

A Multi-Physics 3D Modeling Methodology for Multi-Cylinder Diesel Engine Thermal Management and Fatigue Life Prediction

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ABSTRACT

Durability assessments of modern engines often require accurate modeling of thermal stresses in critical regions such as cylinder head firedecks under severe cyclic thermal loading conditions. A new methodology has been developed and experimentally validated in which transient temperature distributions on cylinder head, crankcase and other components are determined using a Conjugate Heat Transfer (CHT) CFD model and a thermal finite element analysis solution. In the first stage, cycle-averaged gas side boundary conditions are calculated from heat transfer modeling in a transient in-cylinder simulation. In the second stage, a steady-state CHT-CFD analysis of the full engine block is performed. Volume temperatures and surface heat transfer data are subsequently transferred to a thermal finite element model and steady state solutions are obtained which are validated against CFD and experimental results. With the goal of emulating an engine test strategy for durability under extreme warm up and cool down cycles, a transient thermal analysis is performed for several loading cycles. Simulated transient temperature traces demonstrate good agreement with time-varying thermocouple measurements on the cylinder head. Temperature distributions at many times during the thermal shock test are utilized to calculate thermal stresses followed by a fatigue life analysis. It is demonstrated that transient thermal stress modeling is critical for the fatigue predictions at some locations on the cylinder head. This comprehensive methodology significantly improves the fidelity of engine thermal management and durability prediction and reduces the cost of engine development under severe fatigue prone conditions.

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INTRODUCTION

The increasing demands for higher power along with stringent regulations for engine emissions have put significant challenges in the engine design process. Commonly, design criteria such as fuel consumption, emissions levels, material temperature limits and limits on local mechanical stress are considered. However, cracks and failure due to thermo-mechanical fatigue (TMF) specifically in cylinder heads is one of the bottlenecks in the design process that must also be taken into account. Thermal loading due to multiple warm-up and cool-down cycles corresponding to cruising cycles of the vehicles is a key contributor to thermo-mechanical fatigue.

The structural durability of the engine under warm up and cool down cycles is often evaluated through a physical test. The physical test is performed by running the engine at an idle condition for a predefined period of time, then switching to a high-load condition relatively quickly and running for a period of time, then switching back to idle. The engine coolant temperature may also be switched during the high-load to idle transition, as the coolant heats up during the high-load section. We refer to this process in the paper as the engine thermal shock test, see [Figure 1](#). The focus of this paper is a methodology which analytically replicates temperature and stress

distribution in the cylinder head as a function of time in order to mimic a physical thermal shock test. We refer to this as the virtual thermal shock test.

The first step in defining thermal loading for thermal stress analysis and life assessment is to define the thermal map of the engine. The capability to solve simultaneously for solid components and engine cooling fluid flow and heat transfer in a Conjugate Heat Transfer (CHT) model has provided an efficient predictive method to understand engine thermal behavior, see [1,2]. However, a major prerequisite for any accurate CHT model is the fidelity of gas side thermal boundary conditions on engine ports as well as cylinder firedecks and liners. Obtaining such boundary conditions require transient multi-cycle simulations of in-cylinder (IC) flow and heat transfer.

In order to improve the fidelity of wall boundary conditions used in the in-cylinder simulation, resulting better predictions of heat transfer on gas side surfaces, we have followed an iterative process between the IC and CHT models, sometimes called the "decoupled approach". In the decoupled approach, typically, cycle-averaged heat transfer data (heat flux or heat transfer coefficient) from a transient in-cylinder simulation is used to calculate the conduction in various

solid engine components. This approach has been used and validated previously [2,3,4,5,6]. Liu and Reitz [3,4] used an iterative decoupled scheme between a finite difference heat conduction code and the KIVA-II code for combustion to calculate engine component temperatures such as piston, head and liner and validated the scheme against test data. KIVA-3V code has been utilized in [5,6] to get the gas side boundary conditions for a subsequent CHT analysis. ANSYS-Fluent code [7] with multi-phase combustion was used in [2] where the combustion chamber was modeled in a closed mode (IVC to EVO) for a 360 Deg computational domain. Other numerical codes have also been utilized with varying levels of detail and complexity, but subsequent thermal load and stress analyses entailed data transfer from the CFD-CHT models in one code to the FEA model in another code typically in different environments and frameworks. The current process utilizes ANSYS Workbench framework to make the data transfer between CFD-CHT and FEA models straightforward and reduce the sources of numerical errors. Furthermore, spatially-distributed coolant and combustion gas side heat transfer data are calculated and utilized to ramp between miscellaneous loads (e.g., low and high loads), a procedure required to mimic engine temperature cycle tests. This enables the accurate modeling of any real engine thermal shock testing.

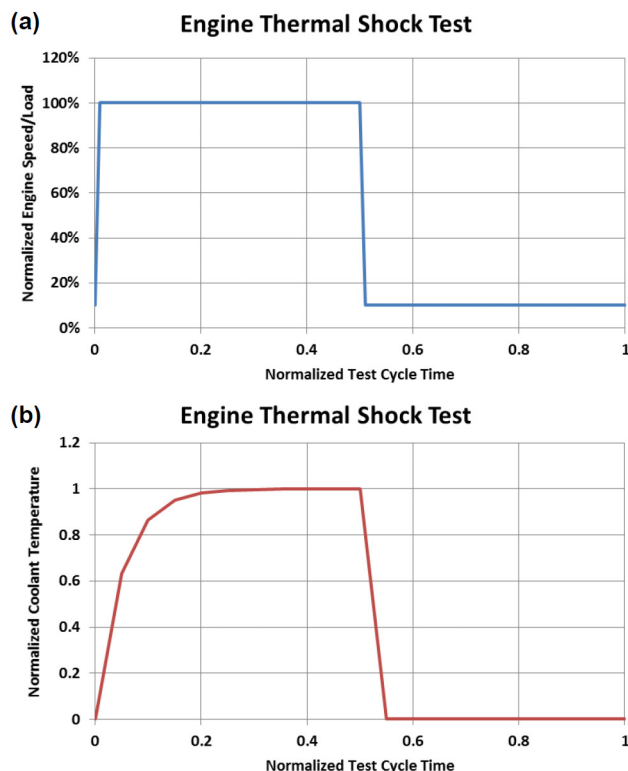


Figure 1. Plots of a) Normalized engine speed and load vs. time, b) Normalized coolant temperature vs. time for engine thermal shock test cycle.

Given the higher loading and strict requirements for durability and reliability of modern engine component designs, traditional testing of engine thermal loading under steady-state mode of operation may not mimic the severe fatigue prone conditions. Instead, test procedures with transient thermal loading cycles (temperature cycles as shown in Figure 1) can better capture thermal shock on cylinder heads and associated components. Such testing procedures are costly, and can

be minimized if predictive simulation techniques are utilized early in the engine development program. Engine cylinder head simulation studies that include effects of transient thermal loading on stress and fatigue are scarce in the literature and often include assumptions or empirical correlations to account for fluid side thermal boundary conditions. Thalmair et al. [8] compared cracks generated on aluminum cylinder heads under a thermal shock engine test with numerical thermo-mechanical fatigue (TMF) results. Leidenfrost et al. [9] have numerically investigated fatigue life in an engine test cycle prone to TMF and compared the results with tests. Their work was recently evolved by Metzger et al. [10] to combine TMF and high cycle fatigue (HCF) loading in order to simulate the microcrack growth on the cast-iron cylinder heads (using a virtual thermal shock test similar to Figure 1). Metzger et al. [10] showed that thermo-mechanical fatigue is the major contributor to lifetime. They reported discrepancies between the predicted lifetimes and the number of cycles when cracks were discovered although reasonably good prediction of the location of cracks was reported.

The aforementioned literature which focuses on valve bridge areas does not focus on the transient temperature calculation because the valve bridge area heats up somewhat uniformly, as we will show later in this paper. However, in other areas of the cylinder head, different sections heat up and cool down at different rates. This leads to significant temperature gradients in the head during the heat up and cool down phases. These temperature gradients lead to thermal stresses. It is important to note that the peak temperature gradients and peak thermal stresses may not occur during the steady state portions of a thermal shock test.

In the valve bridge area, the peak stress occurs at the steady state points. However, some other areas of the cylinder head do not have uniform rates of heating and cooling, and we will show for those areas that the transient mechanism described above is important. To our knowledge, the importance of the transient temperature calculation and how it leads to thermal stress peaks has not been reported in the literature for cylinder heads and blocks, though extensive documentation has been given for exhaust manifolds, see for example [11,12,13].

The main goal of the present work is to accurately predict life and stresses in an engine cylinder head and block under cyclic thermal loading conditions where thermo-mechanical fatigue is significant. In this paper, a six-cylinder heavy-duty diesel engine is modeled using the iterative decoupled process between transient IC and steady-state CHT simulations. Temperature results from the CHT model are compared against multiple thermocouple measurements. Final solid temperatures and thermal boundary conditions at gas and liquid coolant boundary surfaces are interpolated on a FEA grid and utilized to solve for a transient engine test cycle. Transient temperature variations at the same thermocouple locations, used to validate CFD-CHT results, are used to evaluate time-varying temperatures from the transient FEA solver. Finally, the stress and fatigue analyses are performed to examine and determine the critical areas where cyclic thermal loading can adversely affect engine durability and life.

COMPUTATIONAL ANALYSIS METHODOLOGY

In the following sections, computational methodologies are explained for each part of the analysis. In addition, assumptions and inputs are addressed.

Figure 2 presents a schematic of a comprehensive engine simulation procedure where the ultimate goal is to predict engine cylinder head and block life. The overall procedure followed in this work includes three main parts:

1. Calculation of solid temperatures as well as gas and coolant side thermal boundary conditions in the CFD solver;
2. Mapping the solid temperatures and surface boundary conditions from CFD solution to the FEA Thermal solver to perform a transient thermal analysis representing an engine test cycle;
3. Transferring selective temperatures during the thermal shock test to the Structural FEA solver to obtain stresses followed by fatigue life predictions in a fatigue analysis solver.

The first part is explained in the following section.

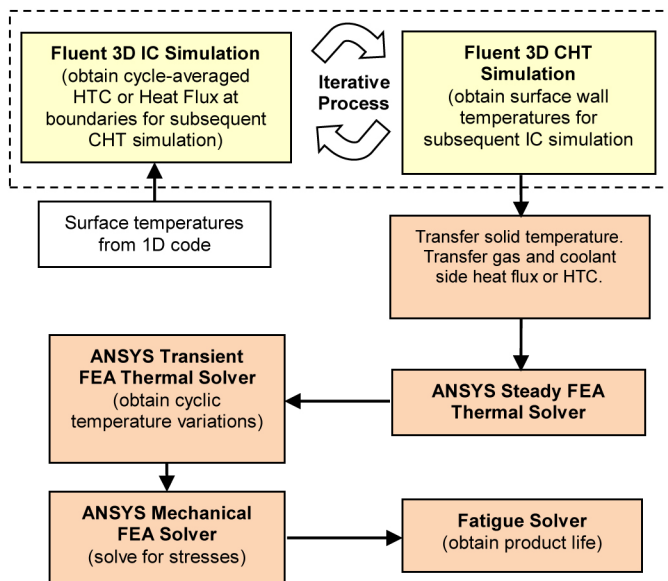


Figure 2. Schematic of the full engine thermal and mechanical simulations leading to engine life predictions. The iterative IC-CHT procedure is depicted on top in the dashed box.

Decoupled Iterative Process between CHT and IC Simulations

To calculate solid temperatures and find thermal boundary conditions, an iterative process is performed. Each solution “iteration” includes two steps:

1. Develop a transient IC model, solve for one or more engine cycles and obtain time-averaged (cycle-averaged) thermal boundary conditions including near wall temperature, heat transfer coefficient or surface heat flux;

2. Develop a Conjugate Heat Transfer (CHT) model of engine solid head and block in addition to coolant fluid in cooling jackets; assign gas side cycle-averaged boundary conditions from the IC model and obtain wall temperatures to use in any subsequent IC simulations (i.e., subsequent IC-CHT iterations).

This procedure has been utilized before [2,5,6] in analyses where gas side time scales are highly different from those of solid side resulting in non-practical full transient analyses for both gas flows and solid conduction in a common model. Iterations between IC and CHT models are performed until temperature is converged; i.e., temperature changes from one iteration to the next fall below a desired threshold. Commercial CFD solver ANSYS-Fluent is utilized for both IC and CHT simulations. Details of the IC and CHT modeling are discussed in the following sections.

In-Cylinder (IC) Simulations

To obtain thermal boundary conditions for the subsequent CHT and FEA Thermal analyses, a transient in-cylinder simulation is performed. The diesel engine in the current study includes six cylinders. Operating conditions of all cylinders are similar and it will be more practical to model only one cylinder whereas a more rigorous study can benefit from modeling six IC simulations, i.e., one simulation per each cylinder, considering phase differences between cylinders and the firing order.

Figure 3 shows a schematic of the diesel engine modeled in this study. The model is set up in ANSYS-Fluent ICE Module. The motion of valves and piston is handled by a moving-deforming mesh strategy that includes: 1) Mesh Layering: When the piston or valves are moving down, prism or hexahedral elements adjacent to moving surface get stretched. The stretched cell is divided to two cells, when the cell height is stretched beyond a threshold; similarly when piston or valves move up and the cell adjacent to the moving surfaces is compressed, it is collapsed and merged with the neighboring cell based on a collapse factor. 2) Smoothing (morphing): Cells are stretched or compressed without adding or removing a new layer of cells. Mesh smoothing is done for small movements of geometry surfaces.

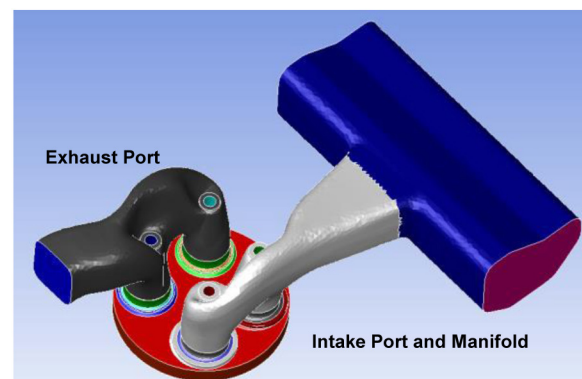


Figure 3. Schematic of the diesel engine in-cylinder (IC) model. Part of the intake manifold is included in the engine model to obtain boundary conditions on manifold.

The ultimate goal of this study is to capture solid temperatures as well as gas and liquid coolant side boundary conditions for a transient FEA thermal analysis. It is not intended to capture spray pattern/penetration or ignition behavior to optimize engine performance or predict emissions. As a result, instead of solving for reacting flow and spray, combustion heat release is considered as a source term in the energy equation. Heat release rate [J/s] is calculated from calibrated GT-Power cylinder pressure data. A user-defined subroutine is utilized to incorporate heat release as a source term at each time within the simulation. It should be noted that an IC simulation with spray modeling and combustion is required if localized heating effects (dependent on number of injector holes and injector orientation and pressure) need to be tracked. Such localized heating pockets are not of significance in the current analysis procedure.

Constant temperature thermal boundary conditions are applied at all IC model wall surfaces. In the first IC iteration, temperature data from GT-Power outputs are used at port, valve insert, valve, firedeck, liner and piston surfaces. At any additional IC iteration, a combination of spatially averaged temperatures and spatially varying temperature profiles from the CHT model are applied at various component surfaces.

Two engine cycles are modeled for the first CHT-IC iteration and one engine cycle is modeled for each subsequent iteration. In the second IC simulation (iteration), X, Y, Z velocities, temperatures and turbulence quantities from the end of cycle 2 are utilized to continue with one more engine cycle. IC simulations start from a few crank angles before IVO.

Cycle-averaged gas side thermal boundary conditions are utilized in the CHT and FEA Thermal models. In order to capture cycle-averaged boundary conditions, the built-in time averaging module in the Fluent code is utilized. Wall-function heat transfer coefficients, total temperature, and total surface heat flux are time averaged. Time averaging is activated only in the final cycle of each CHT-IC iteration (e.g., second cycle in the first iteration). Upon completion of the simulation, cycle-averaged data will be available for post-processing and export to CHT or FEA models. Because the cylinder liner area is deforming by time and new grid elements are added or removed as piston moves, the built-in time averaging cannot be used on liners. A user-defined subroutine is used to obtain time-averaged data on n pre-specified sections on the liner. Data on each liner section will be spatially averaged. Cycle-averaged heat transfer coefficient and near wall temperatures output from the subroutine will be applied on each liner section of the CHT and FEA models.

The IC and CHT simulations are carried out at two operating conditions; a “hot” condition representing high thermal loading (shown as the 100% load condition in Figure 1a) and a “cold” condition representing low thermal loading (shown as the 10% loading condition in Figure 1a).

Conjugate Heat Transfer Simulations

In the CHT model, energy equation is solved for conduction at solid regions while momentum, mass and energy equations are solved in the fluid regions. Figure 4 presents a schematic of the CHT model.

The solid region in the CHT model consists of cylinder head, head bolts, intake and exhaust valve inserts, valves and valve guides, cylinder head gasket, crankcase, and liners. The fluid region includes two cooling jackets, one for the head and the other for the crankcase and liners.

The computational domain has nearly 12 million polyhedral elements. The model is solved in ANSYS Fluent. A coupled pressure-based solver is utilized. Turbulence is modeled by a two-equation realizable k- ϵ model. Near wall treatment is conducted by a scalable wall function [7] that caters for wide ranges of wall Y^+ in the cooling jackets. First order upwind discretization schemes are used for momentum, turbulent kinetic energy and turbulent dissipation rate. Second order upwind discretization scheme is utilized for the energy equation.

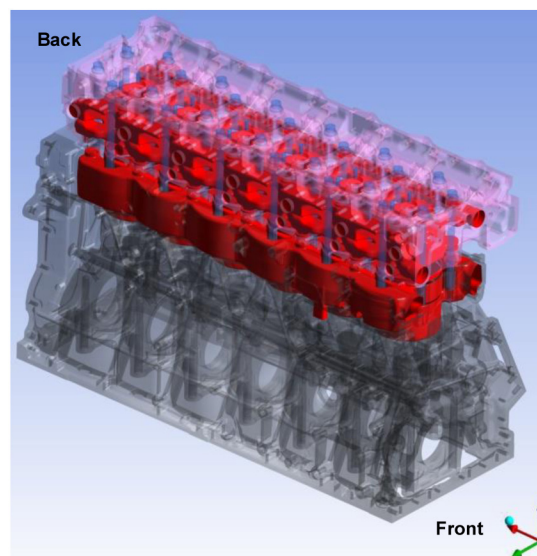


Figure 4. Schematic of engine CHT model. Only cylinder head, crankcase, cooling jackets, exhaust valves and head bolts are shown in this figure.

The convergence of the CHT model is evaluated by 1) balance of heat and mass; 2) conservative iteration residuals (e.g., as low as $1e-9$ for energy equation); 3) tracking temperatures to get constant. Additionally, desired flow distribution must be established at water jacket outlets. The CHT model converges in ~ 1000 iterations.

Cycle-averaged spatially-varying heat transfer coefficients and near wall temperatures from the IC model results are applied on exhaust port surfaces. The wall-function heat transfer coefficient is calculated using the following formulation

$$h_{wall} = \frac{\rho C_p C_\mu^{1/4} k_p^{1/2}}{T^*} \quad (1)$$

here ρ is the density, C_p is the specific heat, k_p is the turbulence kinetic energy at point P, T^* is the dimensionless law-of-the-wall temperature, and C_μ is a constant; see more details in [7].