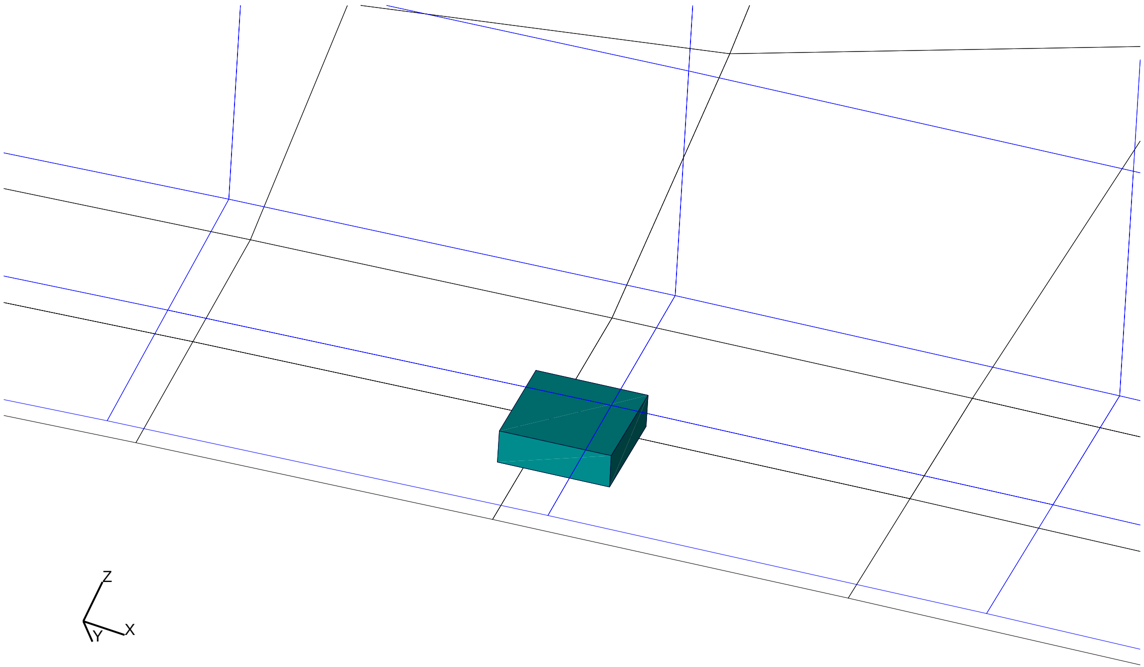

Figure 7: realization of a spot weld using the *linchweld* approach

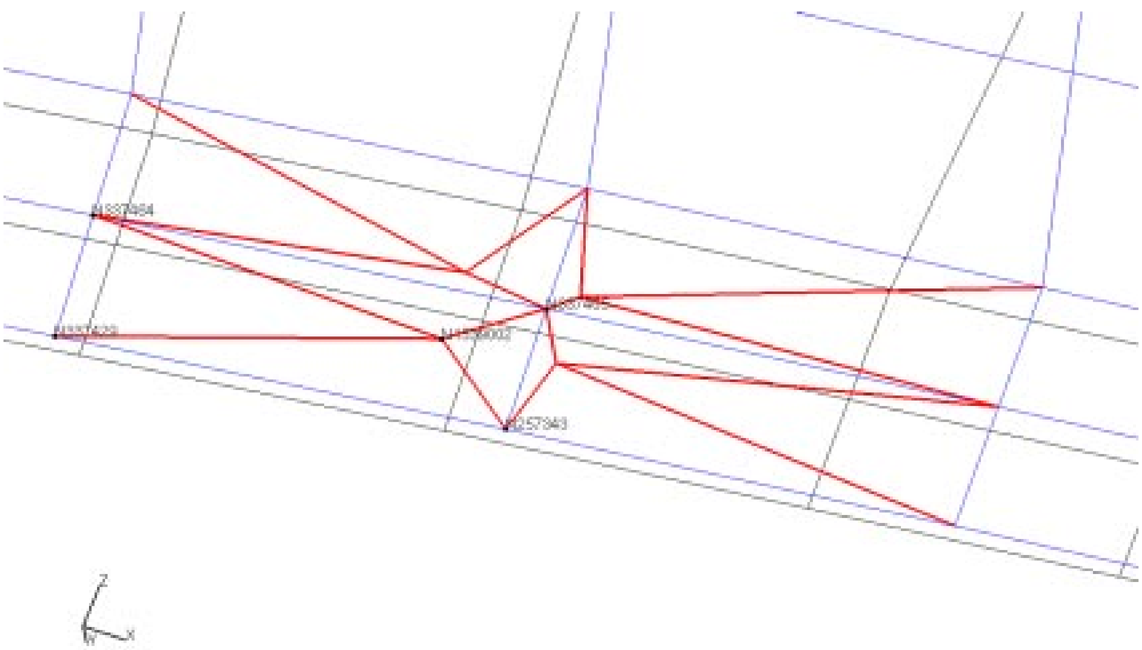

Figure 8: example for the interpolation of the newly introduced grids using RBE3 elements

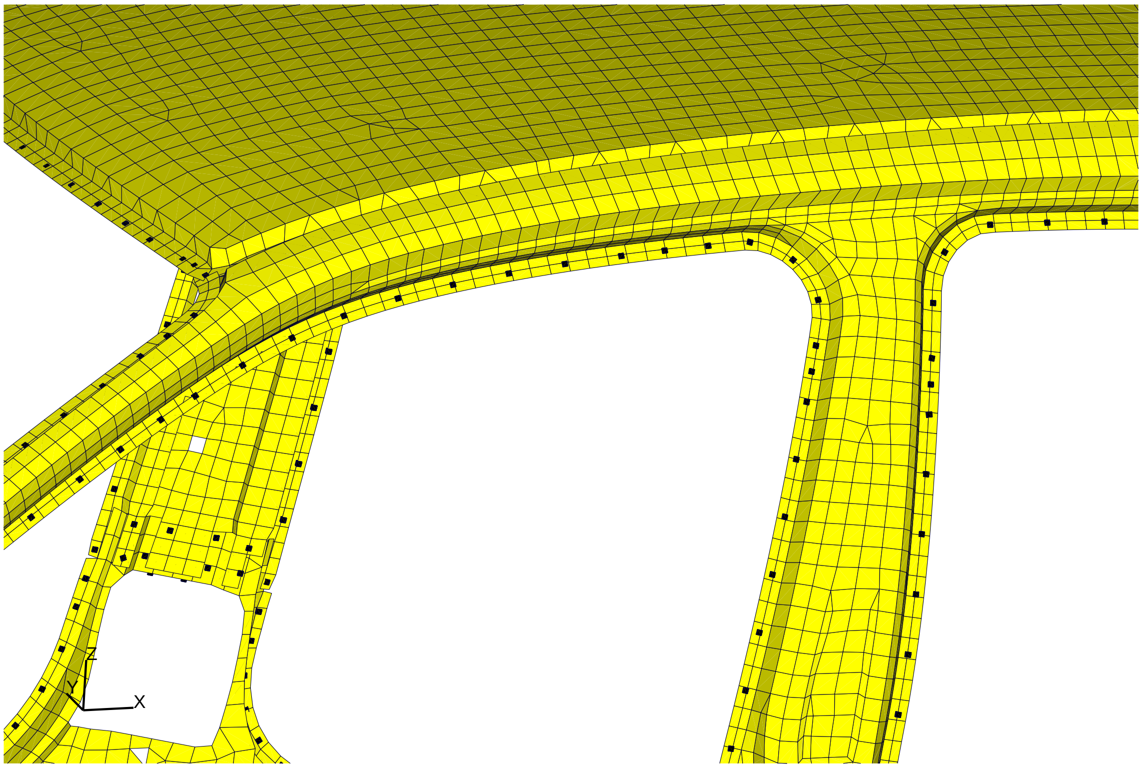

Figure 9: overview of a welded part of a body-in-white

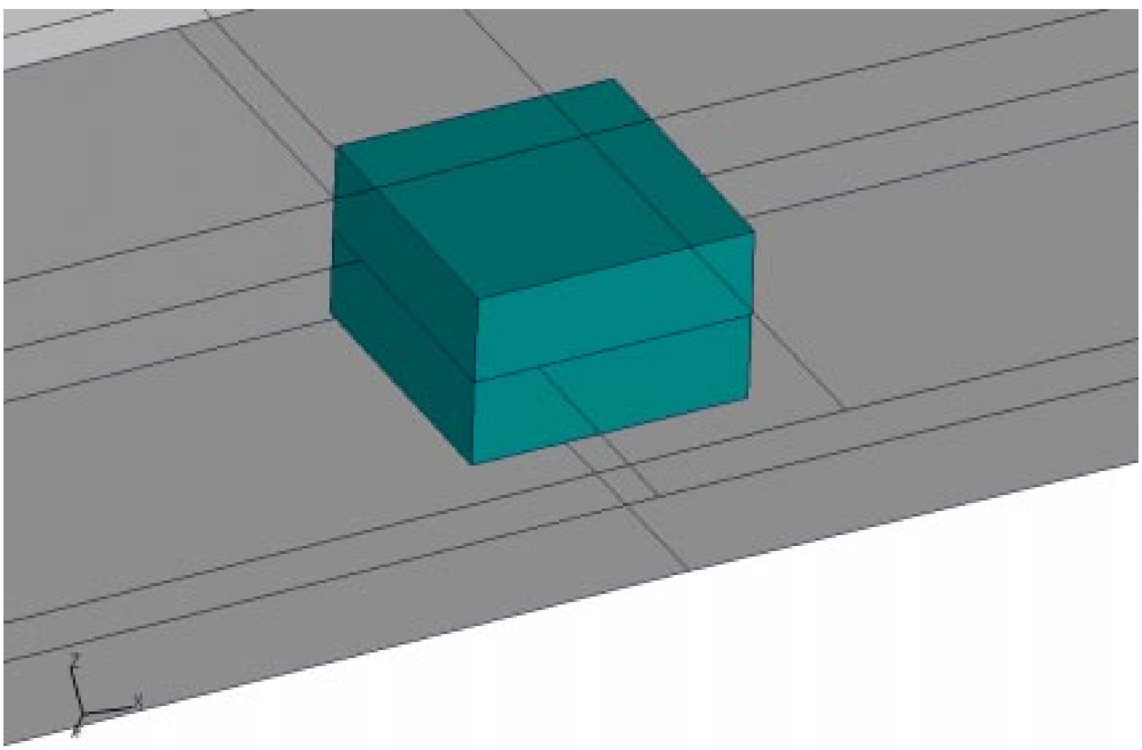

Figure 10: example for welding of 3 sheets

4 Results for the *LINCHWELD* Approach

As mentioned above, a spot weld approach is a multicriteria problem. An improvement in accuracy must not be followed by a tremendous increase in modeling time or total cost. However, the increase in quality of comparison with test was the most significant motivating factor.

4.1 Results compared with test

There are several ways to verify the *linchweld* approach for spot welds. However, it was quickly discovered that the verification was not that simple owing to:

- Static stiffness, of either a simple specimen or a body-in-white, is greatly influenced by the boundary conditions and more questions are raised than answered.
- For dynamic stiffness, the required element size, owing to wavelength considerations, becomes very small for a small specimen and the ratio between the mesh size and the spot welds length is totally different than that for the desired target, a body-in-white.

Because of these reasons, the most important objectives for verification were the normal modes of a body-in-white between 5 and 50 Hz.

For verification, a *BMW X5*, *3 Series*, and *5 Series* were used. Experimental and numerical results for normal modes between 5 and 50 hertz were compared. The modeling techniques used for the spot welds were:

- homogenous
- spot weld models with beams and congruent meshes
- spot weld models with the *linchweld* approach.

The results are shown in Fig. 11. Model updating, as in any multicriteria optimization focused on improving the modeling techniques, has to use a certain *norm* to determine which technique is the best. However, such a norm should not hide the physical content of the problem. The normal modes comparisons were made for all eigenfrequencies in the desired range as well as a comparison of their mode shapes for at least 240 degrees of freedom uniformly distributed over the whole structure.

The x-axis in Fig. 11 shows the absolute value of the relative error between experiment and analysis, while the y-axis shows the Modal Assurance Criterion (MAC) [1] value.

The middle node of the *spiders* represents the mean value of all modes in the range, while each *slave* of the spider represents a mode. The better the model the closer its spider moves to the origin. Comparing Model A, we can see the difference between a homogenous approach and a 1-dimensional spot weld approach using congruent meshes and beam elements. While the 1D spot weld approach improves the MAC value tremendously, it loses in the eigenfrequencies accuracy.

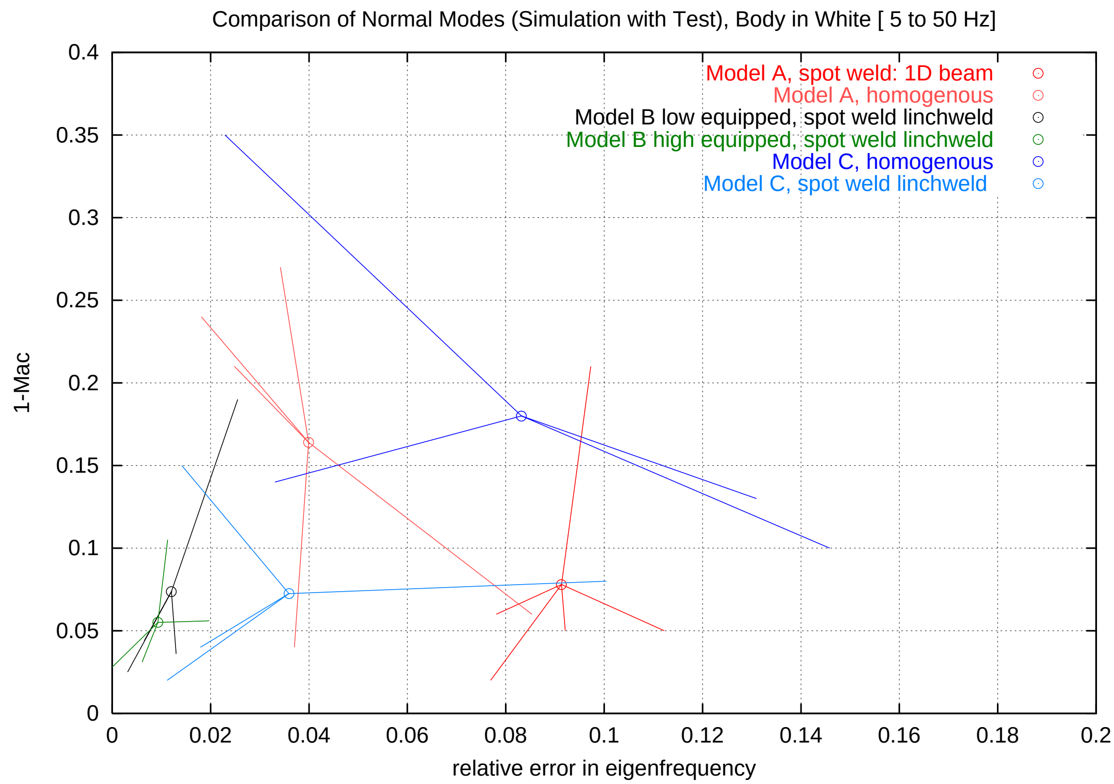

Figure 11: correlation between test and simulation

Model C shows the difference between a homogenous model and a spot weld model using the *linchweld* approach. In the mean sense, there was a dramatic improvement in the MAC value as well as in the eigenfrequencies.

The last example, representing Model B, shows the behavior of the *linchweld* approach for different attachments, such as windshield or other stiffeners. In both cases, the linchweld approach shows good results.

It should be noted that the *linchweld* approach did not improve both the eigenfrequency and MAC value for all modes simultaneously. However, for those few modes, one metric may have been reduced while the other was improved.

4.2 Modeling efficiency considerations

When considering the model generation effort, it is obvious that the invested manpower to build the model containing hundreds of totally independent parts must be much smaller than the case in which all these parts depend on each other.

Our experience has shown that there is a 40% reduction in manpower requirements if there are no mesh dependencies for the body-in-white. It should be noted that this reduction is obtained even though the available software products, processes, and meshing people are "optimized" for the generation of homogeneous, not independent, models. In the future, when the new software is applied and the users properly trained, we should experience further reduction in the modeling effort.

The total calendar time for modeling a full body-in-white is just as important as the invested manpower. A model which contains dependent meshes cannot be meshed totally in parallel because of the dependencies. However, a model which contains no mesh dependencies can be meshed totally in parallel. In other words, instead of one person making a model consisting of 300 parts one *could* use 300 people, each meshing one part. The availability of the final model is now dependent only on the time required to generate the *biggest* part.

Besides the time for meshing, the time for assembling and welding the model also must be considered once a mesh independent approach is chosen. If there is no spot weld data available, or the data needs corrections, the invested manpower for welding is in the order of the original 40% reduction. If there is no efficient preprocessor available and there is no spot weld data, the modeling time for a homogenous model and a inhomogenous model is approximately the same.

An efficient preprocessor for spot welds, such as *MSC/AMS*, helps to dramatically reduce the time spent in the spot preprocessing.

CDH/SPOT program, using the *linchweld* approach, requires about 45 seconds to generate 5000 spot welds for a car body with 200000 elements on a SGI/Octane. An *INTEL* platform with *linux* needs approximately the same amount of time.

CDH/SPOT is also used for preprocessing of spot welds. For spot welds, which cannot be generated, the program issues a problem dependent error code, which can be read into a preprocessor, like *MSC/AMS*, for correction.

4.3 Data management benefits of independent meshes

Another benefit from using the *linchweld* approach is the management of the finite element models. Once a model is generated from scratch it is modified many times in its lifetime. These modifications can be initialized by numerical optimization, variant generation or updates from CAD. Modifications involve mostly the exchange of parts. This offers a very elegant procedure for the inhomogenous models. Only the updated part, including its spot welds, has to be deleted and the new one is directly read into the model, including its spot welds. Simple exchange of parts, which are already modeled, is now accomplished in a few minutes.

Another useful feature of the new approach is that all meshes can be stored in a database and used over and over again once the part is referenced by a given vehicle configuration.

This could not be possible if the meshes depended on each other since the meshes would have to be changed each time.

The final important item is that the exchange of one part does not affect all the other parts as would be the case with mesh dependent models. The effects of part changes are easier to study because they are not disturbed by mesh changes of other parts.

5 Conclusion

The *linchweld* approach was developed and tested in 1997. It has been in continuous use at BMW for two years for all body simulation.

The new approach also enables features such as optimization of spot welds.

This approach shows significant advantages over all other known approaches. The benefit is derived from better correlation to test as well as cost reduction.

Time, Quality, and Management in the finite element sector are simultaneously and dramatically improved by 40 to 50%.

References

[1] Stefan Lammens. *Frequency Response based validation of dynamic structural finite element models.* Ph.D. thesis, Katholieke Universiteit Leuven, 1995.

[2] H.G. Vopel and J. Hillmann. Beruecksichtigung der schweisspunkte bei der fem-modellierung von karosserien. *VDI Berichte*, 1283:171–181, 1996.

[3] O.C. Zienkiewicz and R.L. Taylor. *The Finite Element Method*, volume 1: Basic Formulations and Linear Problems. McGraw–Hill, London, 4. edition, 1989.