Developing a Virtual Engine and Transmission Combined with a Modern Geartrain Simulation Process

ABSTRACT

Simulation tools are an integral part of a modern research and development program to determine the dynamic system response and the component stresses. These investigations are based on the utilization of the latest multi-body and finite-element techniques. The simulation tools are usually applied to analyzing and optimizing shafts, clutches, chain/belt drives, bearings, levers, brackets, housings and many other components.

However, gears are still frequently analyzed using semi-empirical methods that are based on DIN, ISO and AGMA. Geartrain developers have also added to their own knowledge base the experience they have gained in their investigations. The primary difficulty with this type of gear investigation is that the rotating gears have large contact surfaces with complex nonlinear mechanical contact properties.

FEV's research has created a new means of analyzing and optimizing gear drives using commercial multi-body and finite-element software. The variations in tooth load and stresses during the roll-off procedure can constantly be determined. A realistic description of the roll-off contact for each pair of teeth integrates the detailed geometric surfaces and the complex nonlinear contact of the surfaces. This form of analysis can identify the effects modifications of the tooth profile, tooth width and tolerances have on the geartrain. This form of geartrain modeling creates a realistic dynamic system behavior.

INTRODUCTION

The enactment of stringent legislation of emissions during product development and use has added further limitations to progressive powertrain development. Conflict is created for any development project between the technical requirements and the necessary developmental efficiency. This conflict is due to increased global competition and expenses for the automakers as opposed to the demands of the customers wanting to protect the environment, increasing fuel economy and reliability.

The simulation process has demonstrated that it can effectively manage the apparent conflicting requirements for a low power/weight ratio, reduced size, and new technologies, as opposed to a decrease in the development time, diminishing risk and reducing production costs.

Reducing the development period requires that the design (CAD) and calculation functions (CAE) be performed concurrently and that they interact with one another (See Figure 1). The product in this instance must be analyzed and optimized for the various disciplines and coordinated with the progress of the development. During the first half of the development phase the efficiency of the virtual CAE methods is high because of the proactive nature of the methods. In the second half of the development phase, the methods are not as effectives because the processes that occur at that time are strongly characterized as being reactive. Corrections that are made during the final phase of development are associated with high development and alteration costs. Early implementation of CAE methods in the development cycle provides a great deal of initial information (front loading). Applying this modern process to the development of an internal combustion engine allows for the deletion of a construction stage, which facilitates the two remaining CAE-intensive, high-quality construction stages.

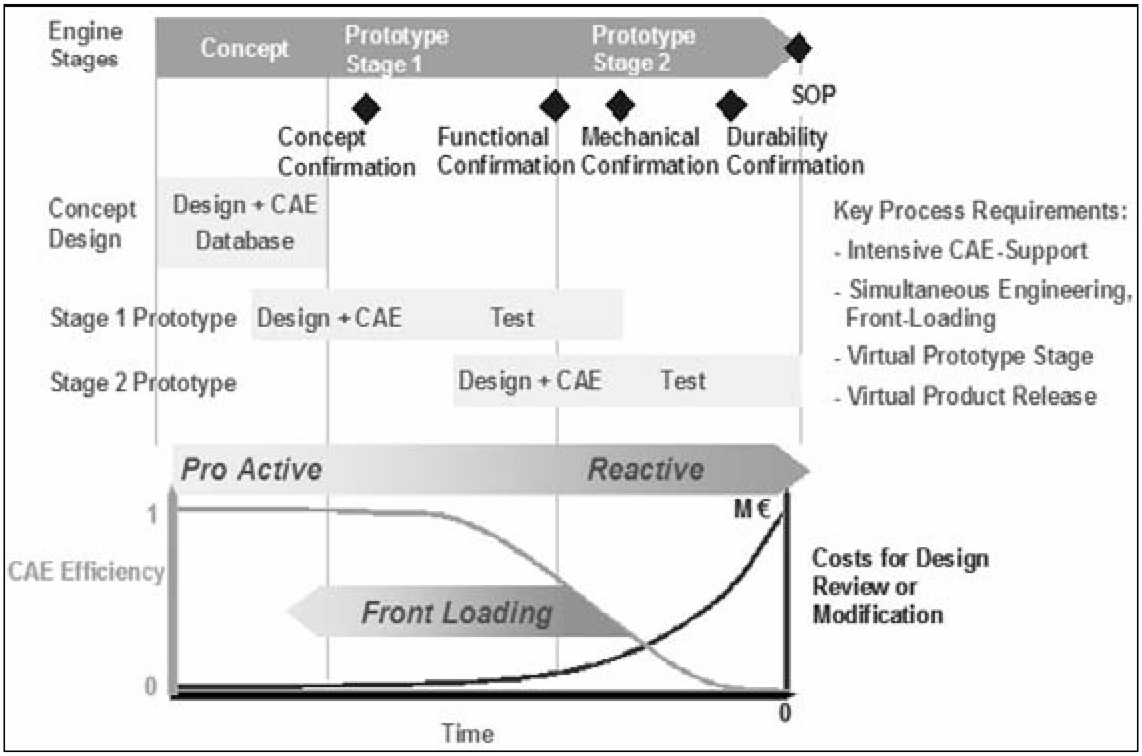

Figure 1: Simultaneous CAE in the Development Plan

Examination of a product's life cycle (Figure 2) shows that each stage leaves some form of impact on the component. A product's metallurgy, primary forming, machining and use by the consumer help define its properties, design strategy and service life. Theses individual stages may be combined into two groups, product formation and product utilization. The product's system structure and the phase of use are recorded as a standard practice using virtual methods, for the purpose of dimensioning and optimization. Current CAE methods also potentially include steps for semifinished goods, unmachined parts production, aftertreatment and their impact on any components. Casting and forging simulations that analyze the solidification processes, crystalline structures, and the remaining internal stresses on the component are two common examples.

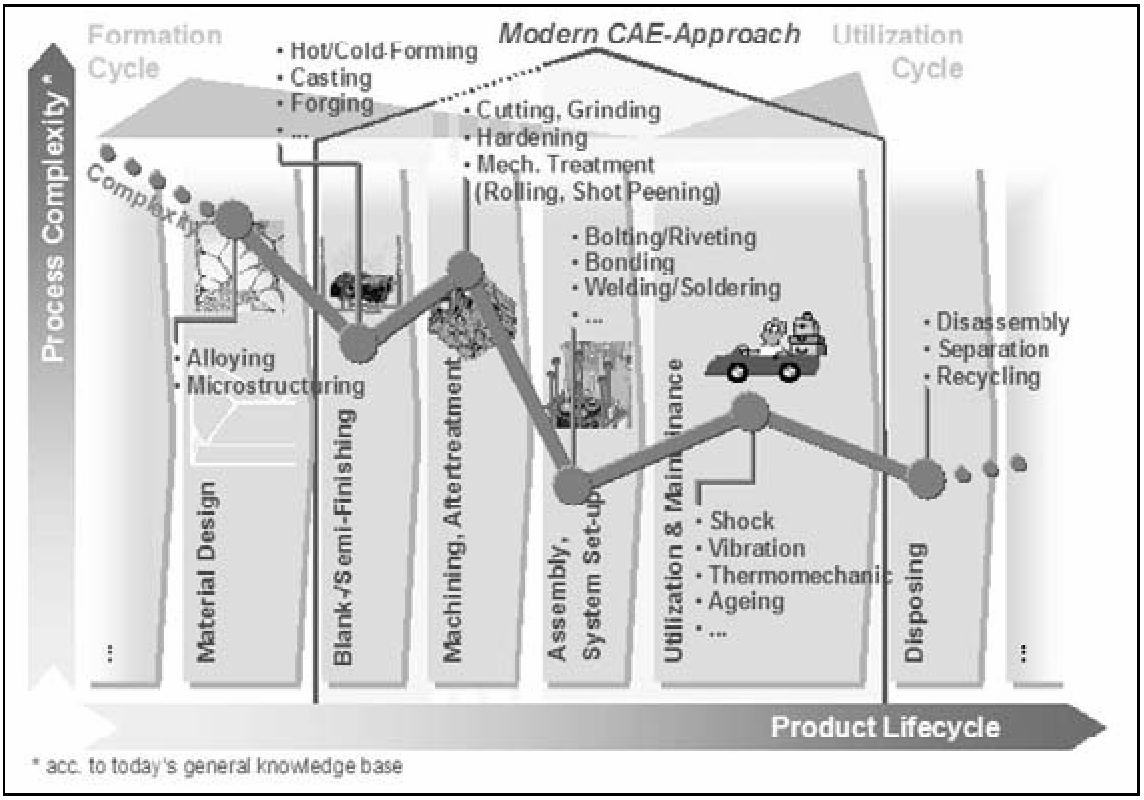

Figure 2: Stages of Product Life Cycle

Gauging the complex nature of the individual processes in opposition to a virtual representation; specifically, the metallurgy and aftertreatment processes are rated very highly. Specific information about the processes taking place within these operations of the components and how they are influenced is still relatively small or is not generally available.

SIMULATION OF POWERTRAIN MECHANICS

The interactive and simultaneous approach between design and CAE during the overall development requires computation models which are adjusted to the respective individual steps. Therefore, a whole portfolio of tools and simulation models from the areas of thermomechanics, structural dynamics and fluid mechanics is applied in powertrain simulation.

According to Figure 3, these are characterized by the model depth on the one hand, and by the modeling scope on the other hand. Models of a smaller scope typically concentrate at an early project stage (concept) on the simulation of product utilization, and in most cases also on the individual components. If applicable, comprehensive models examine the overall system behavior within the scope of a validation and also include the stages of product formation ("life cycle CAE").

A large bandwidth of modeling depth results from the accumulated experiential knowledge in the form of databases on the one hand (least modeling depth) and the predominantly physical model formation, on the other hand. In the ideal case, the physically oriented model formation significantly reduces the required experiential knowledge of the developer of a component. In return, however, it also requires a greater degree of knowledge of methodology. Ranging between these two extremes are the semi-empirical methods which are usually the basis for technical standards or also for expert software systems.

Within this modeling range, there are typical modeling techniques for analyzing and optimizing the following physical effects.

- Rigid body dynamics & system vibrations
- Deformation, structural dynamics, gyroscopy
- Hydrodynamics in conforming contacts (e. g. slide bearings)
- Mechanics of non-conforming (concentrated) contacts
- Pressure, velocity in fluid streams
- Crystalline structures
- Mechanical & thermal stress on components
- Damaging and working life

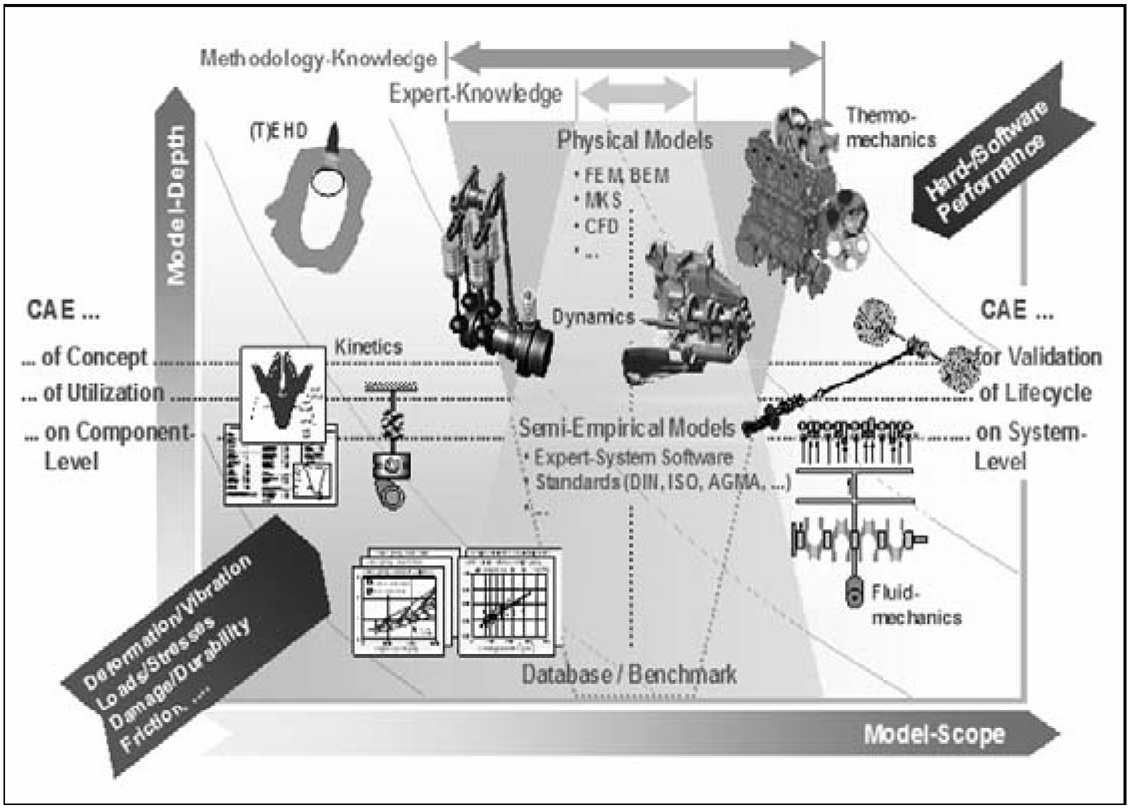

Figure 3: Virtual Testing of Different Model Depths and Scopes (Selected Examples)

The present limits are set by the available computer performance and the functionality of the tools. The rapid and progressive development of computer performance, however, allows a relatively steady expansion toward physical models with a large modeling scope at the same time.

The analytical tools available for individual disciplines are even today usually only partly compatible, i.e. they can only communicate via separate interfaces. The pursuit of high quality of the simulation results therefore requires increasing networking of simulation methods and also a simultaneous examination of coupled physical effects.

GEARTRAIN SIMULATION

With regard to their acoustic effects, the types of vibrations occurring in geartrains are to be subdivided into “whining”, “rattling”, and “hammering”. While the whining of the transmission is caused by the harmonious excitation of the gear meshings running under constant load condition (parametric excitation, engagement/disengagement jerks), rattling phenomena and hammering are effects of backlash where tooth flanks collide. Gear rattling occurs in low-load gear stages, as e.g. the idle gear stages of a manual transmission and is caused by the stimulation of vibrations of neighboring systems. With regard to hammering, the flank changes are forced by strongly alternating external moments. When the flanks collide, a strong impact occurs which decisively determines the load and stress on the teeth. Highly dynamic excitations and hammering are typical of timing drives and injection pump gear drives (Figure 4). Therefore, these place the highest demands on the validity of the development tools and on the quality of the results of the virtual analyses.

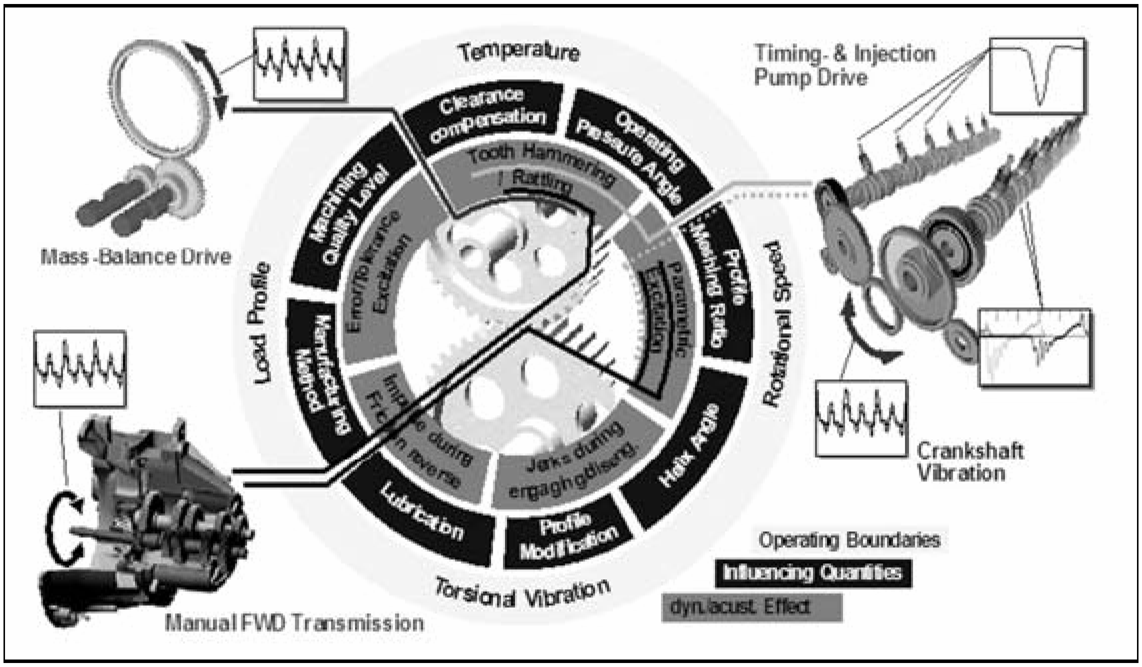

Figure 4: Effects and Influencing Parameters of a Gear Meshing, Typical Vibration Behavior of some Applications

For the geometrical design of the tooth profile, the stress and load capacity analysis of gear meshings, the semiempirical methods of DIN, ISO, or AGMA, or procedures derived from them have been used so far. These methods, however, are based on limited kind of operating modes and tooth geometries. Relatively simple and often also linear simulation models are used to determine the dynamic system behavior. An analysis of scientific investigations of the past with regard to the simulation of dynamic loads and stresses on gear teeth shows an essential potential for tooth contact modeling. These simulations are clearly mathematically oriented due to the treatment as torsional vibrators and the "design" of the meshing parameters by the superposition of individual values. The analytical approaches of the course of tooth stiffness (e. g. according to Diekhans /1/, Weber/Banaschek [9], Winter/Podlesnik [10], Ziegler [11] used there are only valid for limited tooth design and are fraught with uncertainties. Remarkable in this context is that, for the alignment of the model of a timing and injection pump gear drive with measured values, Prestl [5] must reduce the mean meshing stiffness initially determined in accordance with Ziegler or DIN to approx. 1/6 (!) of the original value.

By contrast, the description of a truly rolling contact on the basis of contacting surfaces (Figure 5) opens the general possibility to apply the parameterization of the tooth contact to the individual tooth (individual tooth or tooth-pair selective examination). Utilizing the correct geometrical and mechanical representation, the following may then be recorded in a realistic manner:

- Load share carried by the several, simultaneously engaging teeth
- Load- and speed-dependent (dynamic) meshing ratio
- Jerk during engaging/disengaging, friction and impulse during friction reverse
- Effects due to tooth forming errors and deliberate deviations
- Load-dependent deviation of rotation (transmission errors), parametric excitation
- The reciprocal effect of tooth deformation (error propagation)
- The stresses on individual teeth, due to hertzian stress at the flank and base of the tooth stress
- FE-based damaging and working life

The main reason for the previously relatively simple model approaches for tooth contact was the lack of software functionalities with regard to the formulation of massive and "migrating" contacts, respectively, in nonconforming, concentrated contacts.

The development of numerical models in the contact area (e. g. Hallquist [2], Kikuchi/Oden [3], Simo/Wriggers/Taylor [7] and their implementation in commercial FE software facilitated the formulation of contact conditions, so that the discrete approach can be resolved completely in the FE software. Here, a distinction is made between the Lagrange multiplier method, which accurately fulfils the contact conditions (kinematic contact), and the penalty method. The latter allows a penetration of the surfaces in favor of a numerical stabilization through the introduction of additional contact stiffness (penalty parameter); however, the contact conditions then are only approximately fulfilled.

The contact properties can be shown very well, with the described solutions as the basis of the finite element method, taking into account the real structure (e.g. edge influence with limited contact surfaces). Due to the currently available computer performance, however, these are at best applicable to static analyses and locally limited contact surfaces. Thus, they are suitable for the determination of parameters of machine elements.

For dynamic analyses of gears with massive or “migrating” contact, respectively, contact models are required which, for the purpose of numerical efficiency allow a distinct penetration and a lower degree of discretion with a still very good reproduction of the contact properties. Moreover, a random surface geometry, a random formulation of the contact parameters, and parallel contact are to be demanded. On the basis of such a truly rolling contact model, FEV by standard analyses and optimizes geartrains in powertrains with regard to dynamic operating performance and security against failure with a high quality of results at the same time. The question of load share of the meshing teeth, the dynamic and load-dependent effects of the meshing, and the dynamic contact ratio is therefore directly answered by and during the simulation.

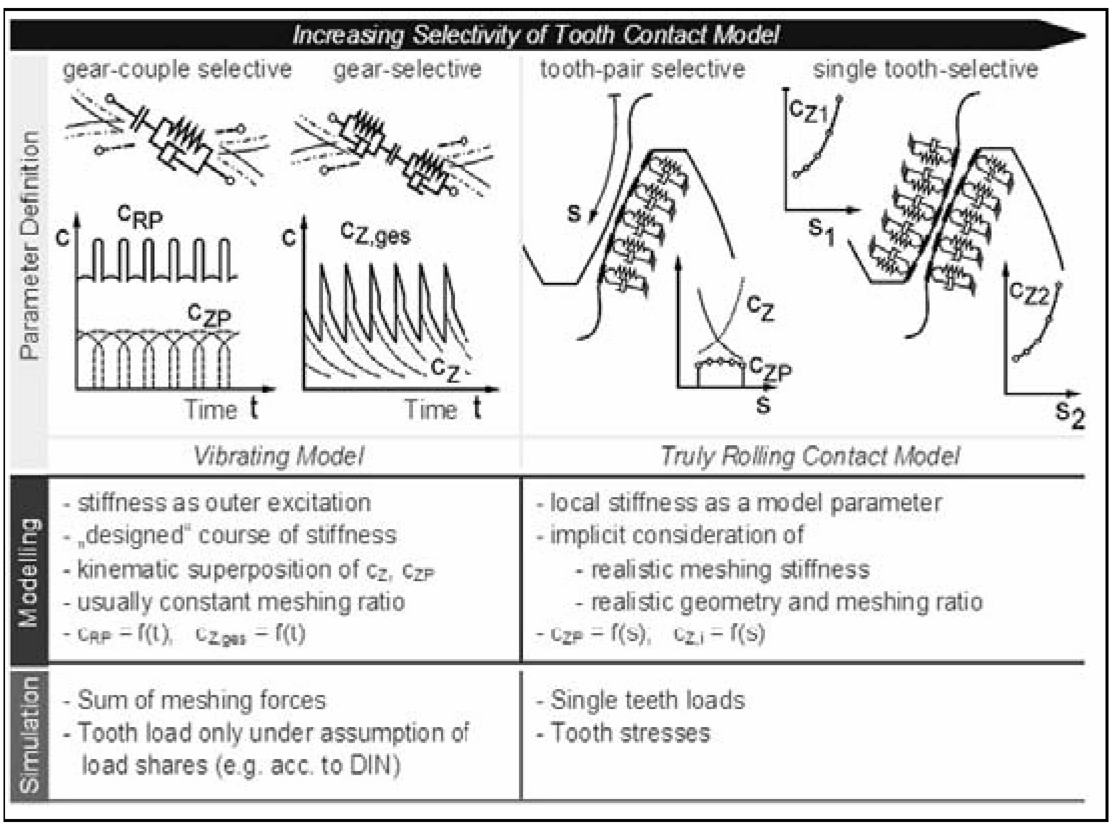

Figure 5: Modeling Techniques and Parameter Assignment in the Meshing

The geartrains are modeled on the basis of the commercial MKS tool ADAMS® which, in addition to the powerful contact algorithm, has also other techniques for representing the hydrodynamic bearings and the flexible structures which are important for the overall system performance. Figure 6 shows exemplary the individual stages of the extensive life cycle analysis process of a transmission.

Since with gear meshings, the design of the profile including the root fillet has a decisive influence on the contact situation and the stress, the tooth profile both in its macro form (module, profile shift, pressure angle, root fillet, etc.) and its micro geometry (profile/flank modification) is determined within the step of a simulation of the manufacturing process. On the basis of this manufacturing simulation, the detailed geometry data of the toothed wheel are used for its FE analysis. The non-linear tooth and wheel body properties (stiffness, hertzian stresses at flank, tooth root stress) are determined in dependence on the contact point (profile and lateral flank trace) and the load on the flank within the step of the rolling contact simulation. The gained tooth properties are then made available to multi-body simulation in the form of multi-dimensional tooth property maps.

Taking an operating load collective of the transmission as a basis and taking further structural boundaries (elasticities, eigenbehavior, flexible bodies) into account, the dynamic drive train behavior as well as the local meshing procedures can be determined and transformed into stress collectives of the single components. It needs to be emphasized that with regard to the stress analysis, this procedure relies on detailed FE results. For the subsequent root damage calculation, instead of tests on standard reference test gearwheels (DIN 3990, part 5) which are only available to a limited extent, fatigue strength values and wöhler curves of simple, non-indented samples may be used which are available for many materials and surface treatments. For the flank damage calculation there are further results available beyond the hertzian stress values like contact misalignment and relative sliding velocity. As a result, for the first time a "uniform" analytical process from fabrication to damaging on the basis of progressive, physically-oriented simulation models is available.

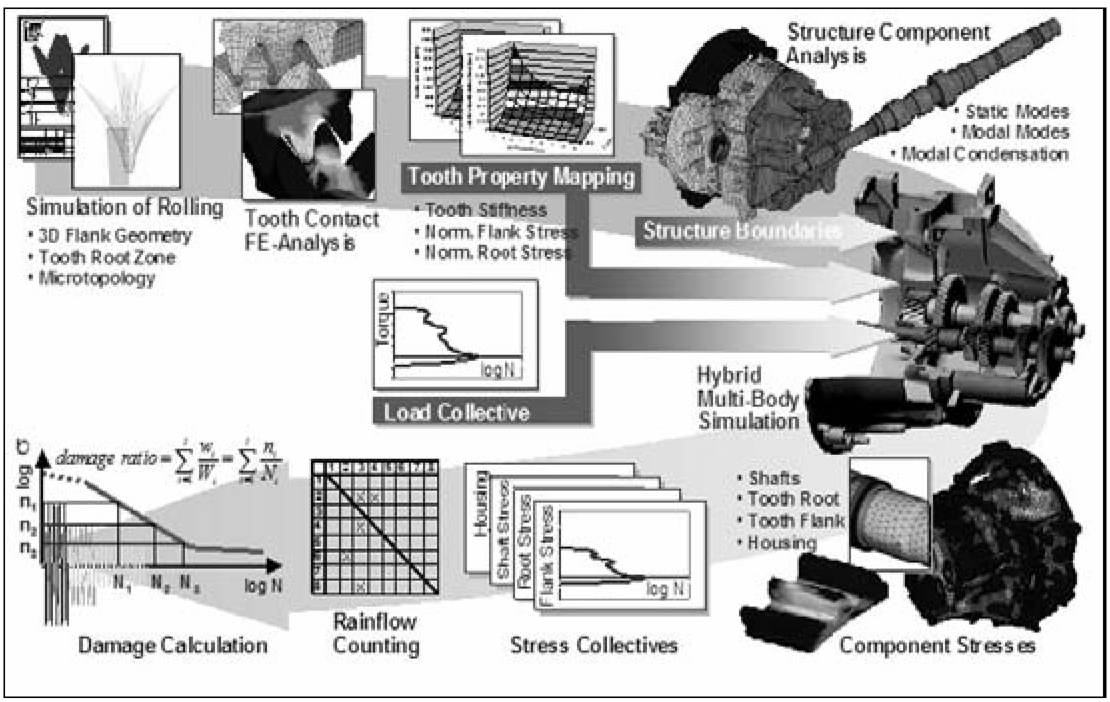

Figure 6: Life Cycle Analysis Process of a Transmission

CONCLUSION

Simulation of the progressive geartrain method is based on a rolling tooth contact model that highlights a detailed geometrical description of the tooth profiles and the equivalent nonlinear tooth properties. Simultaneously, it uses the existing functionalities of efficient multi-body system algorithms in connection with structural dynamics and hydrodynamics. The macro form and the micro-geometrical properties of the teeth are established by simulating the stepd of the machining process. This provides the necessary input for the geometric contact definitions of the multi-body analysis model and provides the foundation for a finite-element based rolling contact simulation where the static mechanical tooth properties are established. The multi-body contact algorithm relies on these finite-element results in the form of tooth property maps that highlight tooth stiffness, tooth root stress and hertzian stress at flank. Utilizing this method allows multiple variations in the design of the tooth profile (e.g. macro form and deliberate micro modifications). Also, meshing conditions like backlash can proceed efficiently with high degree of model depth and accuracy.

Accordingly, a standardized analytical process is available that can be applied from fabrication to malfunction on the basis of a progressive, physically-oriented simulation model.

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