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FACULTY OF MECHANICAL ENGINEERING

INSTITUTE OF TRANSPORT ENGINEERING

Ing. Pavel Novotný

CRANKTRAIN DYNAMICS SIMULATION – CENTRAL MODULE OF VIRTUAL ENGINE

SIMULACE DYNAMIKY HNACÍHO ÚSTROJÍ – CENTRÁLNÍ MODUL VIRTUÁLNÍHO MOTORU

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Supervisor: Prof. Ing. Václav Píštěk, DrSc.

Opponents: Prof. Ing Rudolf Kovář, CSc.

Prof. Ing. Jiří Stodola, DrSc.

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KLÍČOVÁ SLOVA

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ABSTRAKT

V současné době vzrůstá potřeba využití numerických simulací různých fyzikálních dějů. Na současné pohonné jednotky jsou kladené vysoké nároky na spolehlivost a výkon a zároveň na nízkou spotřebu, emise i na úroveň hluku. Při vývoji nových pohonných je účelné aplikovat soudobé výpočtové modely již v prvních fázích návrhu.

Virtuální motor může zahrnovat velké množství modulů. Každý modul lze popsat pomocí výpočtového modelu.Tato práce se detailně zabývá výpočtovým modelováním dynamiky klikového mechanismu. Výpočtový model klikového mechanismu tvoří hlavní modul virtuálního motoru. Jako další moduly lze zmínit především model rozvodového ústrojí, z oblasti termodynamiky pak model procesu spalování. Využití těchto modulů umožňuje zkrátit dobu potřebnou k vývoji pohonné jednotky a dosažení lepších parametrů spalovacích motorů.

Tradiční metody výpočtu klikového mechanismu využívající jednodušších metod založených na řešení pouze torzních kmitů, popřípadě v kombinaci s ohybovými kmity nadále zůstávají základem. Ovšem tyto metody umožňují řešit pouze velice omezené množství problémů a proto je nezbytné použít nové výpočtové metody, které umožní dosáhnout přesnějších výsledků a dovést pohonné jednotky blíže k mechanickým limitům. Kombinace MKP a MBS je velice efektivní cesta k simulaci dynamiky částí pohonné jednotky. Výsledky získané v MBS modelech nemusí sloužit pouze jako okrajové podmínky pro další řešení, je zde možné řešit i problematiku napjatosti a poté i životnosti součástí, ale i problémy týkající se opotřebení jako například pohyb pístu ve válci a mnoho dalších problémů. Se vzrůstající sílou moderních počítačů tyto metody postupně nahrazují tradiční výpočtové metody.

Nadále je nutné optimalizovat interakce mezi jednotlivými CAE programy a stále ověřovat výsledky těchto výpočtových modelů. Experimentální ověření výsledků výpočtových modelů se ukazuje jako nezbytná podmínka k úspěšnému návrhu pohonné jednotky. Zcela logicky ne vždy je k dispozici funkční pohonná jednotka a proto většinou postačuje i měření z podobných již existujících pohonných jednotek.

Výpočtový model klikového mechanismu nadále vyžaduje zpřesňování, týká se to především interakce mezi klikovým hřídelem a blokem motoru, v případě vznětovém řadovém šestiválcovém motoru je to model kluzného ložiska. Zde je další krok řešení hydrodynamického problému současně s řešením strukturálního problému při použití dokonalejších modelů. Rovněž i model viskózního tlumiče s vysokými hodnotami viskozity použitého média je zapotřebí zlepšit. Další významná oblast je zpřesnění konečnoprvkových modelů jednotlivých částí, týká se to především různých konstrukčních detailů jako například rádiusů, vyústění mazacích kanálků atd. Tyto modely lze využít pro detailnější řešení napjatosti a deformace součástí a následné výpočty životnosti jednotlivých součástí.

Hlavní přínos této práce je, že v ucelené formě pojednává o výpočtovém modelování v oblasti dynamiky klikového mechanismu, ať už jde o tvorbě jednotlivých modelů, redukci pružných modelů a interakci mezi nimi, tak i o problematice ověření s využitím měření na skutečném spalovacím motoru. V závěru jsou zde publikovány pouze vybrané výsledky výpočtových modelů. Všechny možné výsledky není ani možné vzhledem k rozsahu práce publikovat. Tyto vybrané výsledky mají za cíl prezentovat možnosti použitých metod řešení.

1 INTRODUCTION

The internal combustion engine development process requires CAE methods. CAE methods are most efficient at the beginning of the engine development process, when different concepts have to be compared or no hardware is yet available. This is why they are mainly used early in an idealized development process. CAE results determine the two primary ones: the concept check (identifying the best approaches) and the virtual check (transferring the chosen concept to series development).

Another requirement concerns how detailed the computer models are to be. Practice has shown that the amount of usable information gathered from the computer simulation increases less steeply than proportionally with the complexity of the model. In a project environment, usable information from a certain model complexity advances because the results become available too late – some of them later than hardware measurement results – or because lengthy response times isolate the CAE engineers from a project dialogue with the CAD area. Expressed in simpler terms, the designers cannot wait for the computing side before making their decisions, regardless of the quality of the simulations.

The next demand for CAE usage is that program platforms should, if possible, not be changed during the development process, so that CAE contributions from different sources (engine developers including pre-series and series development, suppliers, engineering partners, software firms) can be combined with little effort.

Comparison of calculations and measurements is still a decisive quality criterion in the CAE environment. Non-contact measurements make it possible to verify complex calculation models. Another of the progressive approaches in this area can be a strain gauge measurement.

All numerical simulations and measurements are performed on a Diesel six-cylinder in-line engine and Diesel V8 engine.

2 PRESENT STATE OF THE RESEARCH

The internal combustion engines are thermo-mechanical systems and this demands a special solution approach. The cranktrain dynamic model is the main part of a virtual engine. A major problem is the interaction between a crankshaft and an engine block.

Failures and mainly fatigue fractures occurred with an engine power increase and material savings in former times. These fractures were not caused only by forces from combustion pressure processes and inertial forces or incorrect construction but were found that the main portion of these fractures is caused by periodical torsional vibrations of the crankshaft. Three types of crankshaft vibrations exist in agreement with traditional approach: bending, torsional and axial vibrations.

The bending vibrations are caused by periodical forces that operate perpendicular on a crankshaft axis. These forces are radial and tangential forces on the crankshaft and imbalanced eccentrical forces of the cranktrain. Natural frequencies of bending modes are given a free length of the crankshaft between the bearings. Free lengths of crankshaft between two bearings are very small and natural frequencies of bending modes are high and there are no resonances at an engine speed range. Stiffness of the bearings and the engine block has a large influence on bending natural frequencies.

The torsional vibrations of the crankshafts are much more dangerous than bending vibrations. Forced torsional vibrations of the crankshaft are caused by time depended torques. Torsional vibrations reach to high values in resonances when a frequency of forced vibration is equal to natural frequency. The resonances and relevant critical engine speeds cause an increased noise and engine vibration.

Many computational methods have been developed for a cranktrain modeling in former times. These computational models often responded to a computer technique state. Complexity of computational models increases with computer technique development.

The main focus has been engaged with a crankshaft. First, computation models used the crankshaft as a rigid body. The crankshaft is described using a mass, a center of the mass and an inertia tensor. This simpler model is impracticable for many applications.

The crankshaft model using a discrete member for every crank throw is the next level. The crank throw mass with a connecting rod equivalent mass and a piston equivalent mass is centered into a disc with constant moment of inertia. The disc has identical kinetic energy as the relevant part of the crankshaft. An axis of discs is identical as a cranktrain axis. A torsional stiffness for every crank throw can be found from empirical relations. **Fig. 2.1** presents a discrete torsional model of the crankshaft. The simple discrete torsional model gives exact results of torsional vibrations of crankshafts, but bending and axial vibrations of crankshafts cannot be solved.

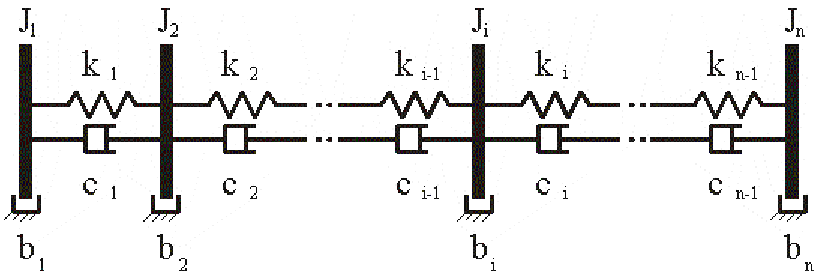

Fig. 2.1 Discrete torsional model of a crankshaft

A beam model is the next step. A parametric beam model represents the torsional and bending stiffness of the crankshaft. **Fig. 2.1** shows the beam model.

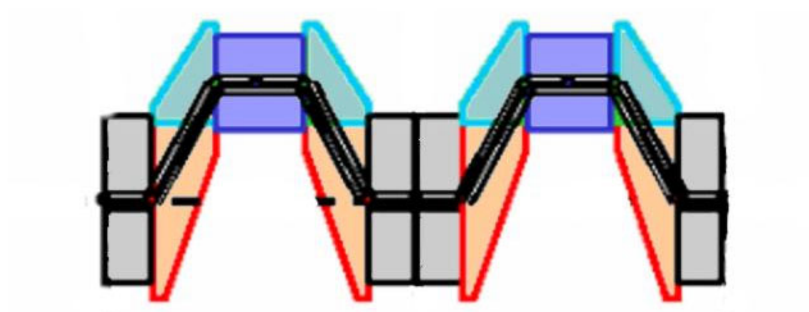

Fig. 2.2 Beam model of a crankshaft

Present computational models of the cranktrain are a very complex one and include many submodels. **Fig. 2.3** shows an example of an adaptive crank train model. All connecting rods and whole engine block around the crankshaft are not shown in **Fig. 2.3**.

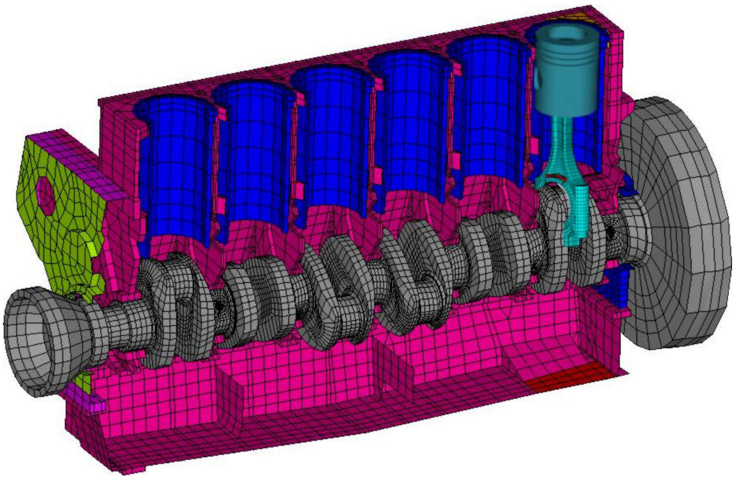

Fig. 2.3 Adaptive cranktrain model with flexible parts

The engine development process starts with a kinematic model made up of rigid bodies, specified with weight and inertia, approximations of which can be derived from the first design sketches for a new engine. The model is excited with its inertia forces under uniform rotation and with a provisional cylinder pressure signal. This simulation supplies important specification criteria for the concept phase (approximate bearing forces, free engine forces and torques, loads on connecting rods and individual cranks).

The crankshaft, flywheel and torsional vibration damper in the dynamic cranktrain model are subsequently replaced with elastic bodies, represented in the FEM program by way of their static deformation and vibration modes and added to the MBS program in this form. The cylinder pressure load can already be defined more precisely now, for example with indicated pressures of the planned combustion process from a single-cylinder engine. The benefit from such a method depends on the specific problem (e.g. strength, acoustics). The same applies to the variety of model stages for the oil film in the crankshaft bearings. Examinations on a freed individual bearing allow a different choice of detail than on a complete-engine dynamic model.

3 FE MODELS OF MAIN PARTS

In the case of dynamic structural calculations, the Finite Element Method (FEM) is 'state of the art'. The calculation of the structural transfer behaviour of single components is efficiently possible in the frequency domain, giving consideration to a large number of degrees of freedom. ANSYS software is used for FE model creations and calculations.

First of all CAD models of Diesel in-line 6-cylinder engine and Diesel V8 engine are created. CAD models give basic geometric information about engine parts and are used for a creation of FE models (e.g. crankshafts or engine blocks) or MBS rigid parts (e.g. pistons or piston pins).

A material model of all flexible parts is linear and isotropic. Young's modulus, Poisson's ratio and density describe each flexible part.

The first analysed engine is the Diesel 6-cylinder in-line 6.2-litres engine. **Fig. 3.1** shows FE models of main parts created on the basis of CAD models. The second one is the Diesel V8 12.6-litres engine. CAD models and FE models of V8 engine are presented in **Fig. 3.2**.

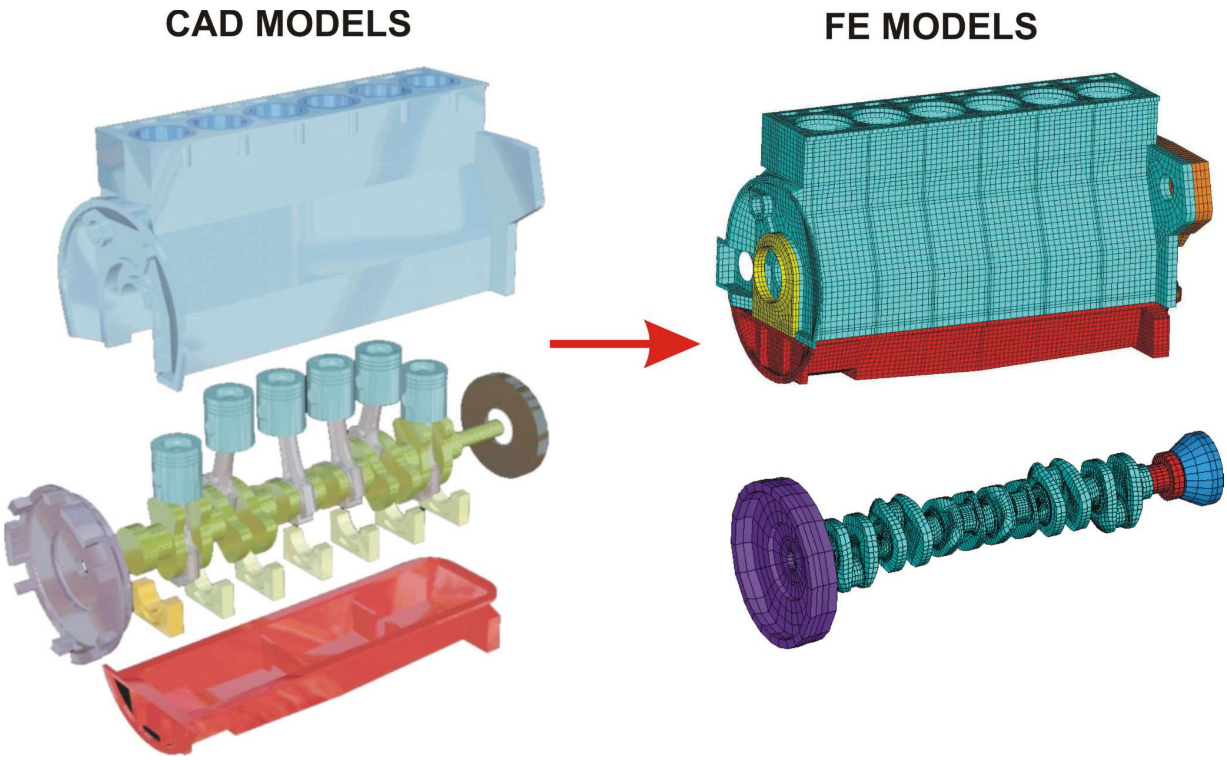

Fig. 3.1 Diesel in-line 6-cylinder engine CAD and FE models

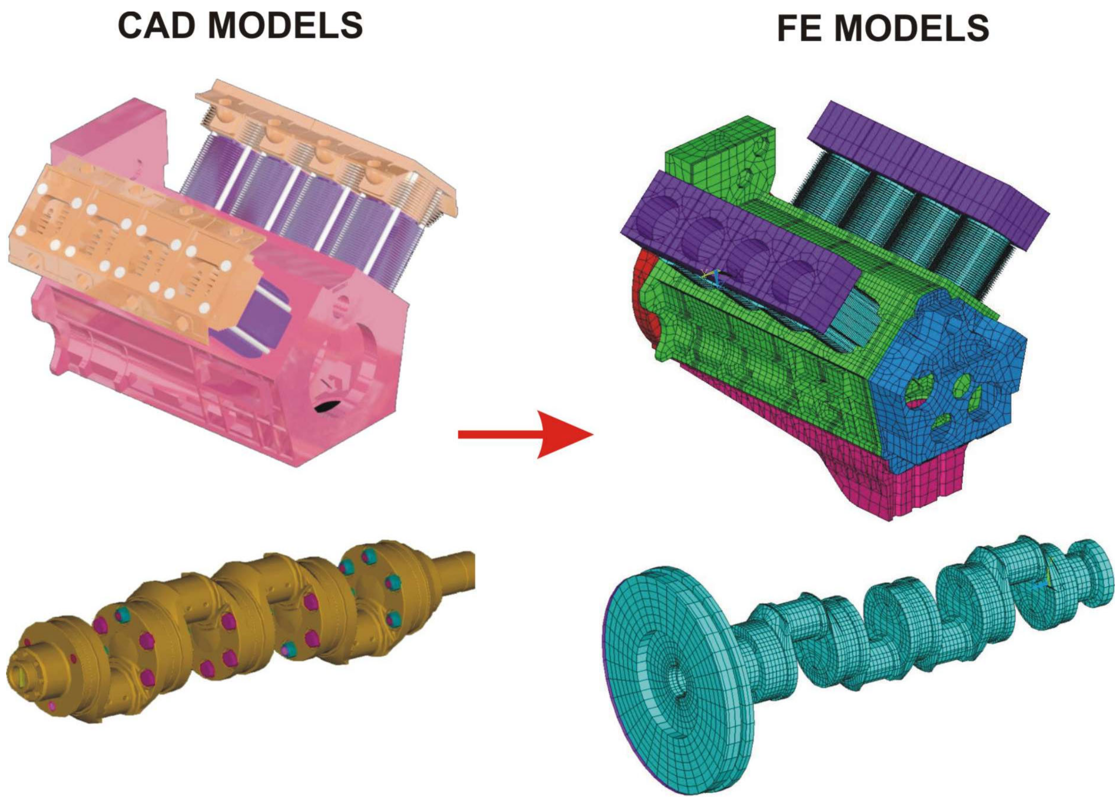

Fig. 3.2 Diesel V8 engine CAD and FE models

All FE models have uniform hexahedral element mesh and don't include small geometric details. The exception is the FE model of the in-line 6-cylinder crankshaft; there are radiuses on main journals and crank pins and lubricating canals on main journals. The uniform element size is suitable for dynamic solutions in time domain. The in-line 6-cylinder crankshaft includes small geometric details and is suitable for a stress-strain analysis.

4 REDUCTION OF FE MODELS

4.1 REDUCTION THEORY SUMMARY

Modal analyses can be performed in ANSYS and are not time-consuming, but for solution in time domain, these models are very large and require reduction. The discretization of a flexible component into a finite element model represents the infinite number of DOF with a finite, but very large number of finite DOF. The linear deformations of the nodes of this finite element mode, *u*, can be approximated as a linear combination of a smaller number of shape vectors, see equation (4.1).

$$u = \sum_{i=1}^{M} \phi_i q_i \quad , \tag{4.1}$$

where M number of mode shapes

ϕ_i mode shape

q_i modal co-ordinates

For reduction of FE models, the Craig-Bampton method is used. The Craig-Bampton method separates the system DOF into boundary DOF, u_B , and interior DOF, u_I. Two sets of mode shapes are defined as follows:

Constrain modes: these modes are static shapes obtained by giving each boundary DOF a unit displacement while holding all other boundary DOF fixed.

Fixed-boundary normal modes: these modes are obtained by fixing the boundary DOF and computing an eigenvalue solution.

The relationship between the physical DOF and Craig-Bampton modes and their modal co-ordinates is illustrated by the following equation (4.2):

$$u = \begin{Bmatrix} u_B \\ u_I \end{Bmatrix} = \begin{vmatrix} I & 0 \\ \phi_{IC} & \phi_{IN} \end{vmatrix} \begin{Bmatrix} q_C \\ q_N \end{Bmatrix} \quad , \tag{4.2}$$

where

u_B column vector of boundary DOF,

u_I column vector of interior DOF,

I , 0 are identity and zero matrices,

ϕ_{IC} physical displacements of interior DOF in the constrain modes,

ϕ_{IN} physical displacements of interior DOF in normal modes,

q_C column vector of the modal co-ordinates of the constrain modes,

q_N column vector of the modal co-ordinates of the fixed-boundary normal modes.

The generalized stiffness and mass matrices corresponding to the Craig-Bampton modal basis are obtained via a modal transformation. The stiffness transformation is

$$\hat{K} = \Phi^T K \Phi = \begin{vmatrix} I & 0 \\ \phi_{IC} & \phi_{IN} \end{vmatrix}^T \begin{vmatrix} K_{BB} & K_{BI} \\ K_{IB} & K_{II} \end{vmatrix} \begin{vmatrix} I & 0 \\ \phi_{IC} & \phi_{IN} \end{vmatrix} = \begin{vmatrix} \hat{K}_{CC} & 0 \\ 0 & \hat{K}_{NN} \end{vmatrix} \quad , \qquad (4.3)$$

and mass transformation is

$$\hat{M} = \Phi^T M \Phi = \begin{vmatrix} I & 0 \\ \phi_{IC} & \phi_{IN} \end{vmatrix}^T \begin{vmatrix} M_{BB} & M_{BI} \\ M_{IB} & M_{II} \end{vmatrix} \begin{vmatrix} I & 0 \\ \phi_{IC} & \phi_{IN} \end{vmatrix} = \begin{vmatrix} \hat{M}_{CC} & \hat{M}_{CN} \\ \hat{M}_{NC} & \hat{M}_{NN} \end{vmatrix} \quad , \qquad (4.4)$$

where the subscripts *I, B, N* and *C* denote interior DOF, boundary DOF, normal mode and constrain mode.

Equations (4.3) and (4.4) have a few properties:

- submatrices $\hat{M}_{NM}$ a $\hat{K}_{NM}$ are diagonal because they are associated with eigenvectors
- matrix $\hat{K}$ is block diagonal. There is no stiffness coupling between the constrain modes and fixed-boundary normal modes
- matrix $\hat{M}$ is not block diagonal because there is an inertia coupling between the constrain modes and fixed-boundary normal modes.

The Craig-Bampton method is a powerful method for tailoring the modal basis to capture both the desired attachment effects and the desired level of dynamic content. However, the raw Craig-Bampton modal basis has certain deficiencies that make it unsuitable for use in a dynamic system simulation. These are:

- Embedded in the Craig-Bampton constrain modes are 6 rigid body DOF which must be eliminated before the ADAMS analysis because ADAMS provides its own large-motion rigid body DOF.
- The Craig-Bampton constrain modes are the result of a static condensation. These modes do not advertise the dynamic frequency content that they must contribute to the flexible body.
- The Craig-Bampton constrain modes cannot be disabled because it would not be possible to apply constrains to the system.

The problems mentioned above can be easily resolved by applying a simple mathematical operation to the Craig-Bampton modes. The Craig-Bampton constrain modes are not an orthogonal set of modes, as evidenced by the fact that their generalized mass and stiffness matrices $\hat{M}$ and $\hat{K}$, encountered in equations (4.3) and (4.4), are not diagonal.

By solving an eigenvalue problem

$$\hat{K}q = \lambda \hat{M} q \quad , \qquad (4.5)$$

we obtain eigenvectors that we arrange in a transformation matrix *N* which transforms the Craig-Bampton modal basis to an equivalent, orthogonal basis with modal coordinates q*.

$$Nq^* = q \quad . \qquad (4.6)$$

The effect on the superposition formula (4.1) is

$$u = \sum_{i=1}^{M} \phi_i q_i = \sum_{i=1}^{M} \phi_i N q^* = \sum_{i=1}^{M} \phi_i^* q_i^* \quad , \qquad (4.7)$$

where

ϕ_i^* orthogonalized Craig-Bampton modes.

The orthogonalized Craig-Bampton modes are not eigenvectors of the original system. They are eigenvectors of the Craig-Bampton representation of the system and as such they have natural frequencies associated with them.

4.2 PRINCIPLE OF FE MODEL REDUCTION IN ANSYS

When FE models are built in ANSYS, these points have to be remembered:

- The interface is designed to support most element types that have displacement degrees of freedom. Exceptions are axisymmetric elements (for example, PLANE25) and explicit dynamic elements (for example, SOLID164).
- Only linear behavior is allowed in the model. If nonlinear elements are specified, they are treated as linear. For example, if nonlinear springs are included (like COMBIN39), the stiffness is calculated based on the initial status and never change.
- Material properties can be linear, isotropic or orthotropic, constant or temperature-dependent. Young's modulus (EX, or stiffness in some form) and density (DENS, or mass in some form) must be both defined for the analysis. Nonlinear properties are ignored.
- Damping is ignored when the interface computes the modal neutral file (MNF). Damping of the flexible component can be added later in the ADAMS program.
- The ADAMS program requires a lumped mass approach (LUMPM,ON). This requirement results in the following special considerations.
    - For most structures that have a reasonably fine mesh, this approximation is acceptable. If a model has a coarse mesh, the inertia properties may have errors. To determine what the effect will be, it is recommended to start a modal analysis with and without LUMPM,ON and compare the frequencies.
    - When using SHELL63, set KEYOPT(3) = 2 to activate a more realistic in-plane rotational stiffness. If the elements are warped, use SHELL181 with KEYOPT(3) = 2 instead.
    - When using two dimensional elements, the corresponding ADAMS model must lie in the X-Y-plane. Remember that ADAMS models are always three dimensional. The 2-D flexible component transferred will not have any component in the Z-direction.
    - Nodes of a plane element only have two degrees of freedom: translations in the X- and Y-direction. Thus, no moment loads (forces, joints) can be applied in the ADAMS analysis. Likewise, nodes of a solid element only have translational degrees of freedom.

- Constraints cannot be applied (D command) to the model. Also, make sure that no master degrees of freedom (M or TOTAL commands) were defined in an earlier analysis.

A model that will be used in an ADAMS simulation is an important consideration how to represent interface points within the structure. An interface point is a node that will have an applied joint or force in the ADAMS program. In ADAMS, the forces can only be applied to interface points.

Special attention must be realized:

- An interface point must have six degrees of freedom (except for 2-D elements).
- Force (applied directly or via a joint) should be applied to the structure by distributing it over an area rather than applying it at a single node.
- If there is no node in the structure where a force or a joint can be applied in ADAMS (for example, a pin center), a geometric location for that point has to be created.

The following guidelines have to be used to determine the best way to model the interface points for the structure:

- To ensure that all loads will be projected on the deformation modes in the ADAMS simulation, all nodes where a joint or a force will be applied have to be defined as interface points.
- Interface points in ANSYS have to always have 6 degrees of freedom, except for 2-D elements. If a model consists of solid elements, constraint equations or a spider web of beam elements have to be used to ensure that the interface node has 6 degrees of freedom.
- A good practice for modeling interface points is to reinforce the area using beam elements or constraint equations. Using one of these techniques will distribute the force over an area rather than applying it to a single node, which would be unrealistic.
- If a spider web of beam elements is used, a high stiffness and a small mass should be used for the beams. Otherwise, the stiffness and mass of your model will be altered, which could result in eigenmodes and frequencies that do not represent the original model.
- Constraint equation commands such as CE and CERIG may be used to attach the interface node (for example, CERIG,MASTE,SLAVE,UXYZ, where MASTE is the interface node). Avoid the RBE3 command since problems can occur with the master degrees of freedom.
- Interface points that lie next to each other and are connected by constraint equations or short beams should not be defined. This type of connection would require too many eigenmodes and result in a model that is not well conditioned.
- The FE models of crankshafts and engine blocks are reduced to the number of boundary nodes selected by the user. Their number is selected according to the application of constrains of the model. Loads are applied on these nodes only.
- The reduction of FE models from ANSYS into ADAMS is executed from ANSYS. The parameters which influence the accuracy of the reduction are the number and location of the boundary nodes and the number of first modes considered during

the reduction. These parameters can be validated, for example, by computing the FE model with constrains in ANSYS and separately in ADAMS and by subsequent comparison of the results.

4.3 REDUCTION OF ENGINE PARTS

The parameters which influence the accuracy of the reduction are the number and location of the boundary nodes and the number of first modes considered during the reduction. These parameters can be validated, for example, by computing a modal analysis of the FE model with constrains in ANSYS and separately in ADAMS and by subsequent comparison of the results.

It is useful if the modal analysis of the flexible part is computed before the reduction. Results from the modal analysis are natural frequencies and mode shapes of the flexible part. These results can be used for suitable reduction parameters. **Fig. 4.1** presents natural frequencies and mode shapes of a Diesel in-line 6-cylinder engine crankshaft.

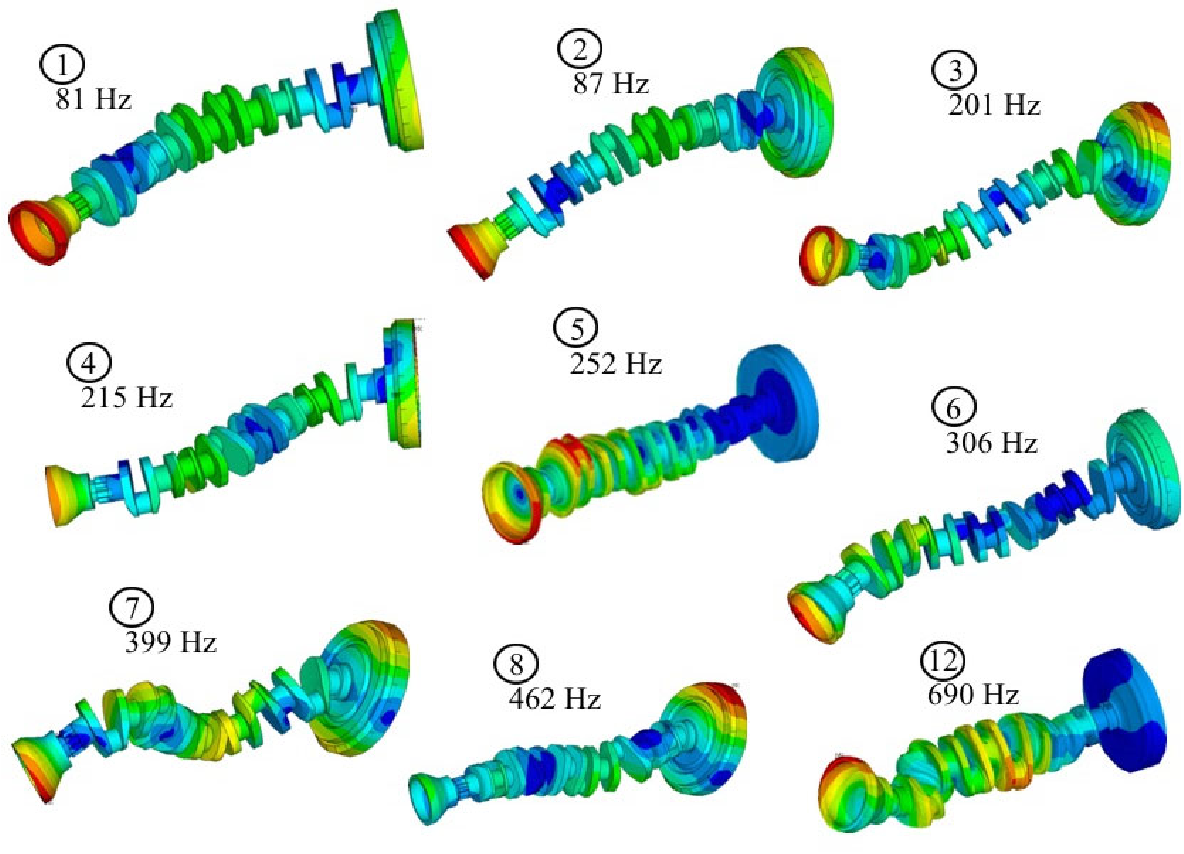

Fig. 4.1 Natural modes of a Diesel in-line 6-cylinder engine crankshaft

The modal analysis results provide a very important step before dynamic solution in time domain. The fundamental first torsional mode shape is at 252 Hz and second torsional mode shape is at 690 Hz. Mode shapes up to 1000 Hz are considered for the reduction.

5 INTERACTION BETWEEN ENGINE PARTS

5.1 PLAIN JOURNAL BEARINGS

A general advantage of plain journal bearings are their simplicity and simple production, mainly composite bearings. A disadvantage is the precious lubricating of the bearings.

A slide bearing is described as a sleeve around a pin with a lubricating fluid. The lubricant is supplied with a suitable slot. Tangential and radial motion in combination with a wedged gap generate a pressure in oil film in the slide bearing. Bearing loading is a periodical and a pin center comes through a bearing trajectory.

Slide bearings are used in the case of a 6-cylinder in-line Diesel engine. Plain journal bearings have great affect on dynamic behavior of the crankshaft and are a very important part of the virtual engine.

5.1.1 Theoretical background

Assuming an incompressible fluid (Newton's fluid) with a constant viscosity in a laminar lubricating flow, an equation describing the force equilibrium in a differential fluid element can be established. This equation is also known as the Navier-Stokes equation and can be written as:

$$\rho w(\nabla w) = -\nabla p + \eta \nabla^2 w + \rho g \quad , \tag{5.1}$$

with $\rho w(\nabla w)$ denoting specific inertia forces, ∇p specific pressure forces, $\eta \nabla^2 w$ specific viscosity forces, ρg specific mass forces and w the velocity vector of the fluid element. Making several reasonable assumptions and applying appropriate transformations and boundary conditions, equation (5.1) can be rewritten to the form of:

$$\frac{\partial^2 \Pi}{\partial \varphi^2} + \left(\frac{D}{B}\right)^2 * \frac{\partial^2 \Pi}{\partial Z^2} + a(\varphi, Z) * \Pi = b(\varphi, Z) \quad , \tag{5.2}$$

which is also known as Reynold's differential equation. $\prod$ denotes a transformed pressure distribution, φ and ***Z*** are the coordinates in circumference and width direction, ***D/B*** a form factor and ***a*** and ***b*** are position-dependent coefficients. Equation (5.2), which is the form of a partial elliptic differential equation, cannot be solved analytically. However, it can solve equation (5.2) numerically, using an iterative algorithm, as it is done in the EHD subroutine.

The implementation of this oil film model into ADAMS is done by a GFORCE User-Written-Subroutine (GFOSUB). The parameters that are needed by the subroutine are: rotational speed of the crank shaft, position and velocity (vector) of journal relative to the bearing shell at each bearing side to consider an inclined position of the crank shaft inside the bearing shell. To keep track of the actual bearing state two initially coincident markers (shell and journal) are placed at each bearing side.

5.1.2 Elasto-Hydrodynamic Oil Film Model in ADAMS

The numerical solution of the equation of the motion in time domain with integration time step corresponding crank angular displacement 0.01° requires effective calculation of the hydrodynamic reaction of a radial slide bearing. The procedure derives from the Reynolds hydrodynamic equation. The solution of hydrodynamic radial and axial slide bearings is implemented in ADAMS, where loads are obtained from a precalculated oil film reaction force database.

When hydrodynamic bearings are modeled, it can choose to include or exclude misalignment effects by using the following methods:

- For the two-dimensional method neglecting misalignment - an empirical analytical equation is used. This approach, which is similar to the impedance method, is the most efficient way to model hydrodynamic bearings.
- For the three-dimensional method accounting for misalignment - the Reynolds equation must be solved explicitly. To keep the simulation effort in a reasonable range, you must decouple the hydrodynamic solution from the dynamic solution of ADAMS/Solver. Therefore, the Reynolds equation is solved for the slide bearing and the results are stored in hydrodynamic databases representing the dimensionless bearing reactions (forces and force-attachment coordinates) to dimensionless states (eccentricity and misalignment value). During the dynamic solution, ADAMS/Solver subroutines do the database access and the necessary analytical steps (coordinate transformations, and so on).

The component outputs give the following results:

- Forces - both radial forces, both moments due to misalignment, friction torque and oil flow amount. If an axial bearing is attached, the axial force is an output too.
- Velocities - relative eccentrical velocities between the shell and pin part in both radial directions, effective eccentrical velocity, relative rotational velocity, rotational gap velocity, and hydrodynamic effective rotational velocity.
- Displacements - eccentricity in radial directions, dimensionless eccentricity, minimal rest gap, angular position of the minimal gap, and misalignment angle.

Fig. 5.1 presents slide bearing solving process in ADAMS/Solver.

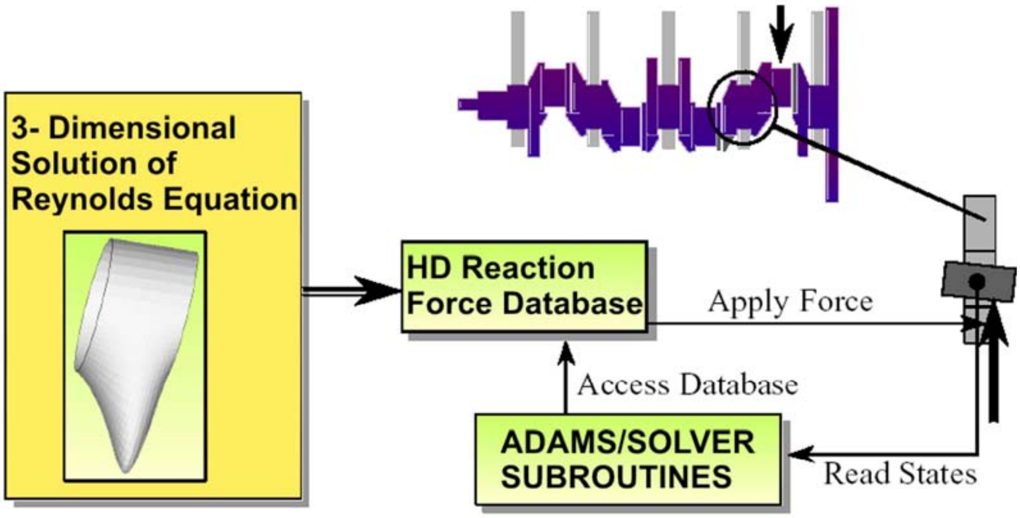

Fig. 5.1 Hydrodynamic bearing model of a slide bearing in ADAMS

5.2 CYLINDRICAL ROLLER BEARING

A low requirement of precious lubricating of roller bearings is a main advantage for their applications on combustion engines. An oil temperature is lower due to low friction and start of the cold engine is smoother. However, a noise level produced by the roller bearings is relatively high.

In the case of a Diesel V8 engine, cylindrical roller bearings are used for main journal bearings. The Diesel V8 includes two types of main journal cylindrical roller bearings because a flywheel is stabilized by a smaller roller bearing.

The roller bearing represents a complicated contact problem with many issues, but for successful simulation it can be solved simpler in a computational model of the cranktrain. First, any hydrodynamic lubricating is neglected and a static model is used for a submodel of the bearing. A complex model is created in FEM and FEM results are inserted in a simpler model in MBS. Linear radial stiffness and linear bending stiffness are computed in FEM and then are used in a MBS bearing model. The FE and MBS bearing models are presented in **Fig. 5.2**.

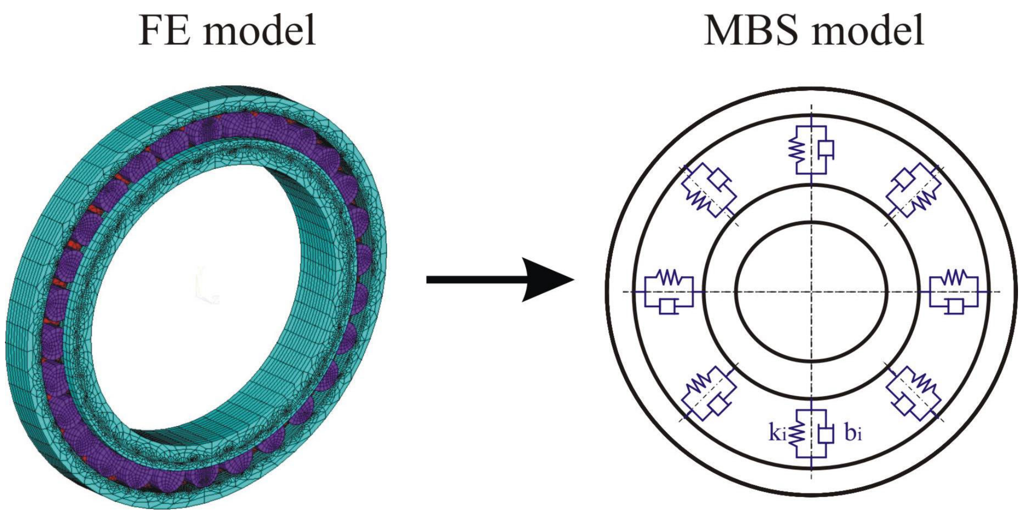

Fig. 5.2 FE and MBS models of Diesel V8 roller bearings

The FEM computed radial stiffness can be effectively verified by theoretical (Williams [36]) or empirical (Stolarski [33]) relations.

5.3 PISTON AND ENGINE CYLINDER INTERACTION

The interaction of the piston in the cylinder is another complicated problem. A piston is moved in the cylinder and is influenced by high accelerations, temperatures and pressures. A contact area of the piston is not stabile and is unevenly covered by an oil film. Piston rings transfer a specific portion of the piston contact.

A more complex solution can be found in Khonsary [10] or Offner [20]. A piston and a cylinder interaction are solved more simply in a computational cranktrain model. The interaction is solved by using beam elements in the FE model. These beam elements have approximate piston stiffness properties and are connected in two layers in the center of the piston. The piston is solved as a rigid body and is described by a mass and an inertia tensor.

6 SOLUTION

The solution of the complex model in time domain runs in ADAMS. Major parts, for example a crankshaft and an engine block (see **Fig 6.1**), are solved as flexible parts. The other parts are solved as rigid. Gas forces on pistons and a torsional damper are added in ADAMS as well. Some temperature elements for dependency on temperature are defined. The first temperature element defines the dependency of the oil temperature in bearings on the engine speed and is referenced by a viscosity element that describes the dependency of oil viscosity on oil temperature. The oil viscosity is then necessary to describe the behaviour of hydrodynamic bearings. The second temperature element defines the dependency of the torsional vibration damper temperature on engine speed, which is referenced again by the torsional damper.

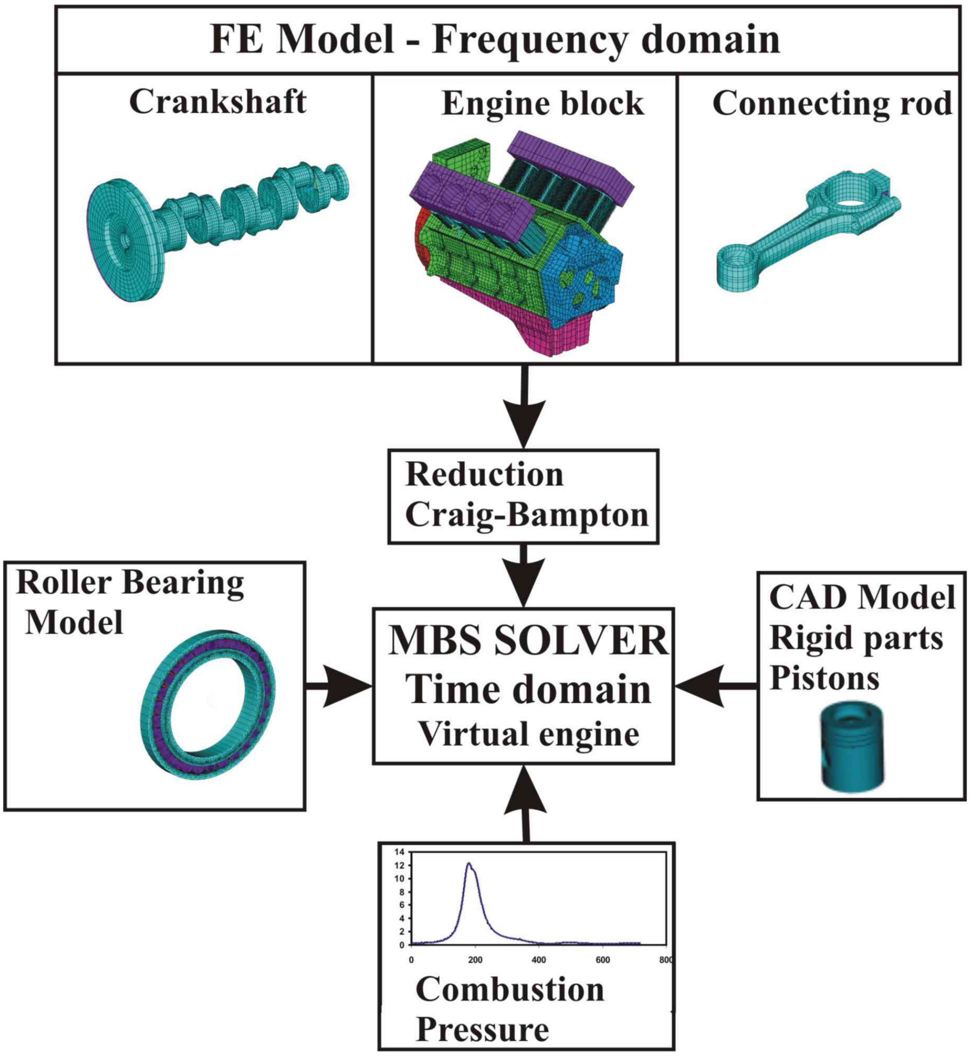

Fig. 6.1 Assemble of a virtual engine with flexible and rigid parts

The solution is very complicated and requires a great deal of computer time. The numerical solution of the equation of the motion in time domain runs with integration time step corresponding to a crank angular displacement of approximately 0.01°.

7 MAIN RESULTS

7.1 DIESEL IN-LINE 6 CYLINDER ENGINE

Computational results are compared with experimental Diesel in-line 6-cylinder engine measurements. The experimental engine is connected through a connecting shaft to the dynamometer Shenck W400 and many parameters can be measured and then can be compared with the calculations.

Only some results are presented in this thesis. The presented results are mainly torsional vibrations and an axial displacement of a crankshaft pulley in the case of a Diesel in-line 6-cylinder engine.

Opening analysis of cranktrains can be a modal analysis. Natural frequencies and mode shapes of the cranktrain can be found. **Tab. 7.1** presents natural frequencies and mode shapes of the in-line 6-cylinder engine cranktrain.

Tab. 7.1 Natural frequencies of in-line 6-cylinder engine cranktrain.

Deisel in-line 6-cylinder engine			
Natural freq. number	Natur. frequency	Natural frequency	mode shape description
[-]	[Hz]	[Hz]	
	Computation	Measurement	
1	208	-	bending
2	238	237	torsional
3	267	-	bending
4	289	-	bending
5	300	292	axial

A first torsional natural frequency has the greatest effect on torsional forced damped vibrations. A second torsional natural frequency occurs at *650 Hz* and has only a smaller effect and is included in the solution of a forced damped solution. Critical engine speeds can be determined from the first torsional natural frequency and dominant harmonic orders of combustion pressure (see **Tab. 7.2**).

Tab. 7.2 Critical Diesel in-line 6-cylinder engine speeds

Diesel in-line 6-cylinder engine		
Harmonic order	Computation	Measurement
κ	n_{kr}	n_{krE}
[–]	[min^{-1}]	[min^{-1}]
3	4760	4740
6	2380	2370
6,5	2197	2188
7,5	1904	1896
9	1587	1580
12	1190	1185

A flexible crankshaft is a major part of an engine model. Some results, for example torsional vibrations or axial vibrations of a crankshaft, are not too sensitive in using a flexible behavior of a engine block. However, the flexible block has a decisive effect on bending vibrations of the crankshaft. The torsional vibrations of the pulley of the Diesel in-line engine without a torsional vibration damper for some critical engine speeds are presented in **Fig. 7.1**.

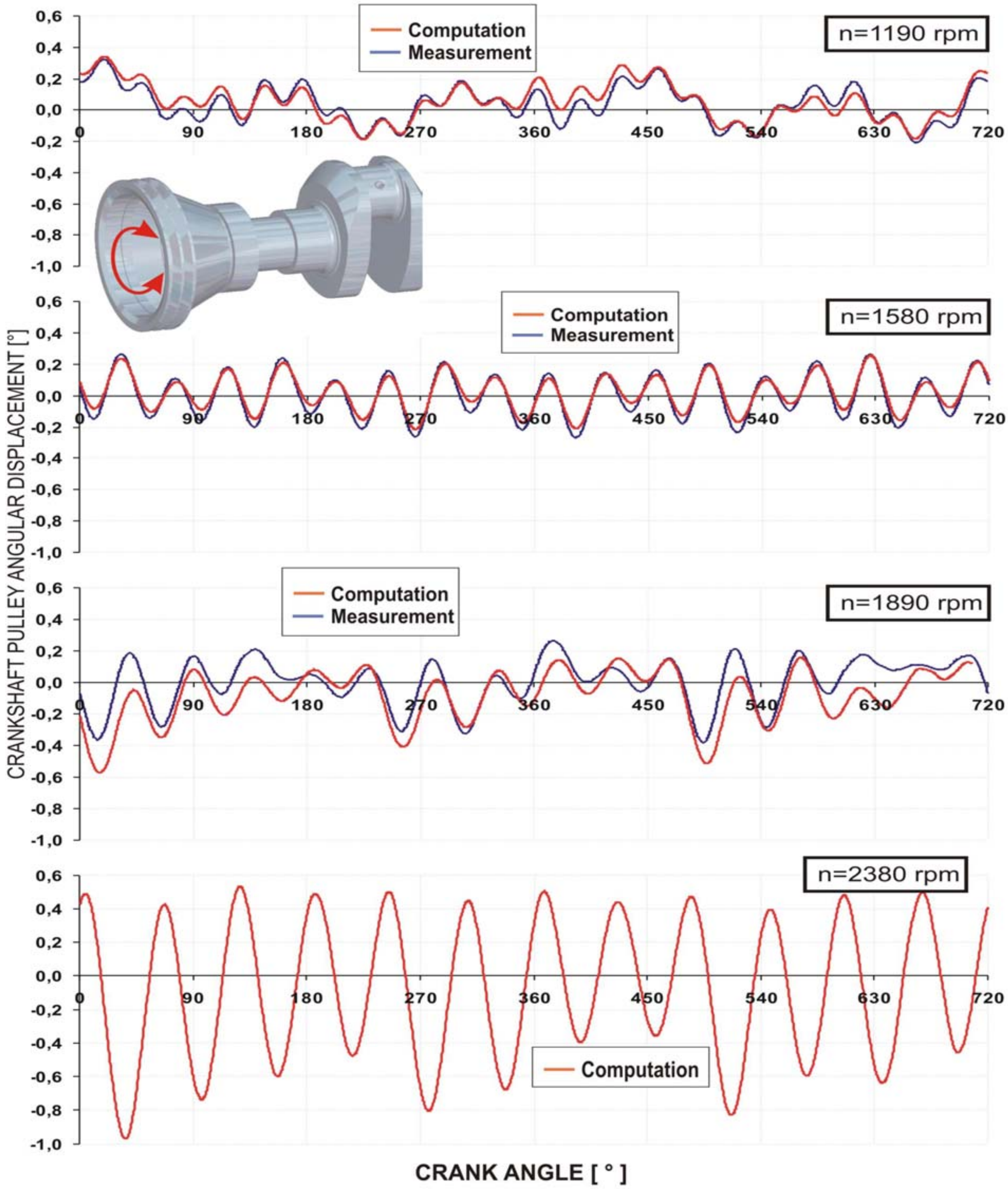

Fig. 7.1 The torsional vibrations of the pulley of the Diesel in-line engine without a damper for some critical engine speeds

The computed and measured torsional vibration order analysis of the pulley of the Diesel in-line engine without a damper for critical engine speeds is presented in **Fig. 7.2** and computed torsional vibration order analysis of the pulley of the Diesel in-line engine without a damper for full range of engine speeds is presented in **Fig. 7.3**.

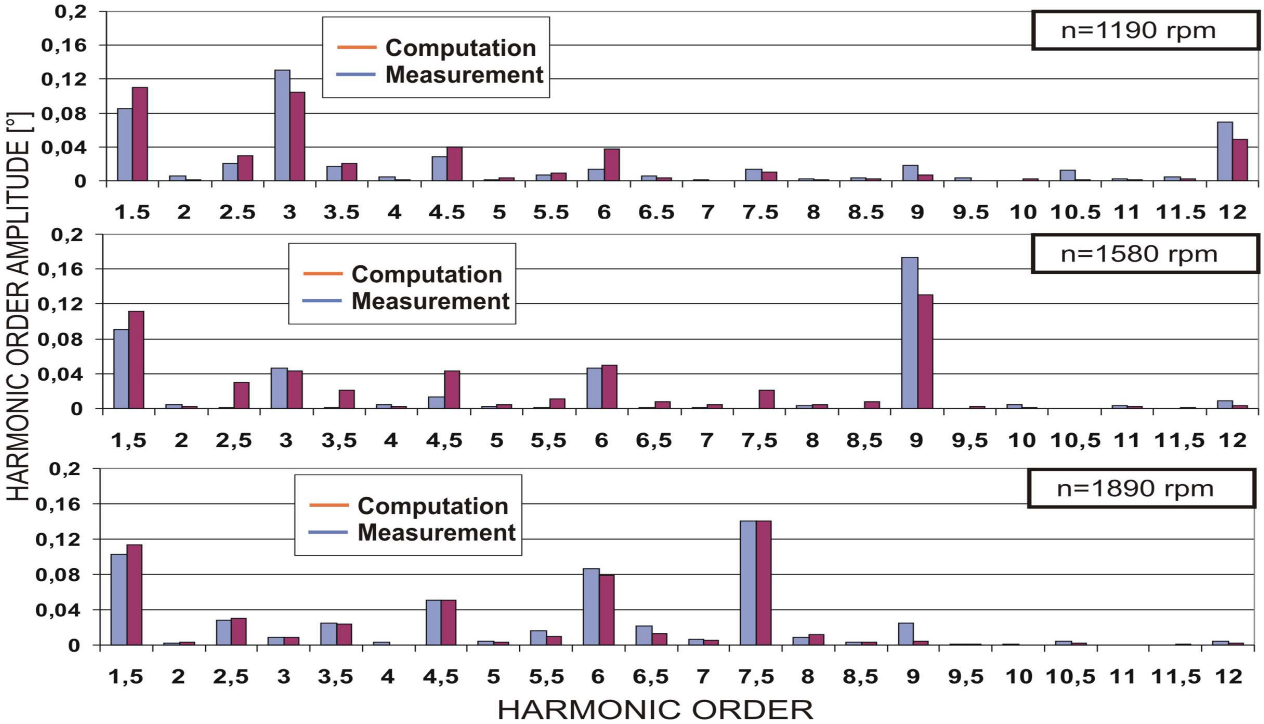

Fig. 7.2 Computed and measured torsional vibration order analysis of an in-line 6-cylinder engine crankshaft pulley without a damper for critical engine speeds

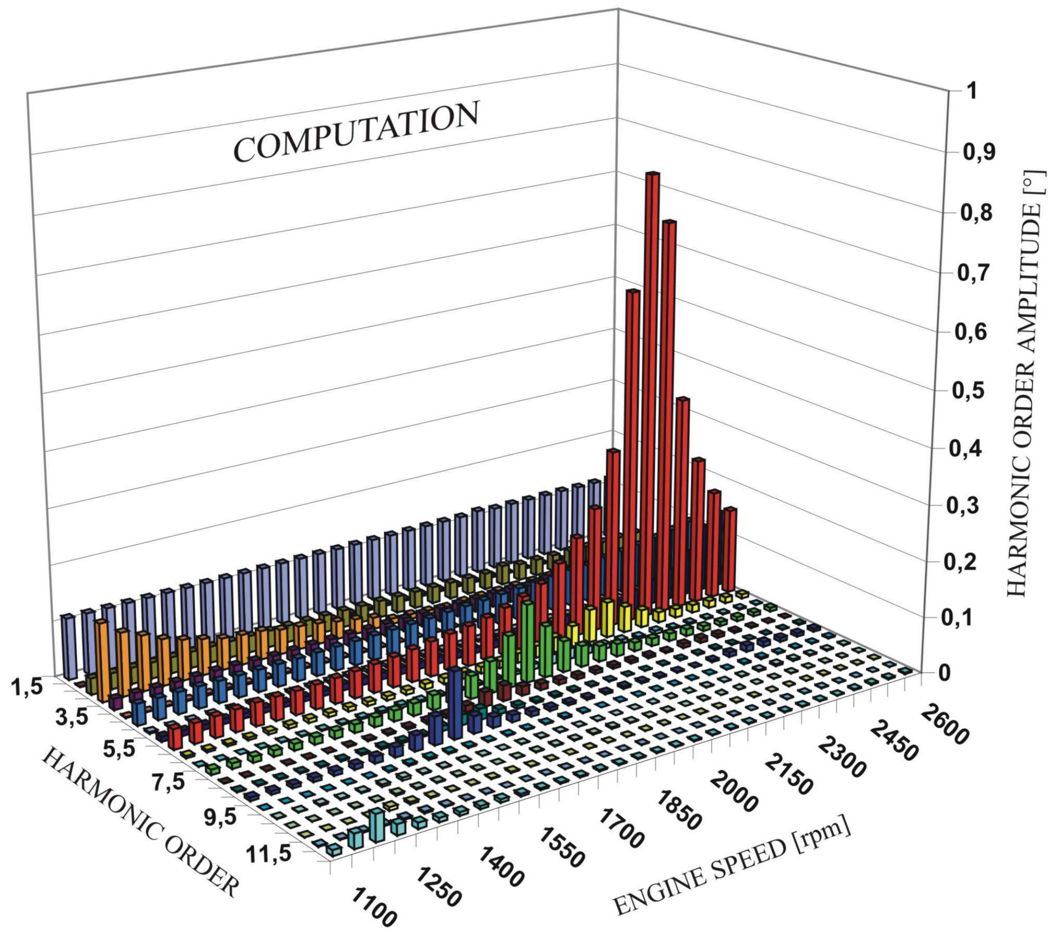

Fig. 7.3 Computed torsional vibration order analysis of an in-line 6-cylinder engine crankshaft pulley without a damper

An axial displacement of the crankshaft pulley is another important section of the results and the computation results and measurements are focused on the pulley axial displacement in detail too.

Several components of the axial displacement of the crankshaft pulley exist. An axial bearing displacement is the first component. An oil film in the bearing is created at the start of the engine and this causes the axial displacement increase. After this, the axial bearing moves periodically within the range of an axial clearance. This component presents a smaller value of total pulley axial displacement in the case of the diesel in-line 6-cylinder engine. An axial vibration of the crankshaft is the second component of total pulley axial displacement. The crankshaft axial displacement increases in engine speeds where some axial resonances exist and presents a major component of total pulley axial displacement. The displacement of an entire engine is the last relevant component of the pulley axial displacement and presents only a very small value. This component depends on the stiffness and damping of engine mounts.

Fig. 7.4 presents computed axial vibration order analysis of an in-line 6-cylinder engine crankshaft pulley without a torsional vibration damper. The vibration order analysis includes all components of the axial pulley displacement.

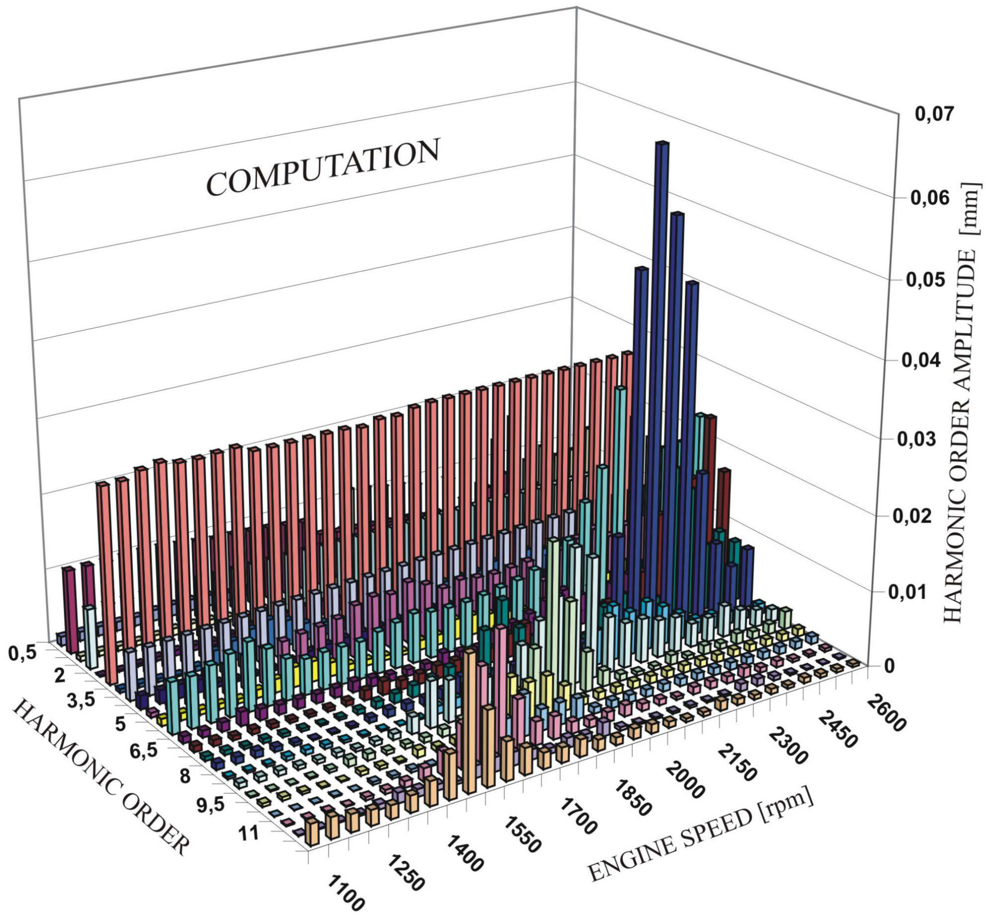

Fig. 7.4 Computed axial vibration order analysis of an in-line 6-cylinder engine crankshaft pulley without a damper

Fig. 7.5 presents computed and measured pulley axial displacements and **Fig. 7.6** presents the confrontation of harmonic orders of computed and measured pulley axial displacements.

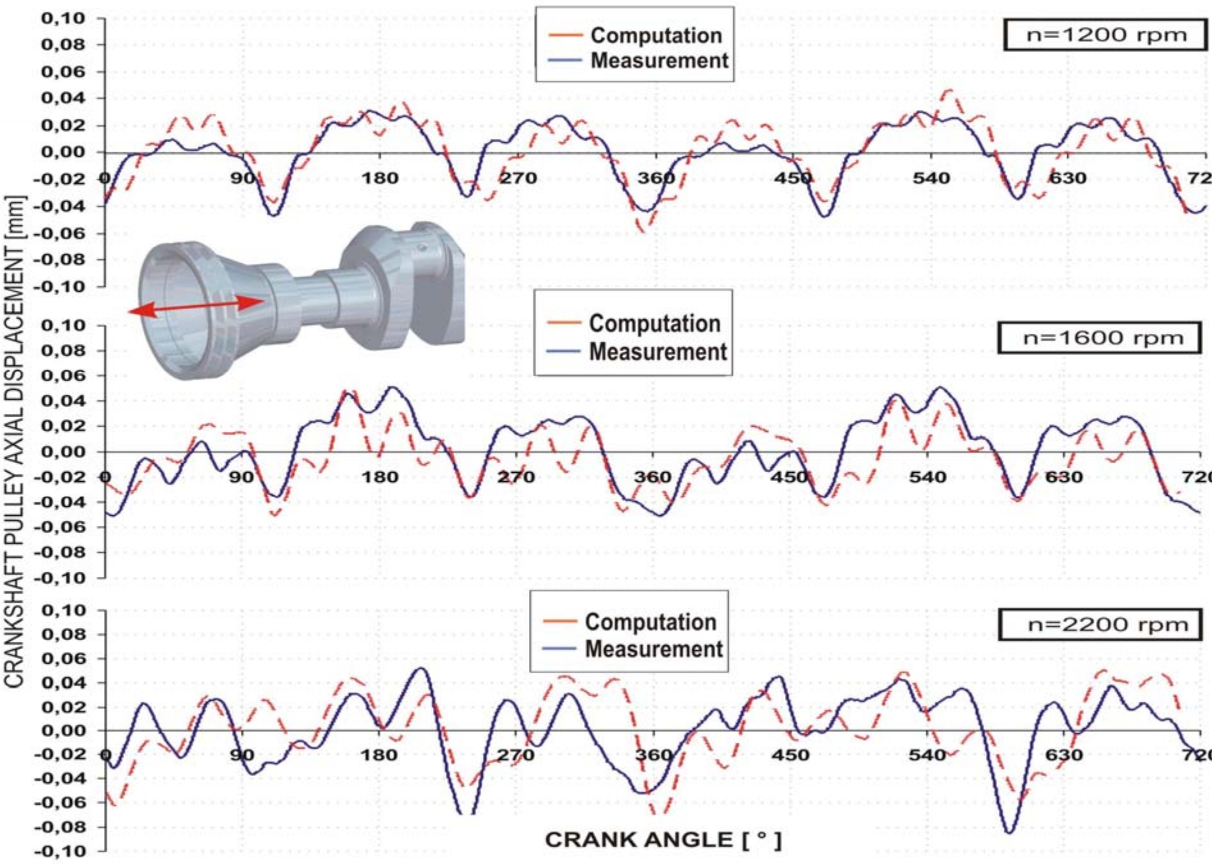

Fig. 7.5 Axial displacements of a Diesel in-line 6-cylinder engine crankshaft pulley without a damper for some engine speeds

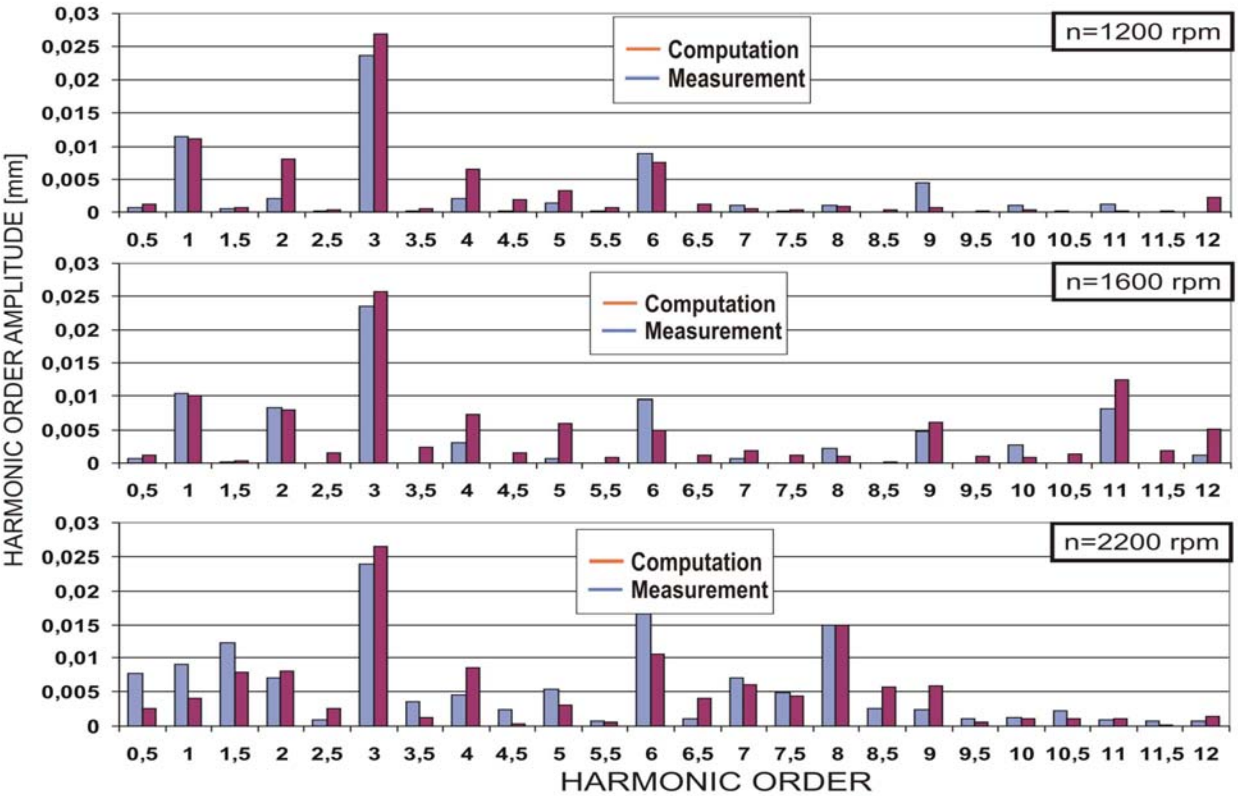

Fig. 7.6 Harmonic orders of axial displacement of a Diesel in-line 6-cylinder engine crankshaft pulley without a damper for some engine speeds

The crankshaft is the major part of the cranktrain. The FE model of a Diesel in-line 6-cylinder crankshaft includes small geometrical details as are the main journal radius, crank pin radius or main journal lubricating canal. Stress or strain results can be found and the stress results can be used for fatigue calculations. **Fig 7.7** shows von Mises stresses for a 90° crank angle step at 2200 rpm during a cranktrain period.

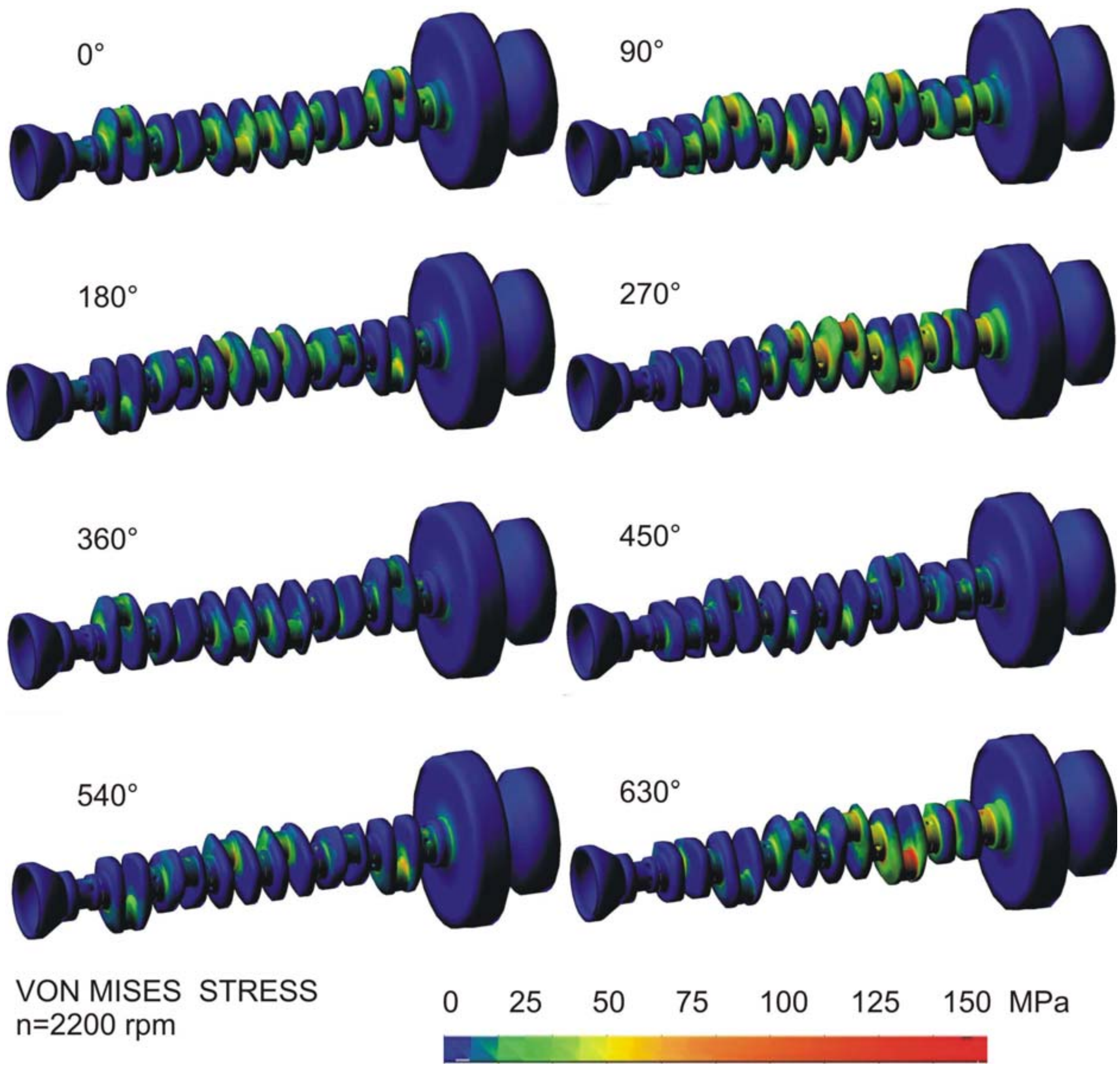

Fig. 7.7 Diesel in-line 6-cylinder crankshaft von Mises stresses for a 90° crank angle step at 2200 rpm

The Diesel in-line 6-cylinder engine block doesn't include small geometrical details and cannot be used for maximum stress spots, but can be used only for basic stress distribution.

7.2 DIESEL V8 ENGINE

The same principles for a Diesel in-line 6-cylinder cranktrain govern for a Diesel V8 cranktrain. Only a few Diesel V8 engine results are presented due to a restricted thesis extension.

A mass of an engine clutch is added to the V8 crankshaft before the solution. Outer ring masses of the cylindrical roller bearing are added to each crank throw as well.

The construction of V8 engine crankcase is very stiff and this caused the bending mode shapes to be moved up to higher frequencies. **Tab. 7.3** presents natural frequencies and mode shapes of a V8 cranktrain.

Tab. 7.3 Natural frequencies and mode shapes of a V8 cranktrain without a torsional damper

Diesel V8 engine crankshaft		
Natural freq. number	Natural freq.	Mode shape
[-]	[Hz]	-
1	252	bending
2	296	torsional
3	312	bending
4	383	axial

Critical engine speeds can be found from the first torsional natural frequency (296 Hz) and from some dominant harmonic orders of a combustion pressure, see **Tab 7.4**.

Tab. 7.4 Critical speeds of a Diesel V8 engine crankshaft

Diesel V8 engine cranktrain	
κ	n_{kr}
[–]	[min^{-1}]
4	4440
8	2220
12	1480

The torsional vibration order analysis of the crankshaft front without a damper is presented in **Fig. 7.8**. It is evident that dominant orders with resonances in an engine speed range are 8th and 12th.

Nominal speed of the Diesel V8 engine is 1800 rpm and torsional vibrations of the crankshaft are one of the major problems. A use of a torsional vibration damper is considered on the basis of computational simulations and measurements.

Torsional vibrations from the dynamic simulation are analyzed and the using of a viscous torsional vibration damper is considered. The torsional vibration order analysis of the crankshaft front with viscous torsional vibration damper is presented in **Fig. 7.9**.

The viscous torsional vibration damper causes a reduction of relevant orders in resonances.

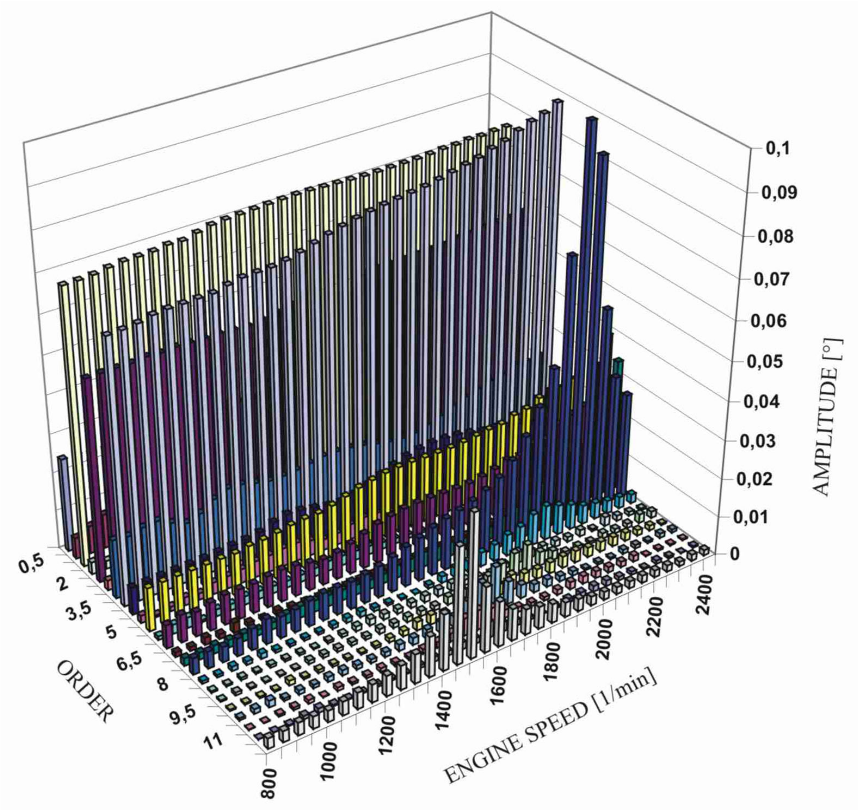

Fig. 7.8 Torsional vibration order analysis of a Diesel V8 engine crankshaft front without a torsional damper

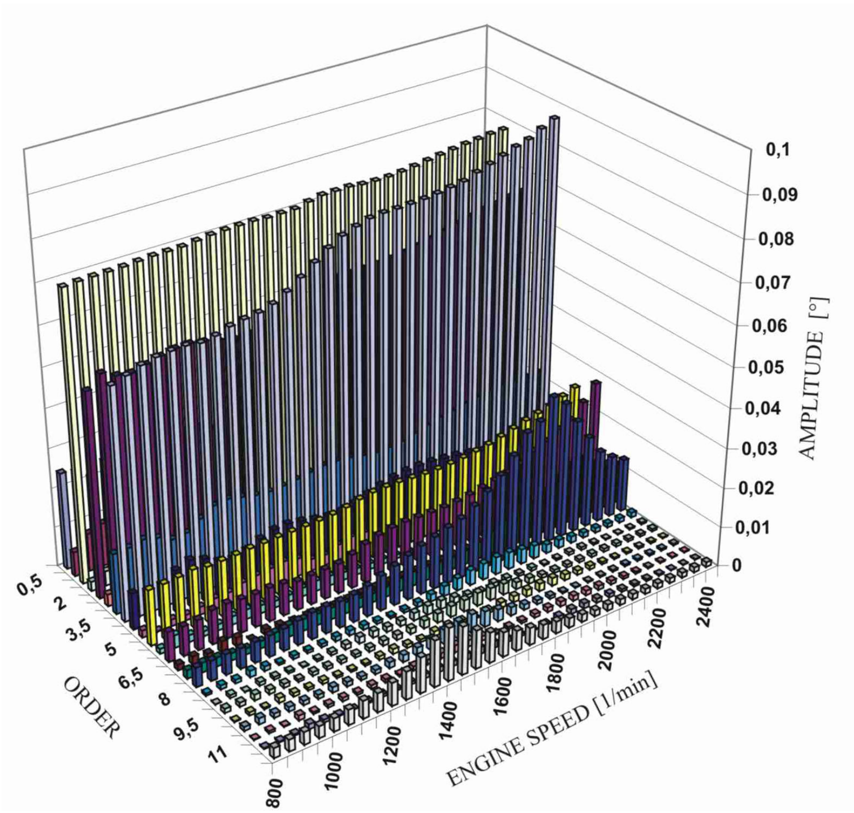

Fig. 7.9 Torsional vibration order analysis of a Diesel V8 engine crankshaft front with a viscous torsional vibration damper

8 CONCLUSION

An internal combustion engine simulation is a very complex problem consisting of many partial issues. Computational modeling of the cranktrain is greatly complicated and this requires a specific approach. The simulation of cranktrain dynamics is a central module of the virtual engine. A measurement of an experimental engine is used for a validation of cranktrain computational model results. Laser measuring technique in combination with special in-house software is an efficient way to verify complex powertrain dynamic models which are being developed as modules of the virtual engine.

Diesel in-line 6-cylinder results correspond well to measurement, mainly the torsional vibrations of the crankshaft. It is necessary to notice that there are many input parameters for the modeling and most of them are not exactly known.

Forced damping vibrations present a fundamental problem for most engines and are well described by many computational models. In the case of this work, cranktrain model results present very good agreement with the measurement. Generally, axial damped vibrations are another important computational problem and don't exist so many computational models as for torsional vibrations. Axial vibration results from the cranktrain computational model present relatively good agreement with the measurement, but for better agreement it is necessary to continuously improve the cranktrain computational model.

The computational cranktrain model makes it possible to find out vibrations of each flexible body part, for example a bottom engine cover. Bottom engine cover vibrations present good agreement with the measurement for lower harmonic orders of vibrations. This is caused by the fact that a combustion pressure is known only for one value per one crank degree. For better agreement, it is necessary to use more accurate combustion pressure data together with more accurate FE models and reduction of FE models up to higher frequencies.

FE model stress results can be used for strength examinations or fatigue calculations. For more precise stress and subsequently fatigue results of all flexible bodies, the more detailed FE models have to be created.
Applied viscous torsional vibration dampers with extremely high viscosity silicone fluids are effective means of modern engine vibration and noise reduction.

For crankshaft dynamic calculations, traditional methods based on simple calculation of torsional and bending vibration are state of the art but new calculation methods are necessary to achieve more accurate results that allow a design that is nearer to the mechanical limit.

The combination of MBS and FEA methods is the most efficient way of calculating cranktrain dynamics. The results are boundary conditions for further investigations, not only for stress issues, but also for questions concerning friction and wearout of bearings or pistons.

The new techniques will replace the traditional methods with the increasing power of modern computers. However, further studies are necessary to optimize the interaction of the CAE programs and to validate the new technology.

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AUTHOR'S CURRICULUM VITAE

M.Sc. Pavel Novotný
Brno University of Technology
Institute of Transport Engineering
Technická 2
616 69 Brno
Czech Republic
E-mail: pnovo@centrum.cz
Phone: +420 541 142 272
Fax: +420 541 142 828

EDUCATION

Brno University of Technology, Institute of Transport Engineering, (ongoing)
Czech Republic, Doctor of Philosophy (Ph.D)
Adviser: Prof. Václav Píštěk, D.Sc
Concentration*: Cranktrain dynamics simulations*

Otto-von-Guericke University Magdeburg, Faculty of Mechanical Engineering (2003)
Institute of Measurement Technique and Reciprocating Machines, Germany,
Study visit
Adviser: Prof. Dr.-Ing. Helmut Tschöke
Concentration: *CFD simulation of diesel particle filter*

Brno University of Technology, Institute of Solid Mechanics, Czech Republic (2001)
M.Sc degree in Engineering Mechanics
Adviser: Doc. Ing Jindřich Petruška, CSc
Master's thesis*: The comparison of ANSYS and DESIGN SPACE on FE gearbox analysis*
M.Sc. diploma with honors

TEACHING/RESEARCH EXPERIENCES

Lecturer: *Statics, Dynamics, Kinematics, Strength of Materials, Machine Parts at University of Defense, Institute of Mechanics and Machine Parts* (from 2001)

Cooperative researcher: *Diesel engine dynamic simulations and measurements at Brno University of Technology*
Bioballistic and armour penetration numeric simulations at University of Defense

LANGUAGES AND SKILLS

Czech: *mother tongue,* English: *fluent,* German: *intermediate*
Knowledges: *Area of mechanics of solids, Dynamics of combustion engines*
Very good knowledge of softwares: FEM - *ANSYS, LS-DYNA;* MBS – *ADAMS;* CAD –*PRO/Engineer*
Intermediate knowledge software: FEM – *NASTRAN*; CFD - *AVL FIRE*; CAD – *CATIA, Invertor*; other – *MATHCAD*, *MATLAB*, *COREL DRAW* and *MS OFFICE*