

Crankshaft and Bearing Analysis Process for a Light Duty Automotive Engine

Ilya L. Piraner and Matthew P. Meek

Cummins Inc

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ABSTRACT

Design of a light duty diesel for an automotive market presents contradictory challenges related to passenger car requirements for a compact, low weight design versus the diesel's base engine that must withstand cylinder pressures that are much greater than that seen on gasoline. This was a particular challenge for Cummins because of two reasons. First, design practices developed for Cummins' traditional heavy duty and industrial markets could lead to over-design, particularly for those items that have wear based life limits like bearings. Secondly, in the pursuit of new engine business it is necessary to be able to quickly yet accurately generate conceptual engine space claims for a variety of vehicle and engine specifications. When applying traditional guidelines for crank and bearing sizing, the resulting base engine size appeared an unsolvable problem relative to size and weight requirements. To overcome these difficulties a number of new processes have been established, two of which are described in this paper.

INTRODUCTION

Design of a light duty diesel for an automotive market presents contradictory challenges related to passenger car requirements for a compact, low weight design versus the diesel's base engine that must withstand cylinder pressures that are much greater than that seen on gasoline. This was a particular challenge for Cummins because of two reasons. First, design practices developed for Cummins' traditional heavy duty and industrial markets could lead to over-design, particularly for those items that have wear based life limits like bearings. Secondly, in the pursuit of new engine business it is necessary to be able to quickly yet accurately generate conceptual engine space claims for a variety of vehicle and engine specifications. When applying traditional guidelines for crank and bearing sizing, the resulting base engine size appeared an unsolvable problem relative to size and weight requirements. To overcome these difficulties a number of new processes have been established, two of which are described in this paper.

The first process is related to crankshaft / bearing system optimization based on Design of Experiment (DOE). In that process all necessary data which normally would come from CAD or FE analysis were curve fitted using DOE and used in a fast optimization routine which defines major crankshaft / bearing parameters for a Vee engine configuration. The second process is related to formulating bearing acceptability criteria based on duty cycle. Standard Cummins procedure for bearing analysis requires consideration of only two primary conditions along the engine torque curve. This approach proved to be a reliable instrument in designing crankshaft and bearing systems for traditional Cummins' applications. However, it resulted in an overly conservative design for a light duty engine which spends limited time at extreme conditions. A new approach which takes into account engine duty cycle has been developed first using simple 1-dimensional rigid bearing model and then implemented also with 2-dimensional elasto-hydrodynamic bearing model.

The current trend in the Cummins' Analysis Lead Design initiative is to develop a two stage approach for most of the analytical processes. The first level processes are used in the preliminary, concept stage of engine development as well as further in the development cycle. The main characteristics of these processes are speed, robustness, simplicity as well as sufficient accuracy. These methods are used to explore the design space via parametric studies, DOE and optimization processes to come close to the final design which would satisfy all major requirements. Normally, the inputs for computational models on that level are based on historical data or easily calculated based on engine major dimensions; execution time for one simulation would be just a few minutes. The second level of modeling is much more complex, it may require either CAD/FE or experimental data, may involve detailed 3-dimensional analysis and multi-body simulation. While adding complexity often helps in understanding true physics of complex interactions, inevitably it adds cost, computational time and may actually reduce robustness of simulation. Therefore,

such methods are used predominately in troubleshooting, discovering common trends and calibrating simpler (1st level) models. This explains the emphasis Cummins places on the first level of computational models.

A number of considerations need to be addressed in the preliminary crankshaft and bearing design analysis. Among these considerations are crank stress, bearing performance, crankshaft balance, integrity of bolted joints, damper durability and the impact of the crank on block stress and deformation. Though each consideration may require a detailed and complicated analysis, simplified approaches have been developed over the years by the industry and successfully applied at Cummins. The current paper is predominantly concerned with crankshaft stress and bearing performance

DESIGN PROCESS AND COMPUTATIONAL MODELS

DESIGN PROCESS OVERVIEW

Other than piston (cylinder bore) diameter, the crankshaft is the component most responsible for setting the base size of an engine. To achieve the ability to navigate through the crankshaft design space it was necessary to map out likely ranges of key defining crankshaft characteristics. With these characteristics and associated ranges mapped, it was then possible to analytically build a variety of potential crankshaft and bearing geometry options. With the order afforded by the use of central composite design of experiments, it became feasible to map key design output variables (crankshaft stress to moment ratios, fillets stresses, rod and main bearing film properties, crank balance, etc.) against key input variables (crankshaft and bearing geometry, cylinder pressure, bearing diametrical clearance, etc.)

Key defining crank geometries for this study were identified in two categories. First were gross sizing parameters; stroke, main diameter and length, rod pin diameter and length, rod pin fillet package width, main fillet package width, and counter weight width and radius. Second were fillet sizing parameters; major radius, minor radius, and undercut depth (for consideration of undercut fillet options.)

To make the parameters independent of the actual crank size, all parameters were taken as a ratio to rod pin diameter. Via study of several existing crankshafts it was possible to determine the likely ranges of these ratios. These were taken into consideration when setting the parameter ranges in the DOE. When mapping the DOE output variables they were done so back to dimensionless parameters, not a specific part size. An illustration of the DOE process for fillet stress definition is shown in Figure 1.

This allowed for significant flexibility in being able to still explore potential design space regardless of the actual key input size, such as desired engine displacement or bore size.

Upon completion of the DOE process it became possible to provide an input file or "front end" to a traditional crank and bearing analysis routine that typically only responds to one set of inputs. This input file would specify actual geometry sizes and other key inputs, while the analysis routine would use the dimensionless, ratio terms to find where on the trade off curve that particular geometry combination was for key output variables like fillet stiffness, stress to moment ratios (all leading to fillet stress) and bearing film properties.

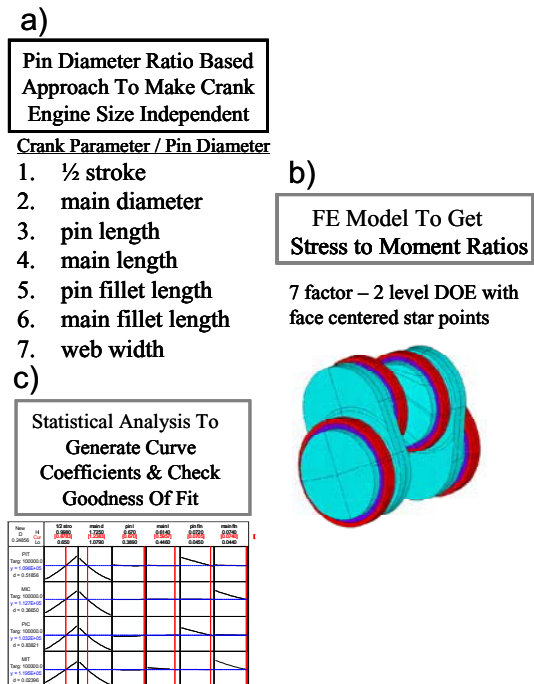


Figure 1. DOE Process Illustration

Since V8 crankshaft has free first order couple it has to be balanced with a proper choice of the counterweights. The most efficient way to cancel the couple is to spread counterweights as far apart as possible. This is why in the V8 engine the end webs are normally made thicker than the interim webs. On the other hand the size of the counterweights is limited by the engine length and height since the counterweight OD is constrained by the available space which in turn is defined by the piston and connecting rod dimensions. To automate crankshaft balance the web and counterweights moment of inertia were parameterized in ProE and curve fitted based on DOE analysis. This allowed for the following automated process:

1. Set limit on the crankshaft length;
2. Set initial value of the crankshaft webs thickness and connecting rod length;
3. Increase width of the end webs until crankshaft balance or maximum length is achieved;
4. If limit is achieved make an incremental increase in the connecting rod length and proceed to step 3 until crankshaft balance is achieved.

The described process leads to a balanced design while satisfying constraints on the overall engine length and minimizing engine height.

With this approach it significantly reduced the time to arrive at a first pass crankshaft and bearing size for any given set of inputs and assumptions. Particularly for new light duty entries, it enables comparison of the resulting part sizes needed based on the design criteria used (such as heavy versus light duty for bearing film properties.)

CRANKSHAFT AND BEARING SIMULATION MODEL

While many commercial software packages exist for crankshaft and bearing analysis, none of them deemed to bridge the gap between simplicity and accuracy. Most of the known packages employ so-called statically determinate (S-D) crankshaft scheme for calculating crankshaft fillet bending stress. This scheme is recommended by International Association of Classification Societies [1] for marine engines and generally gives a good conservative estimate of the fillet stress. However, in the automotive world where special constraints are extremely stringent it is very important to obtain more accurate estimates which would allow for a less conservative design. For this purpose Cummins uses a model which couples statically indeterminate (S-I) crankshaft scheme with a fast model of hydrodynamic bearings. Figure 2 shows that for a V8 crankshaft the difference in the maximum bending stress between S-D and S-I models may achieve 20%.

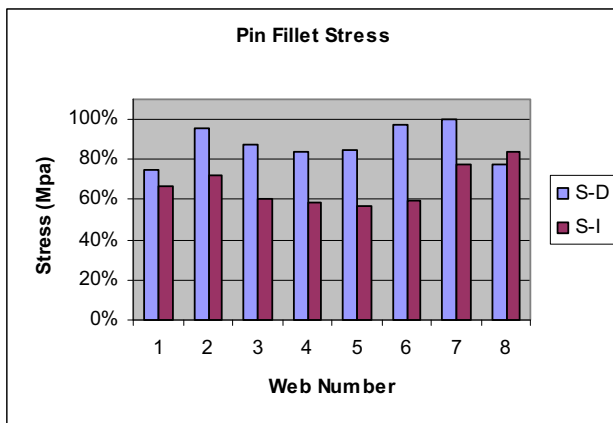


Figure2. Comparison between Statically-Determinate (S-D) and Statically-Indeterminate (S-I) models

In the most common crankshaft design all webs are made the same, therefore S-D scheme yields maximum stress value very close to S-I scheme which shows maximum stress in the end web. In the interim webs stress becomes lower as a result of the unloaded action of the internal bending moments from the neighboring throws. In the V8 engine with a cruciform crank, because of necessity of the balancing free moments in the limited space, the end webs are made thicker which in turn reduces stress in those webs and the maximum

stress may migrate to the interim webs. Therefore, the use of statically indeterminate scheme is imperative. Cummins uses a coupled crankshaft/bearing scheme first suggested by Welch and Booker [2].

Figure 3 shows schematics of the crankshaft model which is solved using matrix transfer method [3]. It allows for a simple expression relating state vectors Z_i^R and Z_{i+1}^L :

$$Z_{i+1}^L = T_i Z_i^R \quad (1)$$

where state vector Z has following components:

$$Z^T = [u_y \ \phi_x \ m_x \ f_y \ u_x \ \phi_y \ m_y \ f_x \ 1],$$

u_y and u_x are displacements along corresponding axes,
 ϕ_x and ϕ_y are angles,
 m_x and m_y are moments and,
 f_y and f_x are forces along the axes.

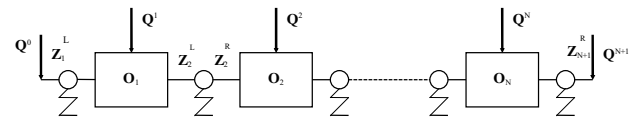


Figure3. Crankshaft Model

Transfer matrix T_i is built based on beam representation of throw. Similarly, a transfer matrix of the support P_j can be created which links the state vectors before and after support #j:

$$Z_j^R = P_j Z_j^L \quad (2)$$

Schematics of main bearing support model is shown in Figure 4, where:

k_{xj} and k_{yj} are support stiffness in two directions;
 ϕ_{xj0} and ϕ_{yj0} are angular misalignments of the support;
 u_{xjh} and u_{yjh} are linear misalignments of the support;
 u_{xj} and u_{yj} are hydrodynamic displacements of the journal center within bearing clearance.

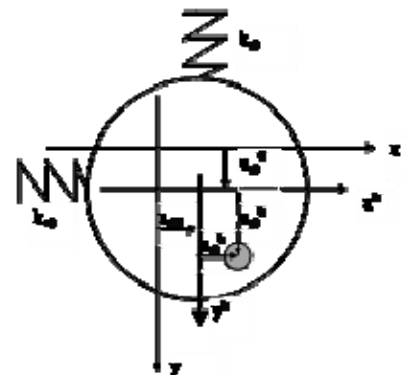


Figure 4. Bearing Support Model