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Simulation and Optimisation of Engine Noise – Predictive Input for the Development Process

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Introduction

Quiet vehicle engines and power units are required to reduce noise pollution in the urban environment and to fulfil the rising comfort demands. Recently developed simulation methodologies and software tools provide detailed information for vibro-acoustic analyses of engines The efficiency in the acoustic development of new engines and power units for modern vehicles can be significantly improved by capable prediction methods. While in the past, the acoustics of power trains were analysed at a rather late stage, the challenge of today is to improve optimisation of acoustics and vibration in the design phase of the development.

Results of numerical simulation of power units, as complex as they may be today, will no longer be satisfactory unless they allow conclusions to be drawn with respect to the specific stages of the development process. The requirements are to predict specific vibro-acoustic phenomena on one hand and to provide conclusions for optimum solutions on the other.

Due to increased efforts for detailed modelling of elastic multi-body systems and relevant non-linear body contacts, simulation models have become very complex and a high amount of computation time is necessary to calculate accurate results. Thus, another challenge is to deliver the results in the short time required for a fully integrated simulation solution in the different stages of the development process.

Various software tools have been developed and different application methodologies have been introduced in order to extend the knowledge of the virtual vibro-acoustic behaviour of new power units. Results have been published showing structure borne noise and air borne noise results calculated with simulation models for engines and gear boxes. The methodologies used for such simulations are Finite Element Method (FEM), Multi-Body Dynamics (MBD) for the structure borne noise and Boundary Element Method (BEM) for the noise radiation simulation. While, with a sufficient model quality, structure borne noise simulations allow for absolute result assessment, until now air borne noise results can only be used for trend predictions.

A number of specific applications are already part of the actual Engine Development Process (EDP). The experience with AVL`s software for NVH simulation in the EDP of AVL is described in the following. This software includes AVL EXCITE for engine dynamics and NVH, AVL TYCON for drive train dynamics and AVL GLIDE for piston and ring dynamics [1, 6].

1 Aspects of vibro-acoustic simulation in the engine development process

1.1 The interdependency of vibro-acoustic simulations and development process

Numerical simulation by means of MBD was originally started with rigid elements (bodies) having global motions. The coupling conditions between the bodies (joints) could be defined as linear and/or non-linear boundary conditions. Based on such models global behaviour for complex dynamic systems could be analysed. Boundary conditions e.g. displacements and loads for each body under engine running conditions could be obtained for further detailed investigations (strain, stress, etc.) on these bodies. The software tools for these applications work in frequency or in time domains. The advantage of the calculation in time domain is that highly non-linear conditions can be taken into account.

This method (rigid MDB) has been used to pre-calculate the dynamic loads generating the acoustic excitations due to valve train, timing chain or belt, gear train and piston slap. In a second step the flexibility of the engine and the gear box has been considered by FEM models, using the pre-calculated loads as excitation boundary conditions.

Further progress was to represent almost all moving parts in the power unit as flexible bodies. Usually, body geometry, elasticity and mass are discretised by a FEM model and the vibration behaviour of the flexible body is represented by the stiffness and mass matrices of their unmoved and undeformed FEM models. In order to simulate the local vibration behaviour of each flexible body in each step of a global movement, the effects of the relative motion has to be taken into account. For example, the interaction between the oil film pressure distribution in the big end bearing of the con-rod and its deformation by mass forces requires to consider the interaction of global motions and local deformations in each time step of the calculation [4].

The enhancement of the flexible MBD has always been driven by new or extended applications. Although, the simulation effort has increased due to the complexity of the required models, the field of application and the result quality have been significantly improved for the engine development. Besides vibration and acoustics analyses, strength and fatigue analyses of engine parts e. g. crankshaft, connecting rod, main bearing wall and even piston under dynamic loads became possible. Due to the increased model size, condensation procedures are necessary to reduce the number of degrees of freedom for the MBD simulation and therefore calculation time.

To improve the quality of the contact load values, detailed joint models in contact areas of bodies have been developed e.g. elasto-hydrodynamic (EHD) joints for slider bearings and for the piston-to-liner contact or joints representing the frequency-dependent behaviour of power unit mounts. For NVH and crankshaft dynamics applications new enhanced HD joints may replace EHD ones to accelerate the calculation. The highly sophisticated EHD joints are required for more detailed analyses of the contact dynamics in the bearings and between piston and liner. They are important for a more detailed input to assess the design of these components in the EDP and enable design optimisation to approach closer to the strength limits and to improve fulfilling design demands like engine weight and cost reduction.

The use of recently developed complex models and methods change both the work of the calculation engineers and the interaction with the EDP itself. Each application procedure in the EDP is described in a specific workflow. Furthermore, the modern numerical simulation of flexible body dynamics requires professional software development. New software packages for numerical simulation have been designed to integrate all possibilities of the former MBD software tools. Their generic approach for the dynamic behaviour of the engine parts and contacts aims to analyse the low frequency engine mount vibrations, the high frequency structure borne noise, the strength evaluation for crankshaft and connecting rods, the durability of the slider bearings, etc. with one software package.

The disadvantage is that the set up of the rather complex simulation models requires more time and input data. This fact creates a conflict with the time frame of the EDP. Although, the simulation results can be obtained with high accuracy, their impact on the design of the engine may come too late in the EDP. Therefore, even today, rather simple calculations for torsional vibration and static strength analyses are often preferred to define the main dimensions of a new or enhanced engine in the important concept phase. Complex multi-body dynamic simulations remain for the design phase. Their results lead to local modifications of the design only and must not impact the main dimensions of the engine.

1.2 Integrated model approach with one software package

To solve the conflicting demands described above an integrated model approach for the whole EDP has to be used. The software solution has to be completed by an intelligent modelling technique for the power unit, (Fig. 1).

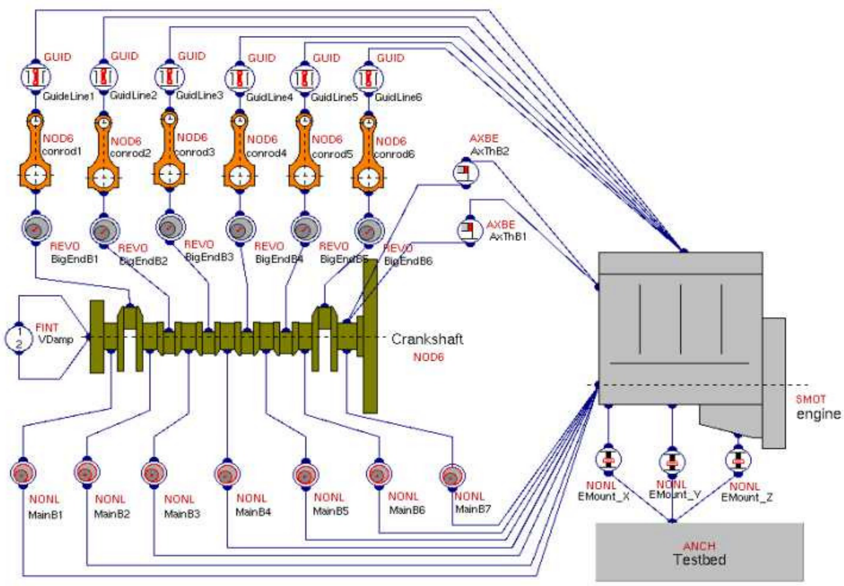


Figure 1. Graphical display of connections between moving and vibrating parts (GUI) for the model of a 6 cylinder engine.

The entire model consists of a number of part or sub-models. The part models have different levels of detailing with respect to the phase in the EDP. Data and model levels can be exchanged for different applications during the engine development. The strategy is to have *one* software, *one* model, but adapted levels of model detailing which allow questions to be answered in the specific EDP stages.

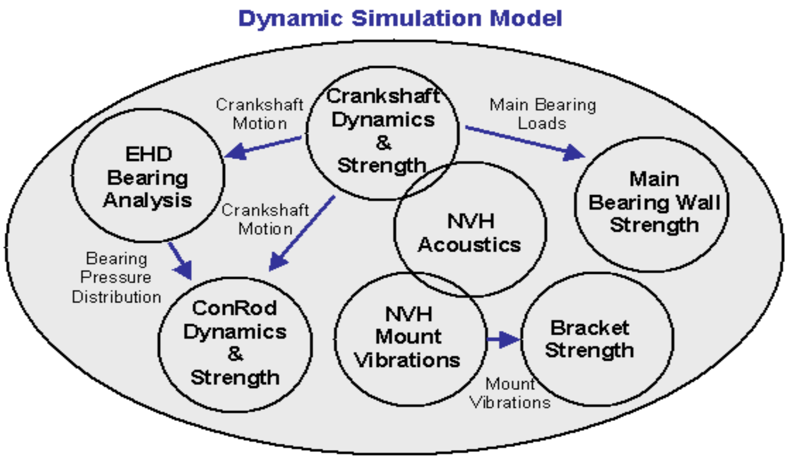


Figure 2. Integrated model approach for various simulation targets with *one* software package.

Fig. 2 shows the dependency of various applications on the integrated models: The numerical simulations of crank train dynamics, low frequency engine mount vibrations and engine structure borne noise (NVH) are started in parallel. Investigations of crank train dynamics focus on the general dynamic behaviour and the crankshaft strength. For these targets detailed flexible models of crankshaft and connecting rods, and a coarse model of the engine block are required. Basic low frequency analysis of engine mount vibrations is investigated using a rigid model of the power unit, while, the investigation of structure borne noise requires a detailed flexible model.

The consecutive build up of the simulation model and the calculations can be described as follows: First, the crank train and coarse block models are finished for the investigation of crank train dynamics. In parallel, the detailed FEM model of the entire engine including all main auxiliaries is generated. As soon as this entire engine model is finished, it replaces the coarse model of the engine block Fig. 3.. Combined with the crank train model, the simulation of engine structure borne noise can be carried out. At the same time, the investigation of the strength of con-rods and main bearing walls can be enhanced replacing the simple joint models of the bearings by detailed EHD ones. The dynamic loads for the parts are taken from the former results. The final step is the completion of the model with the gear box and all additional auxiliaries in order to perform the final low frequency analysis of the entire power unit, Fig. 3.

By correlating model effort and request of specific results at a certain time, the EDP can be improved significantly. The results of the simulation can impact the design process more efficiently. The use of one software package ensures that part models can be replaced and refined any time additional data is available and extended modelling effort is justifiable. These measures improve the EDP by increasing the quality and reducing the process time.

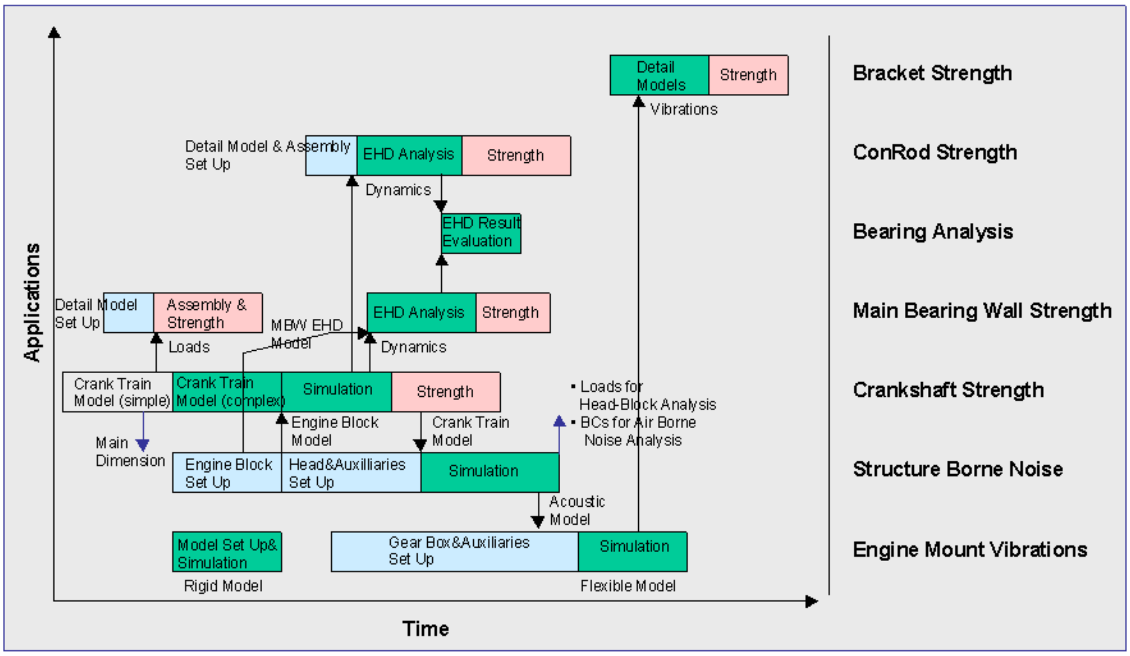


Figure 3. Correlation between different simulation steps in the EDP considering the integrated model approach.

The requirements on the flexibility of the simulation software package are very high. It must be possible to represent all (elastic) bodies and joints of the vibro-acoustic model, i.e. different types of models, simple and complex ones. The exchange between the different types of models must be easy and fast in order to follow requirements inside the EDP. Standard procedures have to be defined and guidelines with detailed workflow descriptions are required. The expected quality of the results (absolute or relative prediction) has to be well proven by verification of the calculation results with experimental data. For example, the prediction of absolute surface noise levels requires a different simulation model compared to the prediction of the trends between two design modifications. Specific simulation procedures have to detect noisy regions of a structure and give a relative assessment of efficient measures to reduce vibrations.

Fig 4 gives an overview of the relative effort in the workflow for NVH simulation by means of the software Excite. The continuous steps for modelling, calculation and evaluation are considered differentiating between applications for a single analysis, for the speed range and for design modifications of an engine.

The extension of a high level software is mainly driven by non-standard applications. These test applications with e.g. new joint models, may be due to requests of the EDP or due to problems of a running engine. First, they have to be tested and the simulation results have to be validated by measurement. In a second step the new applications may become part of the future EDP. In general, the comparison of simulation and measurement is of basic importance to define the quality of results and define application standards. The entire workflow has to be considered starting with setting up the models, collecting required input data and ending with final conclusions. In addition it is necessary to consider all calculation loops for modifications and for the entire load and speed range to be investigated. Therefore, it is insufficient to focus on only reducing CPU time or meshing efforts. In fact, focus has to be setshould be on fast and automated post-processing, bearing in mind the high amount of result data for complex simulations produced, as well as the possibility to assess trends of different design variants.

A valuable NVH prediction in the EDP requires the knowledge of the level of accuracy the simulation can provide. The level of accuracy of simulation models and methods used has to be investigated and proven in a separate research process. Detailed comparisons between measured and simulated results have to be carried out. Furthermore, calculation procedure and data processing have to be verified. As a result of validation and methods development, standard methods and workflows can be derived for the EDP.

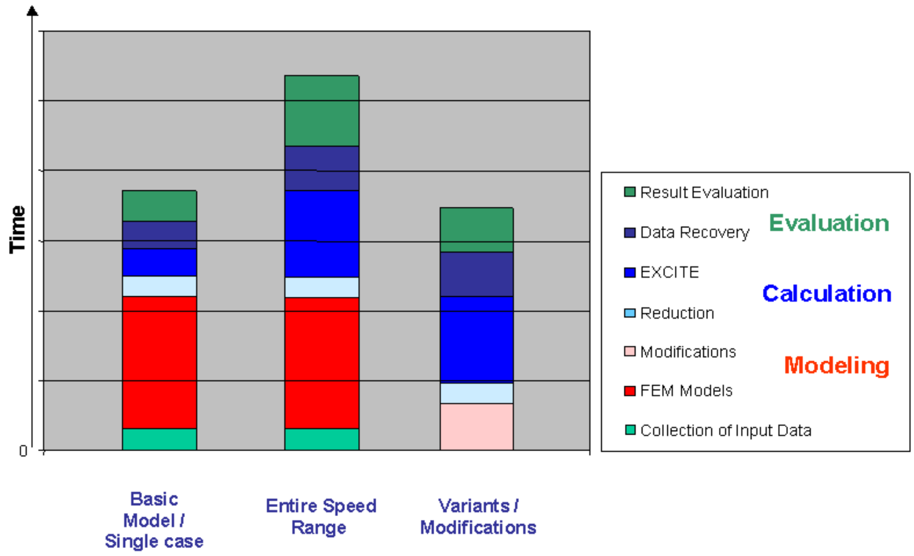


Figure 4. Time effort and diversification of all workflow tasks for one case considering design variants and entire speed range

The development of new methods for non-standard simulations and the improvement of existing standard methods are usually placed in different departments. It is an optimal situation, if the software used is developed in the same company, as the departments for software development and application will profit from direct contact. The exchange of requirements and hints for the development is easier and new features can be tested and validated immediately. In this case the enhancements of the software is driven by the know-how of the internal application team.

2 Simulation process description and validation examples

The following chapter describes application and validation examples of NVH simulation. The examples show results of recently developed improvements of simulation technology driven by the requirements of the EDP at AVL.

2.1 Validation of structure borne noise results

The first example describes the results of the validation of engine surface vibrations of a 6 cylinder truck Diesel engine. Fig. 5 shows details of the engine model required in order to simulate structure borne noise up to 3 kHz. The model is discretised by means of FEM (software MSC NASTRAN). The required mesh density of the model depends on its modal behaviour in frequency domain. The higher the modal density of the structure parts in the analysed frequency range the smaller the maximum size of the FEM elements in the meshing. Automatic mesh generation accelerates the model build up significantly. For the vibro-acoustic simulation, the models of the crank train and the model of the engine are composed.

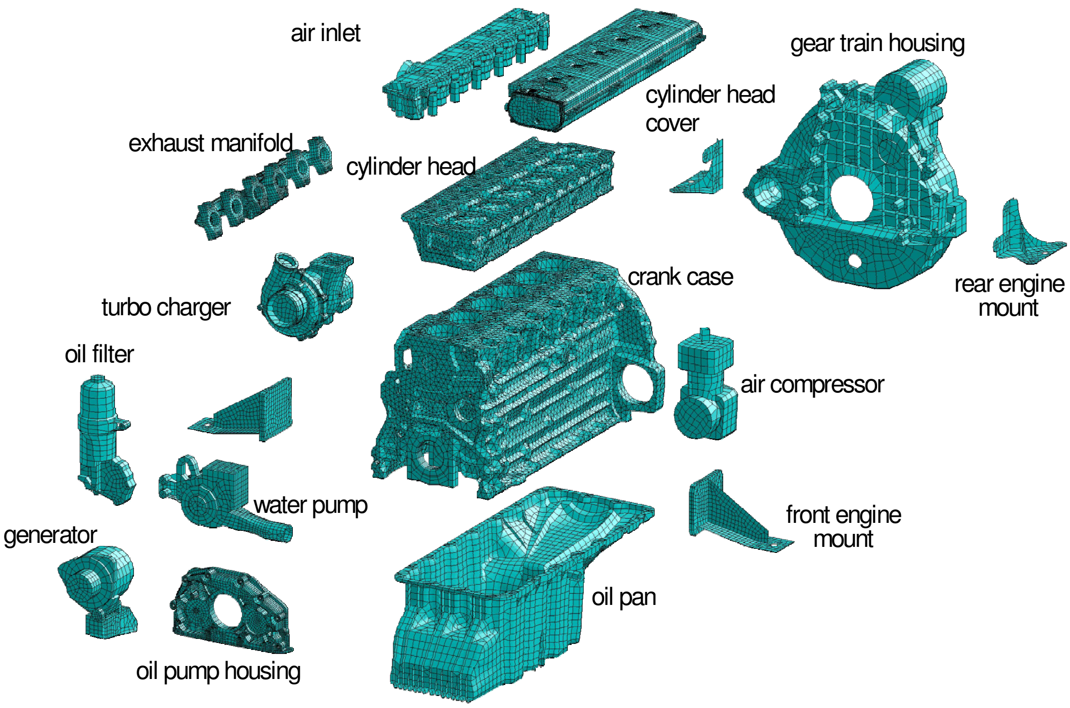
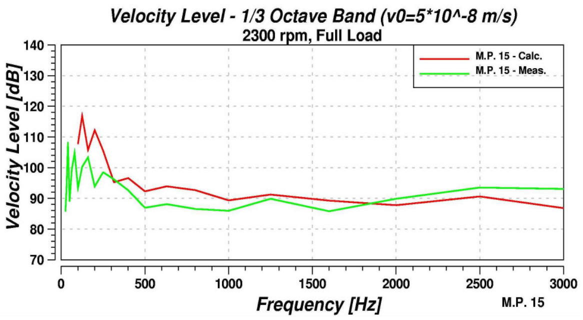
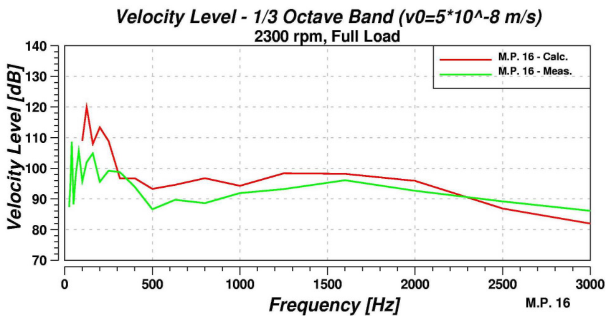


Fig. 5 FEM models of engine parts of a truck engine, w/o crank train, exploded view

The following forced vibration calculation is carried out with the software AVL EXCITE. The mathematical background principles of the workflow are described in various publications [1, 2, 3]. In the sliding contacts (bearings, etc.) contact joins are defined (see Fig. 1.) The vibration simulation is carried out in time domain for different speed and load conditions. The structure excitation is pre-calculated and considers the following sources of noise excitation: the gas load in the cylinders, the timing drive impacts in the cylinder head (valve seats, camshaft bearings, etc.) and the piston slap impacts in the cylinder block. The simulation results give the engine surface vibrations in time domain (perpendicular to the engine surface) and are converted to frequency domain by means of Fast Fourier Transformation (FFT).

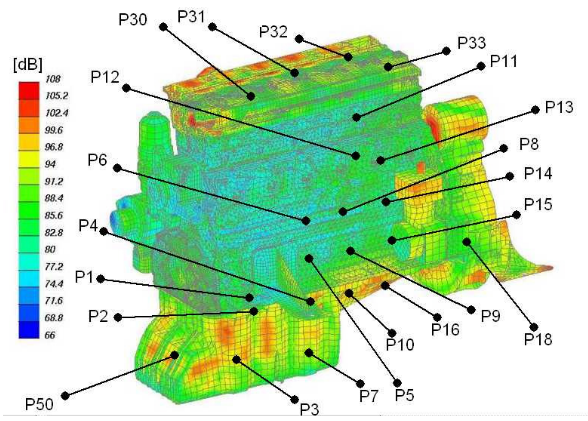
Fig. 6 shows typical simulation results up to 3 kHz. In the upper part of Fig. 6 velocity levels of two surface points are plotted in frequency domain. Both observation points are situated at cylinder no. 5, one in the cylinder centre and one at the oil pan flange. The comparison with results measured at the running engine at 2300 rpm and full load underlines the quality of the simulation results. The lower part of Fig. 6 shows integral vibration levels in the third octave band of 1,6 kHz. Comparing the results of measurement and calculation, the quality of results possible for the prediction of absolute noise levels for the EDP can be observed. This precision is presently in the range of about 3 dB and requires precise models for the structure and the excitation as mentioned in the previous part of the paper.





Comparison of measured (green) and calculated (red) normal vibration velocity levels at P 15

Comparison of measured (green) and calculated (red) normal vibration velocity levels at P 16



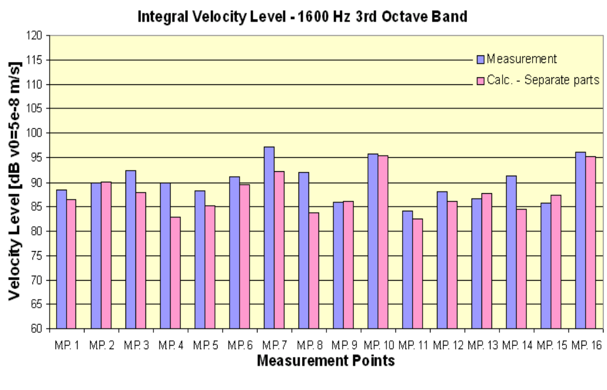


Figure 6. Observation points (measurement and calculation) and vibration velocity levels at engine surface 1/3 octave band of 1600 Hz; 6 cylinder truck Diesel engine, 2300 rpm, full load.

2.2 The quality of simulation models for acoustics

To check the quality of the structure model, modal analysis is performed. This analysis gives important basic information about the possible dynamic behaviour. If hardware is available, modal tests can be carried out. Usually comparison of measured and calculated results is carried out within the method development process to clarify and define model requirements. An example of a comparison is given in Fig. 7 and shows the model validation of a passenger car engine using the Modal Assurance Criterion (MAC). MAC compares the shapes of the natural modes. Any values greater than 0,7 attest good model quality. The upper frequency limit of relevant comparisons depends on the modal density, and thus on the relation of stiffness and mass distribution of the relevant engine part. In general the first 10 to 20 modes of each engine part can be resolved easily in the measurement. Comparing 15 modes means the model is validated up to about 400 Hz in the case of the oil pan, up to 1.1 kHz for the crank case and up to 3 kHz for the cylinder head (Fig. 7). At higher frequencies the modal density and damping in the test hinder a precise validation using the natural modes. Furthermore, the precision of loads and joint models are more important for the correlation at higher frequencies.

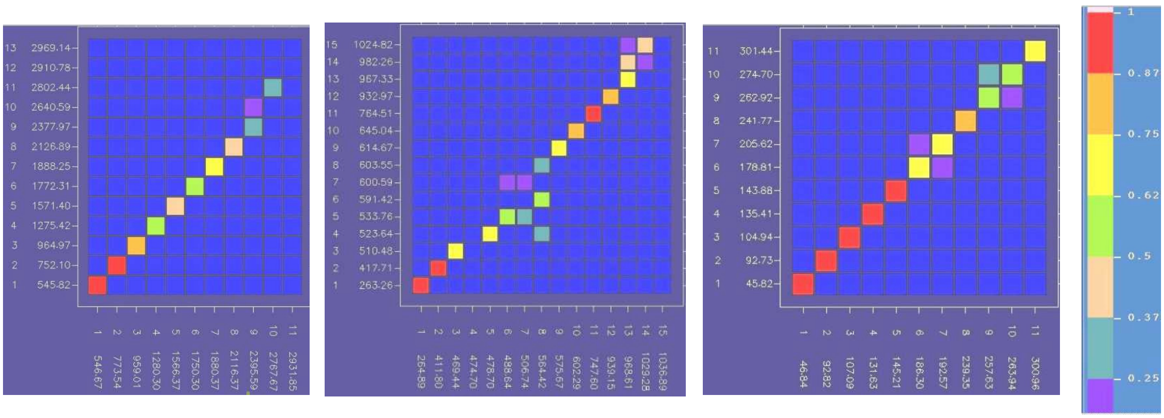
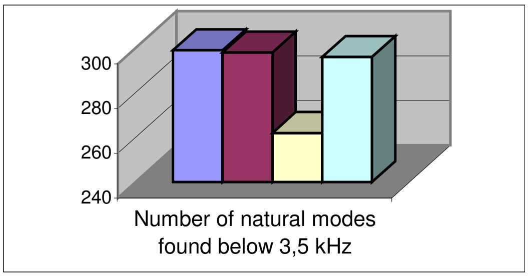


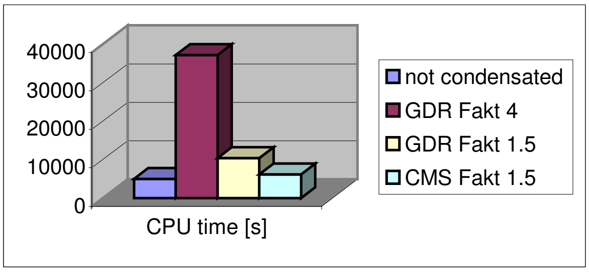
Figure 7 Model validation of a passenger car engine by means of the Modal Assurance Criterion (MAC): Cylinder head (left), crank case (centre) and oil pan (right).

Another key point for high model quality with efficient modelling effort is the condensation procedure. Condensation of structure models e.g. engine block, oil pan, is a requirement due to the high number of degrees of freedom (DOF) of the dynamic models. Because of the non-linear characteristics, the final entire model must not contain more than a few thousand DOF. Therefore, condensation of the model parts with more than a few hundred thousand DOF is required. Condensation is very time consuming in terms of CPU time.

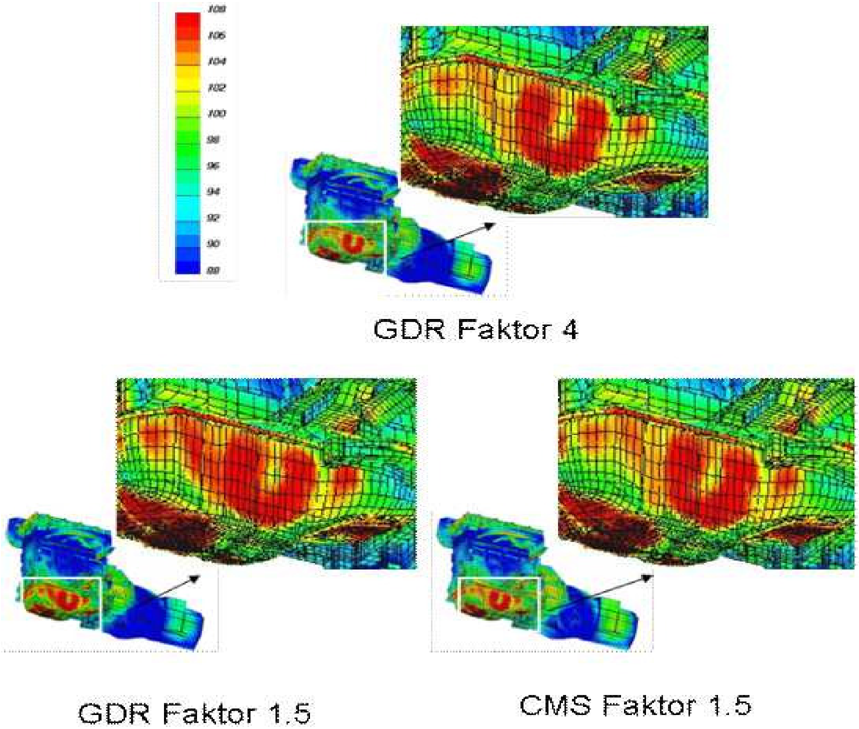
Various condensation methods can be used to generate the final calculation model for the power unit. Comparing the two most common methods, General Dynamic Reduction (GDR) and Component Mode Synthesis (CMS), a clear advantage can be seen considering the effort for CPU time and the quality of the results for CMS. If the aim for the condensed structure is to contain 97 % of the natural modes existing in the uncondensed structure, the condensation procedure using GDR Factor 4 (4 times the frequency range) requires about seven times the CPU required with the CMS method.

The influence on the results can be proven comparing the vibration velocities calculated with two models after two different condensations. The results are shown in Fig. 8 for an engine model of a 4 cylinder passenger car after data recovery to the entire FEM model for the octave band at 2 kHz. The differences in the results caused by the condensation quality can be seen very clearly in the oil pan area.





Comparison of quality (left) and effort (right) of different condensation methodologies (GDR = General Dynamic Reduction, *CMS = Component Mode Synthesis*)



Integral vibration velocity levels of a 4 cylinder passenger car engine, comparing the quality of different condensation methods to generate the final simulation model.

Figure 8 Comparison of quality and effort using different condensation methodologies

2.3 Noise excitation

Dynamic vibration analysis should consider the boundary conditions of each investigated model part, as each model has to cut out a certain area of the reality. All effects relevant for the NVH result have to be taken into account at the boundaries. While MBD software considers internal effects e.g. inertia forces and gyroscopic loads commonly, specific excitation loads have to be defined externally: cylinder gas pressure, piston secondary motion and timing drive excitation (belt, chain or gear train excitations). Gas pressure is directly applied to the gas chambers and pistons of the simulation model. For the other excitation sources, dynamic loads for the simulation can be provided by pre-calculation or measurement on one hand and by co-simulations on the other. Co-simulation (MBD and elastic MBD) means the calculation of the excitation (e.g. valve train) and the vibration simulation of the engine consider continuous data exchange and interaction at the same time. Furthermore, new developments allow integral simulation to be applied for specific applications and targets using extended models (e.g. example in chapter 2.5). Thus, the interaction between the crank train and timing drive can be considered in time domain.

While co-simulation has the advantage of considering the interactions between the dynamics in the load generation and structure vibration, pre-calculation is less time consuming and gives a straight forward picture of trends in a specific model part. Trend analyses are easier and faster. Fig. 9 gives an impression of the relevance of different excitation sources. The contribution of the noise excitation of the entire timing drive (including valve train and gear train) leads to significantly higher noise levels and a change in the distribution over the engine surface. Fig. 9 approves that the consideration of additional noise sources may be more important than the model refinement. Therefore, aiming for NVH simulation results of high quality, the effort for refining mesh density or joint models has to be carefully balanced against the effort of improving the boundary conditions and excitation sources.

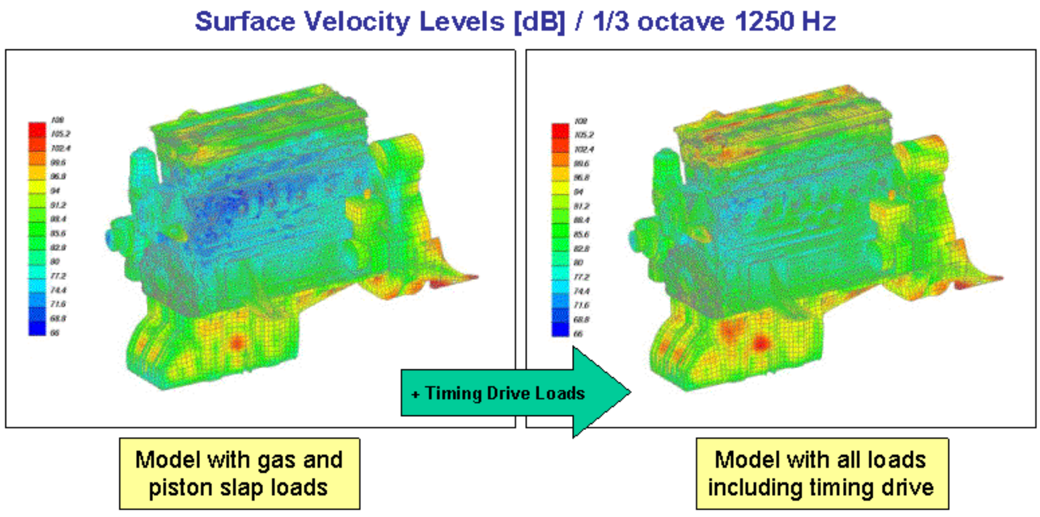


Figure 9. Integral vibration velocity levels of a truck Diesel engine w/o and with timing drive excitations.

Furthermore, speed sweeps gain in importance for the NVH prediction. While, in many cases the simulation of single speeds under full load condition of the engine provide valuable information for local design improvement, the assessment of the overall noise behaviour requires to calculate the engine run up considering speed sweeps and different loading conditions to cover all dynamic phenomenas.

2.4 Model complexity

The target of the simulation compels the detailing of the model and the loads to be considered in the calculation. NVH analysis models require all parts radiating noise, all additional parts effecting the radiation and all excitation loads having a significant effect up to 3 kHz. Auxiliaries generally influence the engine surface vibration at the position they are mounted at the engine.

Models for low frequency vibration analyses focus on engine mount vibrations and require accurate mass distribution of the entire engine model. Therefore, the model has to consider all auxiliaries but may neglect effects of piston slap and timing drive excitations. However, engine mount models have to contain data of frequency depending stiffness and damping properties, whose effect on the engine vibration behaviour is of significant importance.

For precise noise prediction, the joint models may play a decisive role also. The precision of the joint models influences the quality of the results on one hand and the CPU time effort on the other. Examples for such contacts are slider bearings and piston to liner contacts. Therefore the simulation tool has to provide different levels of joint models e.g. for the bearings. The level of joint models ranges from non-linear spring-damper elements to enhanced hydrodynamic and elasto-hydrodynamic models. An example for the contact between piston and liner will be explained in more detail in the following chapter.

2.5 Piston slap noise excitation

Piston slap induced excitation may lead to significant contribution for mechanical engine noise. Piston slap noise occurs with high torque engines and causes a broad band excitation. The noise is caused by impacts between piston and liner, some degree crank angle after firing TDC. In this position the piston performs a transverse movement in the running clearance due to the turning of the con-rod, [3, 5].

For a numerical simulation of piston slap noise excitation different possibilities have been developed. MBD pre-calculation of excitation forces at the liner, interaction of elastic piston and liner, and additional consideration of oil film in the elastic contact are three levels of detailing the excitation contact. The comparison of results from the latter two versions is shown in Fig. 10. Two observation points are plotted: acc8 in the water jacket and acc10 on the block surface. On the right of Fig. 10 the comparison of the plots show that lubricated contact (hydrodynamic + asperity pressure) and the dry contact (asperity contact pressure only) have similar results during the first contact event. After 10 degree crank angle (CRA) the results of the lubricated contact fit better to the measured accelerations, [6]. As the first impact is significant for noise excitation it can be concluded that the dry contact model is sufficient for NVH calculations.

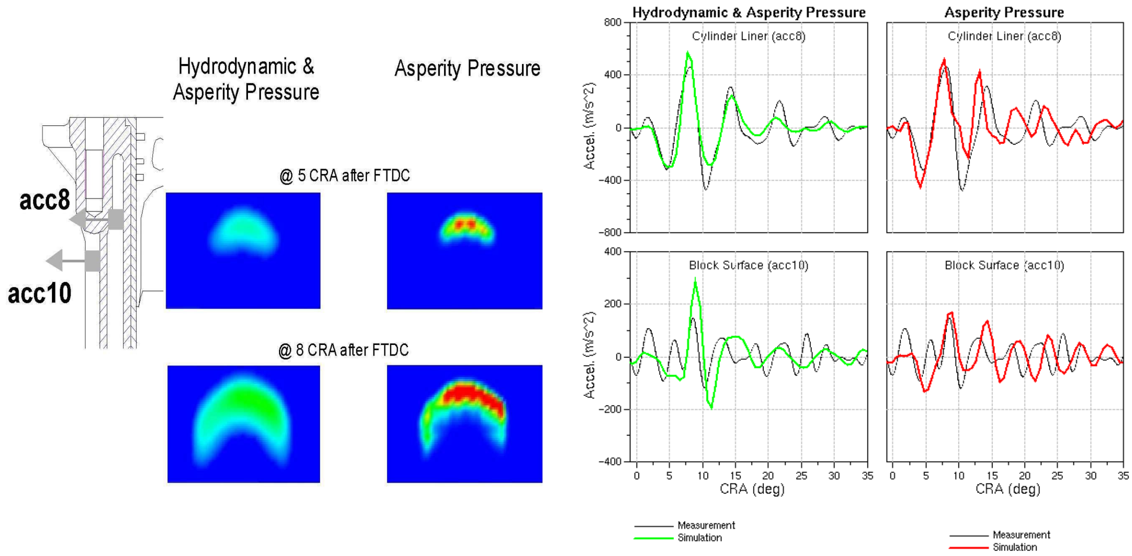


Figure 10. Comparison of pressure distribution between piston and liner 5 and 8 Degree CRA after FTDC for cases mixed lubricated and "no oil" at a 4 Cylinder truck Diesel engine, full load, 2000 rpm.

The example gives an impression about valuable extended post-processing. Although, the integral contact forces are similar in both contact models, the details of the contact events are quite different. The figure shows the contact pressure distribution on the major thrust side of the liner in two piston positions after FTDC. While the pressures at 5 degrees CRA are quite similar for the two cases, at 3 degrees CRA later the fully starved case turns into a highly localised pressure zone characterised as asperity pressure. With oil film between the surfaces the pressure is distributed over a larger area according to hydrodynamic film theory.

3 Outlook

Further reduction of simulation time is still of great importance, in order to allow for the entire speed and load range of an engine to be simulated in a reasonable project time. Furthermore, the main potential for future simulation enhancement is seen in the post-processing. Assessment of simulation results and conclusions for design modifications rely on intelligent and instructive post-processing. Existing graphical features are highly important for NVH analyses. In addition to diagrams (2D, 3D) of all motional components in any DOF of the model animations (crank shaft vibration in time domain), integral velocity levels in different views, etc, are necessary.

These features must be enhanced in the near future in order to provide the engineer with guidance and understanding of effects of input parameters. The user needs clear feedback about trends e.g. the influence of the stiffness of crankshaft and engine block on the main bearing forces and the resulting excitation of NVH. He needs feedback of the effect of FEM mesh density on the result quality, e.g. a warning about the relation between frequency range and element length. Sensitivity analysis as well as design modifications must be considered to ensure the best solution of the engine for NVH requirements.

Measurement and simulation result data need to be networked in order to allow conclusions to be drawn from one another. Such data network requires a precise documentation strategy. Furthermore, there is a demand for material data specifically for the local damping effects for sealing and joints to consider damping effects in connection areas between different parts as well as for NVH analyses of plastic parts.

Future time reduction of simulation work in the EDP will very much depend on design optimisation procedures. Although, optimisation methods exist for static aims (e.g. optimisation of mass and stiffness distribution), there is no efficient software available for complex analysis processes and models as required for the optimisation of vibration or acoustic targets. Finally, the simulation of absolute levels of air borne noise simulation is required where noise sources e.g. turbo charger, injection pump can be considered in addition to the structure borne noise.

Conclusions

The authors show the advanced state-of-the-art in simulation and optimisation of engine noise prediction with respect to improve and accelerate simulation in the engine development process (EDP). From investigations and results shown in the application examples the following conclusions can be drawn:

- Precise prediction of structure borne noise and mounting vibrations of engines and power units can be provided with respect to the different steps in the EDP.

- The models for the applications in each stage of the EDP have to be part of an integrated model approach. The component models must include different levels of detailing with respect to the development phase. Data and model levels have to be exchangeable for different applications during the development of the engine.

- The required software has to provide complete work and data flows also. This will allow the different levels of models to be exchanged easily for the different application targets to reduce overall project time and to avoid problems with interfaces and data exchange for the engineer.

- The need for reliable results to make the right design decisions requires a detailed validation of new features for the software.

Acknowledgements

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