Simulation and Optimization of Part Connections

R. Helfrich, I. Pflieger

INTES GmbH, Stuttgart, Germany

Summary:

The modelling of part connections essentially determine the quality of simulation results in Finite Element (FE) analysis. On the other hand modeling of connection details is sometimes complex and time consuming. Hence, analysts want to have simplified models for various connections giving satisfactory simulation results. Consequently, part connection is a typical FE modeling feature in the area of tension between modelling effort and result quality. Because the past years saw a growing use of automatic optimization to improve part behavior, part connections have to be included in the optimization process, too. This results in extended capabilities for the design of parts and assemblies.

In this paper, the focus lies on bolting associated with incompatible meshes and contact. After introducing the general concept of incompatible meshing and part coupling, modelling bolts and pretension of bolts is explained in more detailed. Then, shape optimization is used in conjunction with incompatible meshing in order to automatically find optimum bolt and rib positions in static and dynamic analysis.

Zusammenfassung:

Die Modellierung von Bauteilverbindungen bestimmt in entscheidendem Maße die Qualität der Simulationsergebnisse. Andererseits ist die Abbildung von Verbindungsdetails aufwändig und man sucht daher nach vereinfachten Verbindungsmodellen mit ausreichender Genauigkeit. Bauteilverbindungen befinden sich also im Spannungsfeld zwischen Modellierungsaufwand und Ergebnisgüte. Nachdem sich in den vergangenen Jahren in der Simulation auch die Optimierung von Bauteilen immer weiter verbreitet, sollen natürlich auch die Bauteilverbindungen in den Optimierungsprozess eingebunden werden. Daraus ergeben sich erweiterte Potenziale für den Entwurf von Bauteilen und ihrem Zusammenbau.

Hauptgegenstand des Vortrags sind Schraubverbindungen. Dabei spielen auch inkompatible Netze der verbundenen Teile und Kontakt in der Verbindungsfläche eine wesentliche Rolle. Der Nutzen dieser Vorgehensweise bei der Optimierung von Schrauben- und Rippenpositionen in statischen und dynamischen Anwendungen wird vorgestellt.

1 Introduction

The modelling of part connections essentially determines the quality of simulation results in Finite Element (FE) analysis. On the other hand modeling of connection details is sometimes complex and time consuming. Hence, analysts want to have simplified models for various connections giving satisfactory simulation results. Consequently, part connection is a typical FE modeling feature in the area of tension between modelling effort and result quality. The aim of simplified models of part connections is to meet the global behavior of a structure first. The local behavior is often left to a separate investigation where refined models of connection areas are used. If the number of part connections is very high (like for spotwelds in a body-in-white), then a separation between local and global models is not the best way, because modelling effort for many local models is not affordable. There, a clear requirement is to improve simplified models in order to fulfill minimum quality standards for local behavior without increasing modelling effort.

In this paper, the focus lies on bolting associated with incompatible meshes and contact. After introducing the general concept of incompatible meshing and part coupling, modelling bolts and pretension of bolts is explained in more detailed. Then, shape optimization is used in conjunction with incompatible meshing in order to find optimum bolt positions automatically. Afterwards, engine analysis with bolt pretension is introduced and how one can make the transition from static to dynamic engine analysis by contact locking. Finally, shape optimization of engine block rib positions by application of incompatible meshing is shown to reduce sound radiation power of engine block surface. The paper ends with some concluding remarks.

2 Part Coupling by Incompatible Meshes

The concept of incompatible meshes facilitates modeling of complex parts and assemblies. Among others, the following applications benefit from incompatible meshes:

- Local mesh refinement,
- Coupling of parts meshed with different element types,

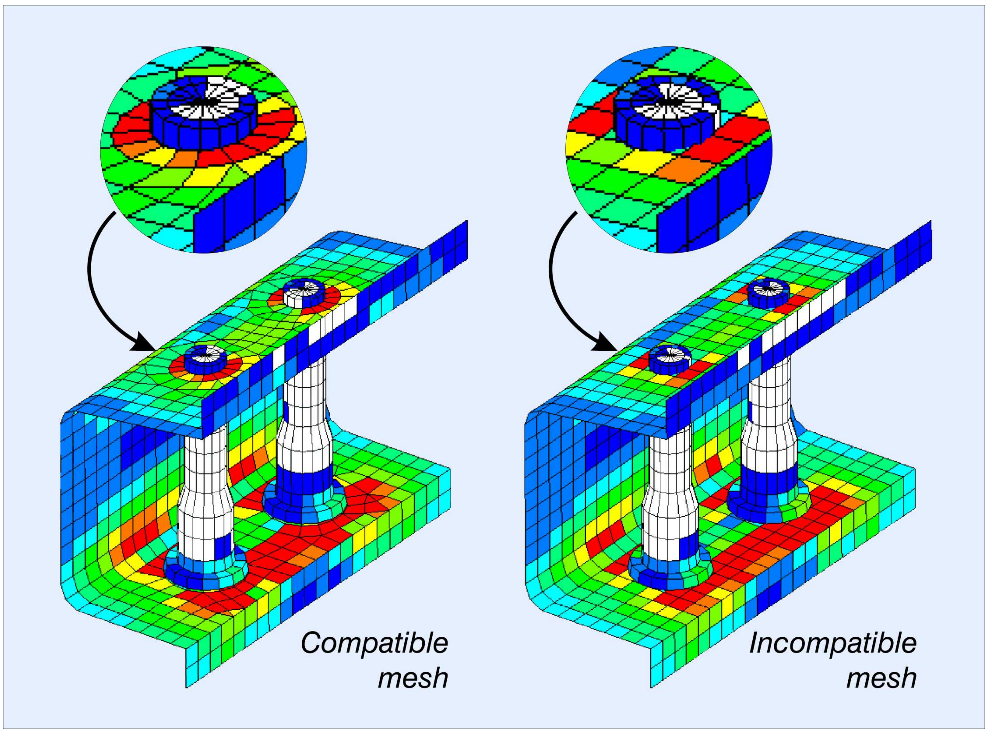

Fig. 1: Bolt connection with compatible and incompatible meshing

- Separately meshed fasteners facilitate an easy change of position and distance of the bolts (see Fig. 1),
- Separately meshed ribs facilitate an easy change of position and orientation of the ribs (see Fig. 2),

- Automated generation of spotweld elements and their coupling with incompatibly meshed flanges,
- Contact analysis between differently meshed parts (see Fig. 8).

The merit of this concept is the substantial raise in productivity of FE modelling. The advantages of the concept are

- mesh-independent modelling of complex parts,
- faster exchange of parts by keeping other already meshed parts unchanged,
- more flexible modelling by part-oriented meshing,
- improved mesh quality by accurate mesh transitions,
- applicable for all analysis types.

One has to keep in mind that incompatible meshes concentrate the mesh transition in the coupled surface. So, secondary results like stresses are not as reliable as in other undisturbed areas. If durability has to be considered, where accurate stress values are required in the mesh transition area, then either compatible meshing or submodelling techniques are an option. But this option is not available with spotwelds, because the number of spotwelds in a model can be so huge. Altogether, it is the job of the analyst to select and justify the appropriate method of meshing dependent on the task characteristics and results.

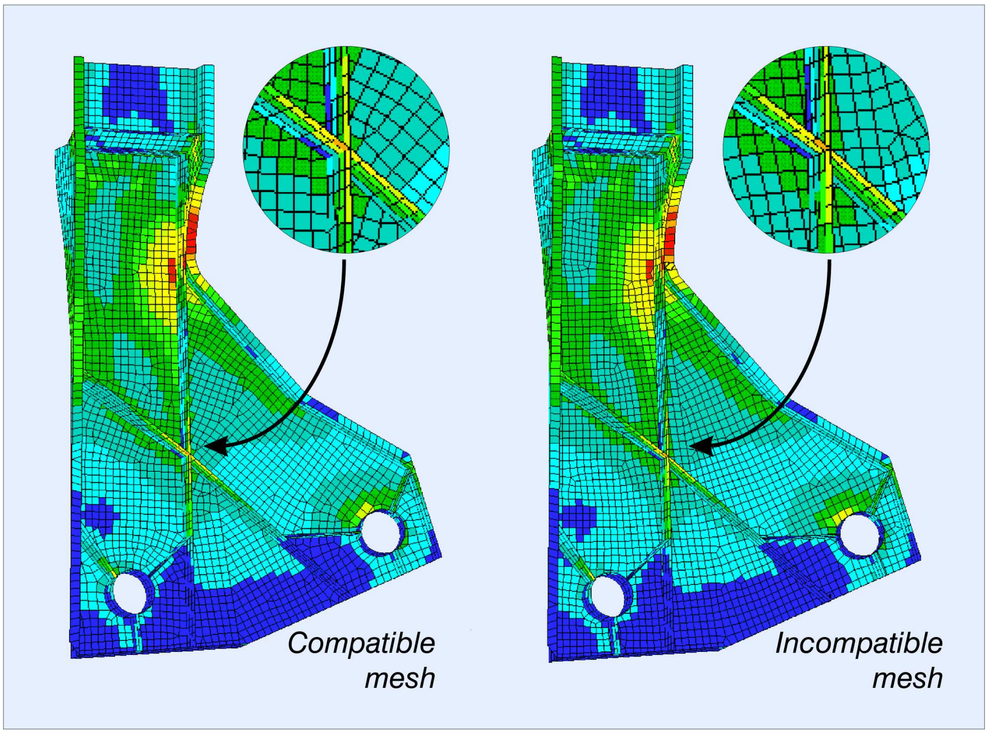

Fig. 2: Ribs on body with compatible and incompatible meshing

3 Modelling Bolts

There is a wide variety of modelling options for bolts due to different application cases (see Fig. 3):

- **Beam model**: The end nodes are coupled to the structure using rigid regions. A bolt hole is not modelled explicitly. A pretension can be induced by initial strains (e.g. by temperature defference). Two beams are necessary, if contact is used to pre-stress the bolt.

- **Bolt with cutting plane**: A solid model of the bolt is used without thread. There is an explicit bolt hole. The thread area and bolt hole are coupled (e.g. by node-to-node or surface-to-node coupling). In the undisturbed part of the bolt a cutting plane is used with contact between two nodes to model a pre-stressed bolt. This is not recommendable for short bolts due to the warping of the cross section. Bolt pretension leads to correct stresses in a long bolt but also to a shortening of the bolt which is not physical.

- **Bolt with thread coupling**: A solid model of the bolt is used without thread and a bolt hole or nut is modelled explicitly. An automatic procedure generates coupling between hole and thread, where both bolt and bolt hole or nut remain elastic and no artificial constraint is introduced. In addition, pitch and flank angle of the thread can be specified and is taken into account. This leads to non-axial thread forces which cause an opening of the bolt hole. The coupling procedure also induces pretension by contact in the thread region. Therefore, this method is correct for short bolts as well. The bolt is elongated and shows correct stresses.

- **Bolt with thread modelling**: A solid model of the bolt with thread is used and contact is required along the full thread. Of course, the bolt hole has to be modelled accordingly. The mesh of the thread is rather coarse and cannot respresent stresses in the thread properly. Pretension cannot be induced easily but just by rotation of the bolt.

- **Bolt with precise thread modelling**: This adds a detailed mesh of the thread to the previously mentioned approach. In this way, stresses in the thread are represented very good. Nevertheless, the effort needed for modelling and analysis makes this approach only suitable for a detailed analysis of a single bolt connection.

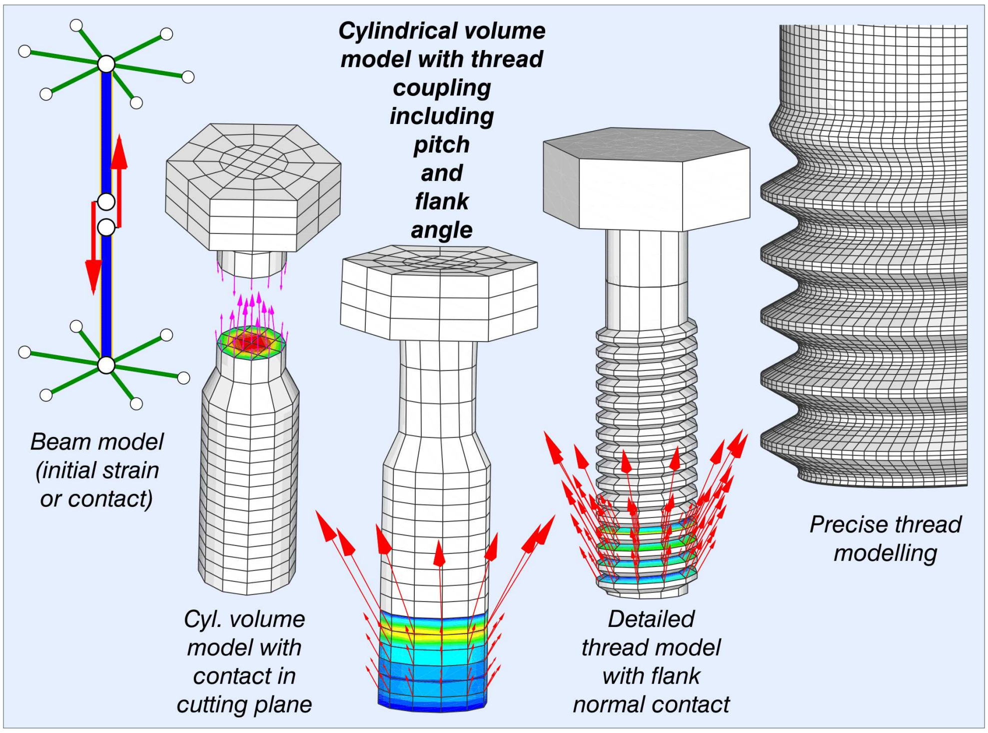

Fig. 3: Wide spectrum of bolt modelling concepts

From these different examples one can easily draw the conclusion that the bolt with thread coupling represents a good compromise between modelling effort and quality of results. Fig. 4 shows three application cases where bolt elongation, bolt hole expansion, and bolt torsion are considered. More effects include left-hand and right-hand threads as well as threads with single and double helix.

It is worth mentioning that the meshing of the bolt has an influence on the distortion of the bolt. A first condition is a symmetric mesh of the bolt in order to avoid bending effects just by elongating forces. A second condition is to use brick elements instead of tetrahedra, because tetrahedra deform in preferred directions and not uniformly in all directions. So, automatic tetrahedra meshing of bolts and nuts is not recommendable.

On the other hand, bolt head and structure as well as bolt and bolt hole or nut need not to be meshed compatibly but allow for incompatible meshing, too.

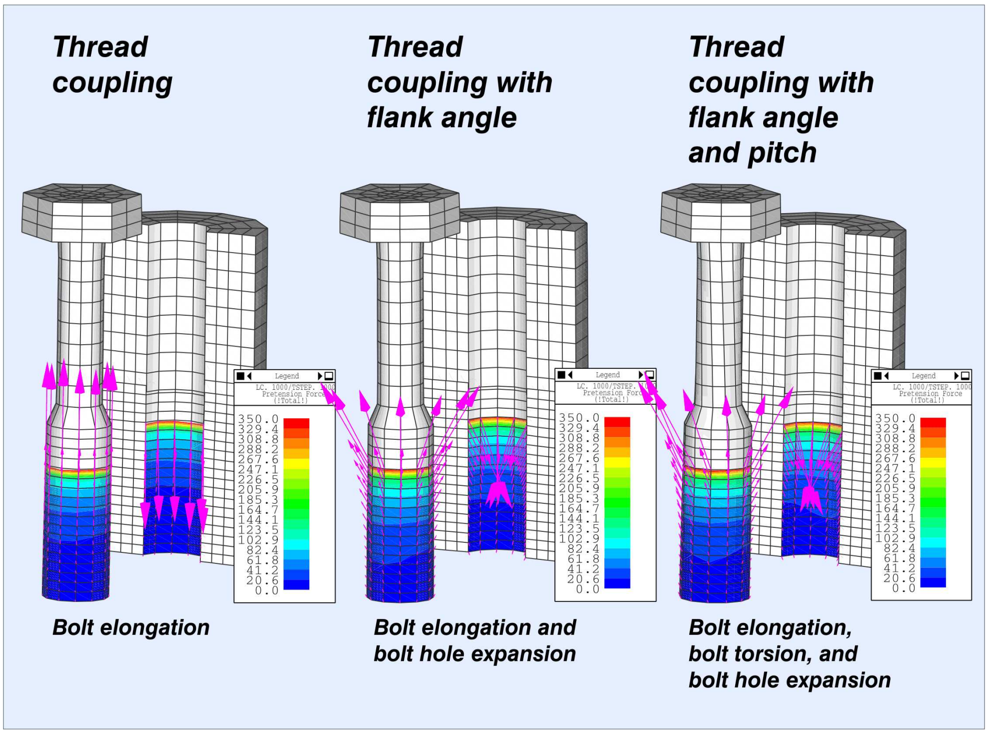

Fig. 4: Simplified bolt model with pitch and flank angle

4 Bolt Pretension

A good example for the advantage of thread coupling against the cutting plane in a bolt is the bolt connection of metal sheets applying short bolts. In this case, thread coupling between bolt thread and nut enables the warping of the bolt cross section which can not be accomplished by cutting the bolt. Fig. 5 shows the distortion of a bolt under pretension and a subsequent pressure load.

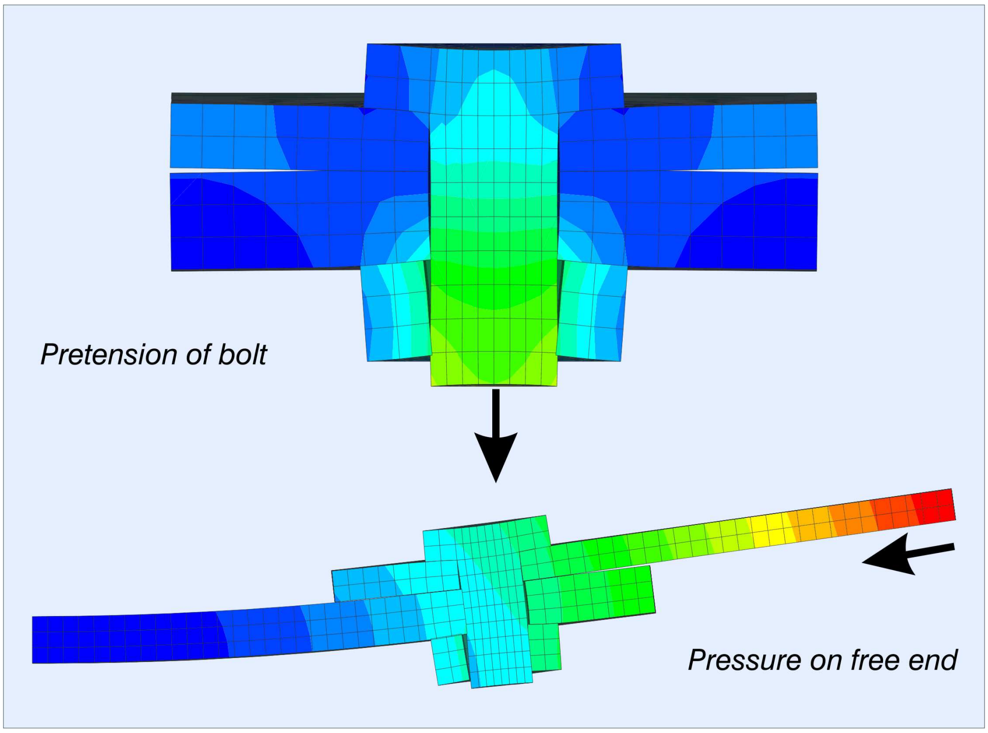

Fig. 5: Pre-stressed bolt under pressure loading

The bolt pretension leads to a contact zone under the bolt head and nut. But ouside this area we see a lift-off, where the sheets show no contact any more but a gap.

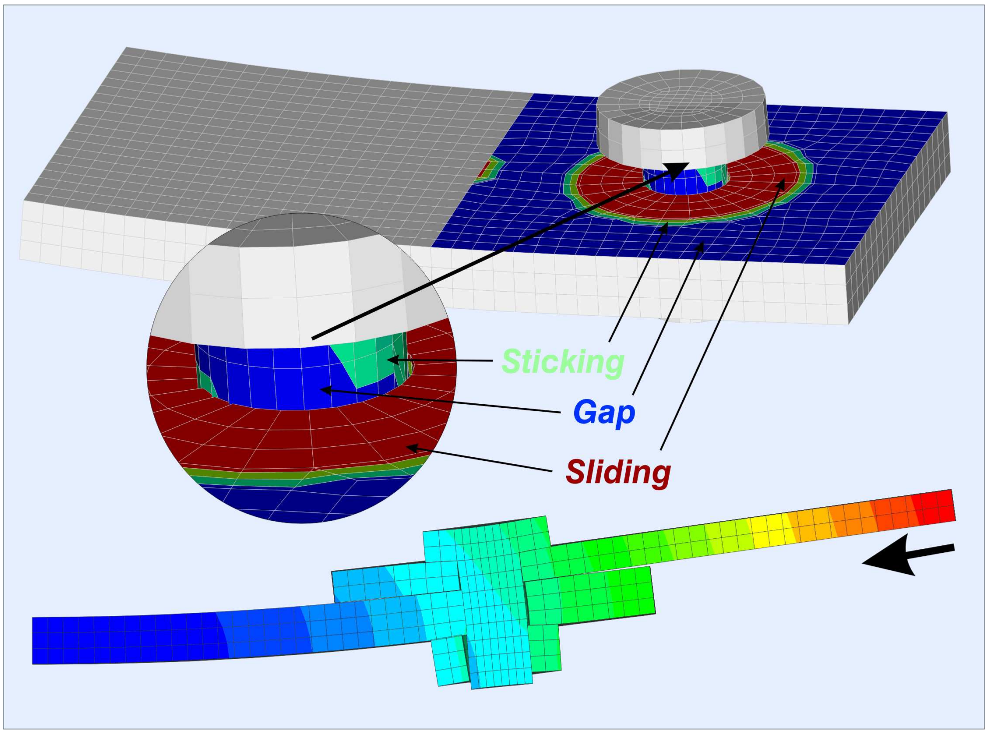

Fig. 6: Contact status of pre-stressed bolt connection

This example also shows the sliding effect between the connected metal sheets. Due to the high pretension force in the bolt, there is frictional contact between the metal sheets. This frictional contact is sticking after the pretension of the bolt is applied and changes to sliding when the pressure load is applied. In addition, the upper sheet gets in contact with the bolt shaft (see Fig. 6). This also holds for the lower sheet.

Outside the frictional area, there is no contact between the sheets due to the pretension of the bolt, except where the bending of the lower sheet meets the upper sheet at one touching node. There, sliding occurs, too.

5 Optimization of Bolt Positions

Fig. 7 shows an example for the fruitful combination of bolt connections, incompatible meshing, and shape optimization. The bracket is loaded by two forces on the upper blade and supported at the rear end of the lower blade. Both blades are connected by three bolts in order to apply the stiffness of the lower blade to bear the external forces by the full bracket.

The modelling of the bracket and the bolts is fully incompatible, i.e. the bracket is modelled without any consideration of the bolts, and the bolts are modelled without any perturbation of the bracket mesh.

Shape optimization is used to move the three bolts from their initial position to an optimized position, where all three bolts show the same bolt force. The diagram in Fig. 7 shows the bolt force history during the optimization process leading to the final position of the bolts. During this process, penetration of bolts is prevented. The final position of the bolts shows a symmetric configuration as it should be.

This is an example for a shape optimization without any mesh modification, i.e. the relative position of all nodes is retained. But it is still a shape optimization, because the coordinates of the bolts are modified.

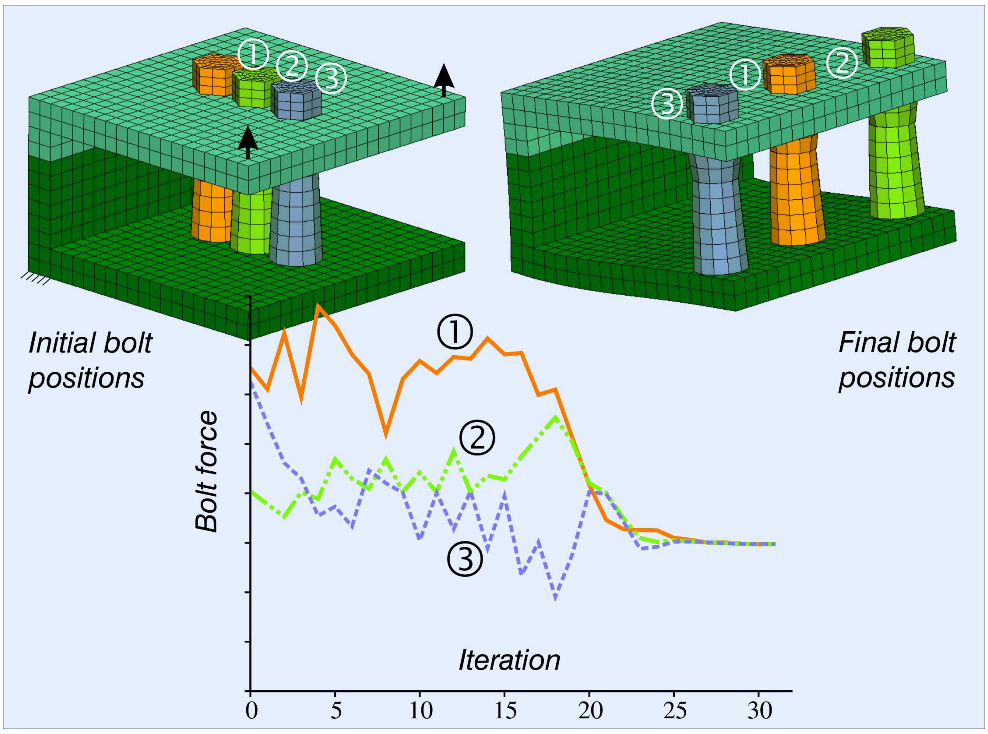

Fig. 7: Optimization of bolt positions in a bracket

6 Engine Model with Bolt Pretension

In engine analysis both incompatible meshing and bolt pretension are frequently used. Fig. 8 shows a typical engine model with cylinder head, engine block, liners, bolts, and gasket.

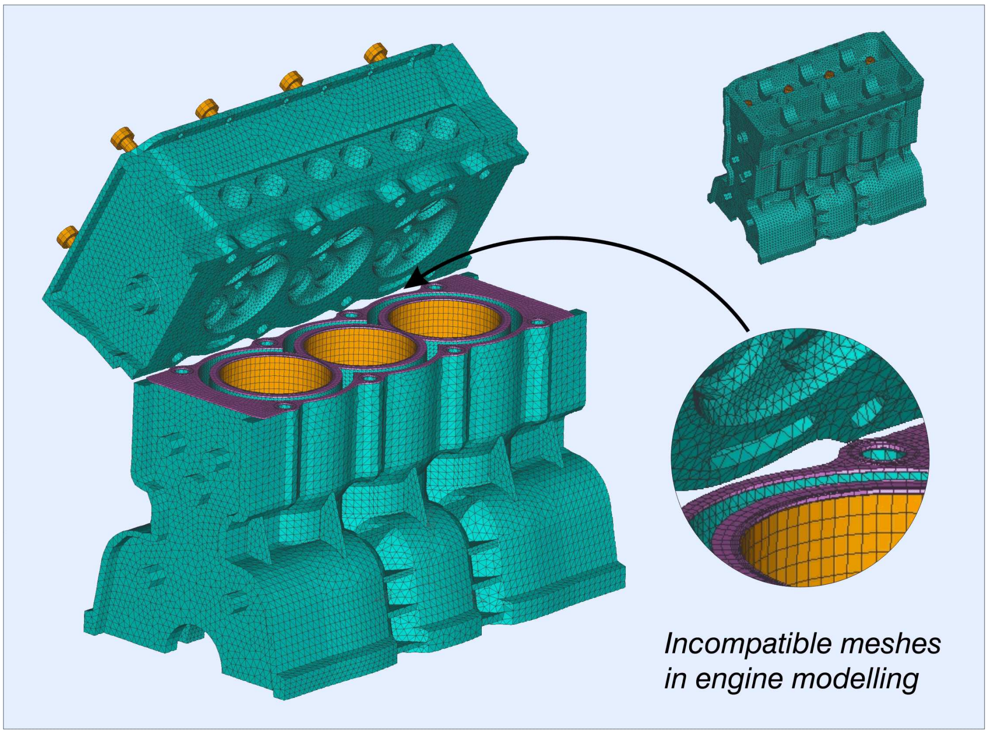

Fig. 8: Engine model with cylinder head, engine block, liners, bolts, and gasket

While the meshes of cylinder head and block are generated with tetrahedra elements, liners are modeled with hexahedra elements, and bolts and gasket parts are modelled using hexahedra and pentahedra elements. All connected surfaces show incompatible meshes. In addition, contact specifications are used between gasket and cylinder head as well as between bolt head and cylinder head.

Fig. 9 shows the effect of bolt pretension on the equivalent stresses. The bolts are highly stressed under the bolt head due to extensional and bending forces and at the beginning of the thread.

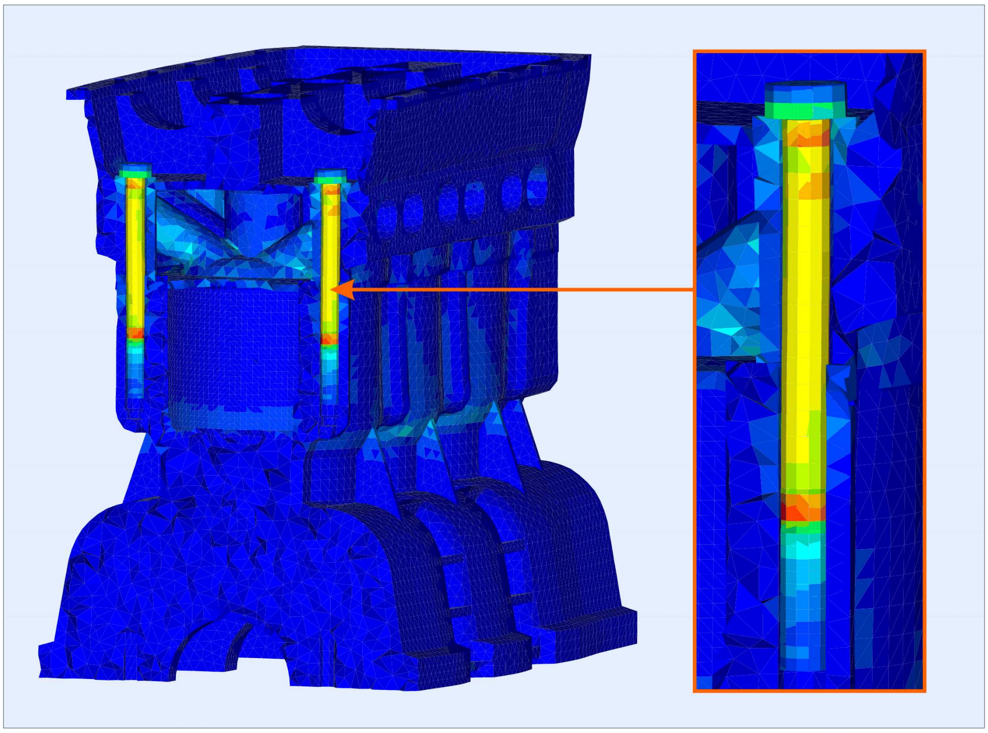

Fig. 9: Stresses induced by bolt pretension

7 Engine Sound Radiation

Beside a thermo-mechanical analysis of engines, vibrations of engines are also very important, e.g. to reduce sound radiation power of engine blocks. Before bolt pretension the parts of an engine can vibrate separately, but after bolt pretension the engine assembly vibrates as one single body. From an analysis point of view, this behavior is achieved by locking the contact areas where the parts are in contact. Other areas are kept uncoupled, where no contact is in place.

Fig. 10 shows the sound radiation power at the block surface as a frequency response function due to an harmonic excitation at the cylinder head. The upper curve with the significant peaks is obtained from a modal frequency response after contact locking as described above. The lower curve is obtained after full coupling of the surface between gasket and cylinder head. The differences are highly significant and put emphasis on the coupling strategy between static and dynamic analysis.

Sound radiation power density is a result showing potential areas with high acoustic radiation activity on the surface of a structure. It depends on the square of the surface velocity normal to the surface. From Fig. 10 the lower mid-side of the engine block surface can be identified as most acoustically active surface at the frequency with the highest peak for the sound radiation power summed up over the full engine block surface.

Now, shape optimization is used again applying incompatible meshing in order to reduce the amplitude of the frequency response function for the sound radiation power by moving the ribs on the engine block surface. In Fig. 11 the left engine shows the initial position of the ribs and the right engine shows the optimized positions of the ribs. Because the most active surface area is in the middle of the engine block surface, shape optimization is moving the ribs to the center of the surface. The new response function shows a slightly reduced amplitude while the peak is shifted to a higher frequency.

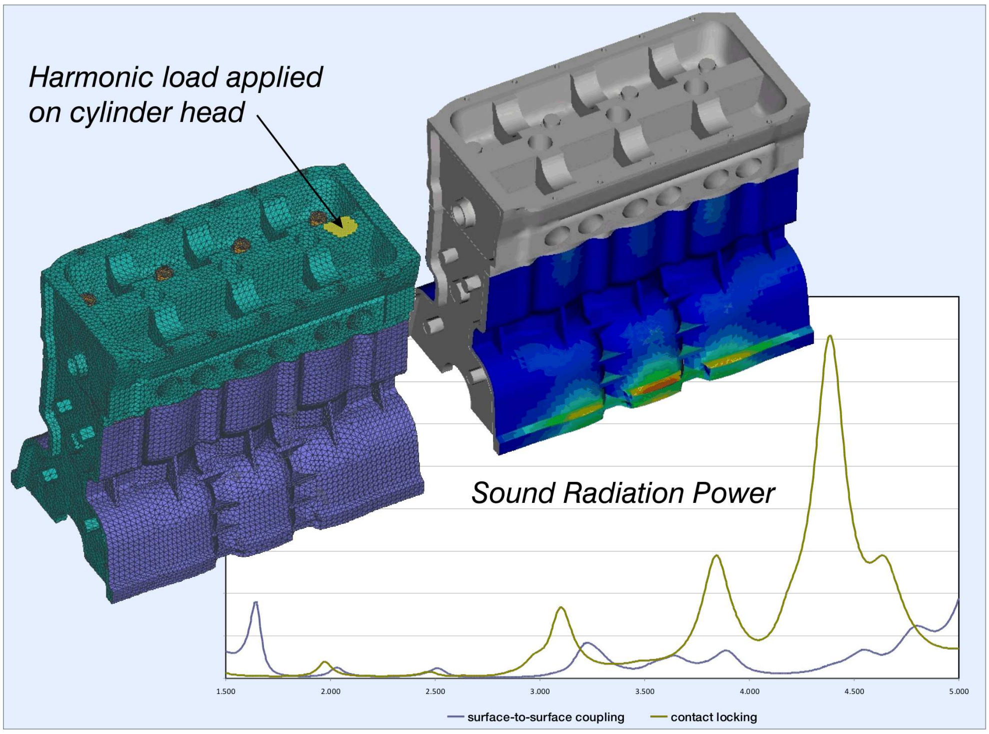

Fig. 10: Sound radiation power on partial engine surface - Frequency response function

This again is an example for a shape optimization with incompatible meshes without any change of the engine block or rib meshes. The ribs are just moved on the engine block surface in order to fulfil a number of specific constraints.

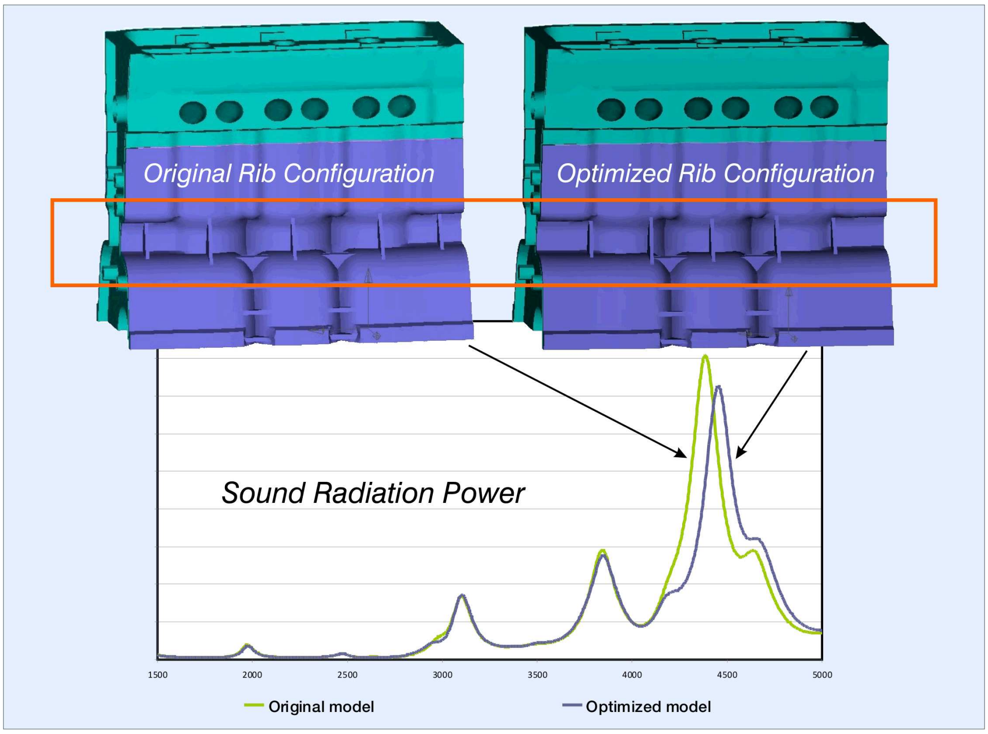

Fig. 11: Optimization of rib positions to reduce sound radiation power amplitude

8 Conclusions

This paper shows the combination of some innovative simulation features in the connection of parts in an assembly:

- Use of incompatible meshing,
- Bolt modelling and pretension with pitch and flank angle,
- Shape optimization for bolt and rib positions,
- Dynamic vibrations of pre-stressed parts with contact.

This combination enables simulation driven design by reducing modelling effort for connections, by facilitating the generation of model variants, by getting results for assemblies faster, and by improving behavior of parts and assemblies by applying automatic optimization techniques.

Full benefit of simulation driven design can be exploited by a unified FE software integrating all required analysis features including a full set of optimization functions. Besides, integration of all analysis functions with optimization saves repetitive operations and gives the shortest possible run times.