

Transient Thermal Modeling of Power Train Components

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ABSTRACT

This paper discusses simplified lumped parameter thermal modeling of power train components. In particular, it discusses the tradeoff between model complexity and the ability to correlate the predicted temperatures and flow rates with measured data. The benefits and problems associated with using a three lumped mass model are explained and the value of this simpler model is promoted. The process for correlation and optimization using modern software tools is explained. Examples of models for engines and transmissions are illustrated along with their predictive abilities over typical driving cycles.

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INTRODUCTION

Increasingly rigorous fuel economy standards have brought about the need to better predict power train temperatures, heat, and work flows during transient conditions. Indeed, considerable fuel economy and emissions improvements can be realized by improving the cold start performance of vehicles. The study of this topic in the past twenty years has been extensive [1,2].

Design of sophisticated engine control systems for coolant, oil, auxiliaries, and features like exhaust gas recirculation, involves simulation and testing of various options. Conducting useful tests can require considerable investment in equipment, time, and engine modification[3]. Mathematical models can be of considerable assistance in this effort. Indeed, most of the computational tools required for such modeling are now common place [4].

All computational models require information about the system being modeled. Power train simulations need masses, fluid flows, friction power, combustion heat release, heat exchanger models and various heat transfer coefficients. Since simple models lump many factors together, a

correlation activity becomes a major part of modeling. This is one of the topics we wish to explore in this paper.

Creating mathematical models from which power train transient thermal performance can be predicted with fidelity poses some challenges

First, the majority of the data we have on engine power, heat rejection, engine friction, and the like, is available at steady state conditions when the engine is already warm.

Second, most transient experimental procedures are designed for verification of designs, not support of mathematical models. We have standard tests for emissions and fuel economy over various cycles. But the required measured parameters do not include many of the measures we need to support and correlate power train component models.

Third, transient power train thermal modeling requires expertise in a wide variety of subjects that few individual engineers have. Thus a collaborative effort employing several different modeling tools and methods is needed.

The model discussed in this paper divides the major power train elements into the minimum number of parts required to accomplish the desired task. It then draws upon a

wide range of references to assemble the relations needed for engine friction division between parts, oil flow divisions, convection heat transfer to the ambient, etc. As we shall illustrate below, the number of unknowns in such models can easily exceed the number of equations and the number of measureable parameters. That means one must either design and conduct many special tests or have the wisdom to extract useful sub-models from the literature. A number of references illustrate similar modeling techniques[5,6] and show that considerable experience is needed for successful execution of simple models.

The approach in this paper uses a commercial thermo-fluid analysis software package to run the thermo-fluids analysis. Other software is linked to provide such information as transmission shift schedules, and to translate power train temperatures and operating conditions into fuel consumption. We do not deal with these other linked routines in this discussion.

The commercial package considers the effects of short pipes, component interaction in internal flows, and various heat transfers. We update this with some additional information to improve the internal flow modeling[7]. We also need to consider the effects of free stream turbulence on convection over objects [8,9] such as transmission casings, and the characteristic of real flow over blunt objects [10]. Thus, there is a coefficient that needs to be “adjusted” for every standard heat transfer relation one inserts.

The process of model development involves formulation and correlation with actual power train performance. We illustrate this using the engine model. Then we describe some transmission models that admit to the same correlation methodology.

One of the additional objectives of this work was to use these models to quantify trends of engine warm-up and point to rules of thumb for predictive modeling on future platforms.

ENGINE MODEL EXAMPLE

OVERVIEW

We consider an engine transient model whose objective is to assist in the prediction of fuel consumption during engine warmup, and in the design of cooling and oil system thermal control. A separate model determines engine required shaft power and RPM during any given drive cycle. An algorithm for fuel consumption based on engine calibration was used which will not be discussed in this paper. The engine model discussed here is designed to predict engine coolant, engine oil, and, coupled to another model, transmission oil temperatures.

The engine is only one part of the system of components that determines the vehicle performance and the associated fuel consumption. The overall system model includes the controls, external fluid circuits for cooling, engine oil, and transmission oil along with associated heat exchanger models.

The simplest engine model is a single lumped mass. However, because of the significant difference between coolant and oil temperatures during engine warm up and their effects on engine friction and coolant temperature, we need to use more than one thermal mass.

We divide the engine into the fewest number of parts possible to accomplish the task. In this case we select three “lumped” masses as the block, crank, and sump. The “block” contains all the coolant passages and, assuming well mixed coolant, the coolant outlet temperature equals the block average temperature. The “crank” consists of the crankshaft, pistons, connecting rods, main bearings, and supporting structure. The “sump” is the oil reservoir and oil pan structure. These are shown in Figure (1).

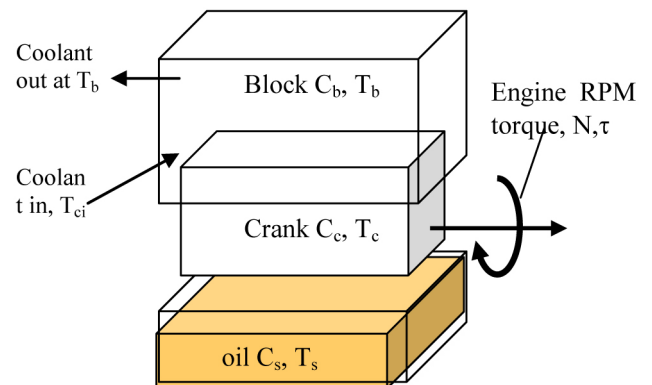


FIG. 1a. Three Mass Engine Model

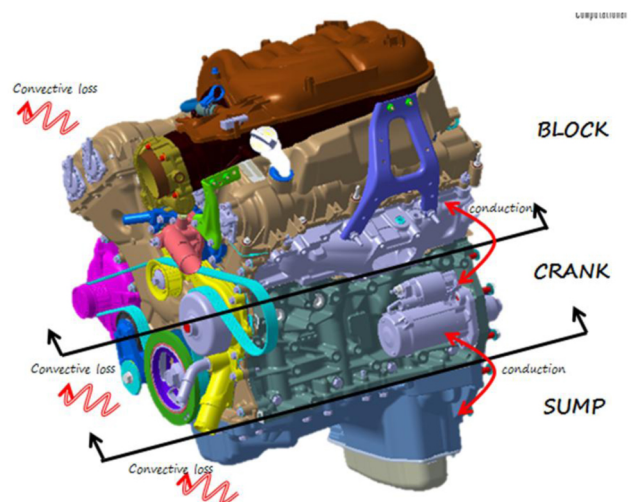


FIG. 1b. General Engine Division Into Parts

The division between the block and the crank is taken at the bottom of the coolant passages. Thus there is both oil and coolant flow to the “block” but no coolant flow in the “crank”. The crank-sump division is at the oil pan gasket.

Figure (2) shows a 1D lumped parameter model diagram for the three masses. A Flowmaster® model manages the coolant and oil flow rates as a known function of engine

speed, load, and coolant temperature. We assume the required engine brake power and speed are known from a prescribed vehicle cycle combined with the models of drive train components. The cylinder head and some part of the exhaust manifold are lumped with the “block” mass. We know that the exhaust manifold will be at a significantly different temperature. So this is one of many distortions of the real situation which we make.

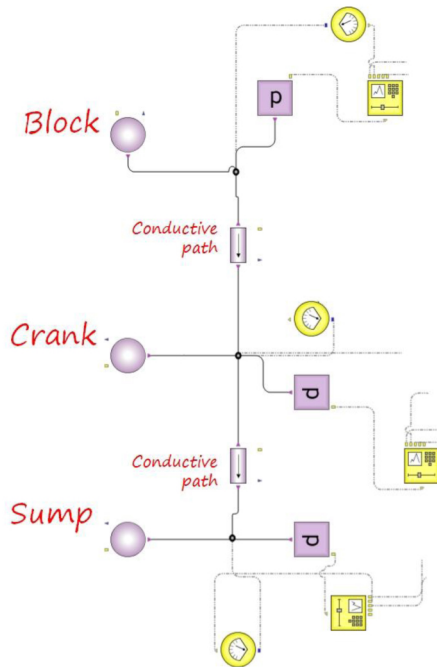


FIG. 2. Associated Lumped Parameter Model

Given the large variation in temperature within the single lumped mass of the block, the authors are conscious that this violates the Biot number requirement ($Bi < 0.1$). We can compensate for this partially by adding a conduction term to the surface convection loss as one will see in [equation \(9\)](#) below. But still, there is a trade-off required between the complexity of the model and the amount of data required to make the model work. Each added correlation for various heat flows introduces one or more additional “coefficients” which must be estimated or determined by experiment. So the fewer thermal masses one can get away with, the easier is the modeling task. As we shall show, the three mass model illustrated here does a reasonable job for the task at hand. If we wanted to predict emissions or exhaust gas temperature, we would divide the “block” into more parts. The model by Kaplan[5] is of this type as well as Bohac[11] and others.

ENERGY FLOWS IN THE ENGINE

Before writing the equations for conservation of energy for the three masses, we present our view of the energy flows within the engine. [Figure \(3\)](#) illustrates the major energy paths.

The dashed line shows the conventional boundary used to define the system for the steady state energy balance used in

evaluating heat loss to coolant, oil, and convection/radiation. Thus for the engine as a whole, energy enters with the fuel and air and leaves with the exhaust gas, brake power, and heat rejected to the coolant, oil, and surrounding ambient. At steady state, the energy flows must balance.

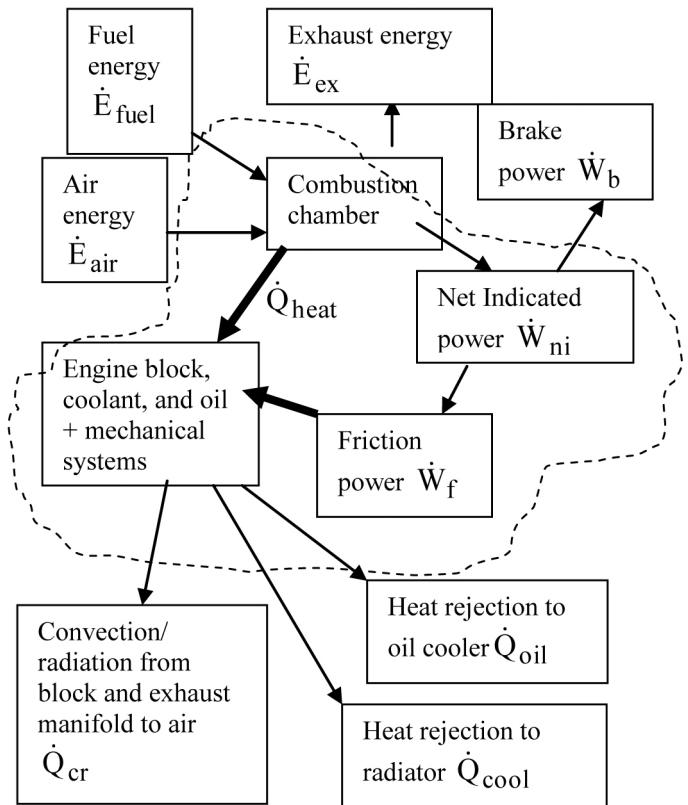


FIG. 3. Energy Flows in the Engine

For a transient analysis, we require information about what takes place inside the dashed boundary.

An indicator diagram for the engine will provide the gross indicated mechanical power released by the combustion process. This minus the pumping work is the net indicated mechanical power that acts on the piston face shown as $\dot{W}_{ni}$ in the diagram. The pumping work is converted to heat. This plus the heat from the combustion is transferred to the cylinder walls, head, and piston and enters the engine metal. Because modern engine testing includes an indicator diagram, we can know the net mechanical work on the piston, $\dot{W}_{ni}$. From this and the fuel, air, and exhaust state we can determine the heat energy that enters the engine from the energy balance on the combustion chamber:

$$\dot{Q}_{heat} = \dot{E}_{air} + \dot{E}_{fuel} - \dot{E}_{ex} - \dot{W}_{ni} \quad (1)$$

The availability of direct measured indicated power and pumping work means we can determine the friction power on the directly by subtraction:

$$\dot{W}_f = \dot{W}_{ni} - \dot{W}_b \quad (2)$$

This is a significant improvement over the past when one had to estimate friction from motoring data and various models.

We can determine Q_{heat} and W_f from steady state tests on the engine. We then correlate these with engine operating parameters. For any required brake power and RPM, we can then find Q_{heat} and W_f for use in the transient model. Various corrections for cold start conditions can be applied from the literature [12]

Having clarified the general energy flows, we turn to the division of these flows between the three thermal masses.

We take the friction work to be divided between the crank, which includes the pistons, and the “block” which includes the valve train. We do not separate the cylinder head out as a separate thermal mass in this model.

The total friction work must thus be divided in two parts. So we let f_v be the fraction of the total mechanical friction work that is assigned to the “block” mass.

requires another fraction f_h which is the fraction of Q_{heat} that goes to the block.

The energy transfer between the block, crank, and sump takes place through the conduction paths and the oil flow.

These divisions of oil, friction, and heat can be estimated from many reported studies in the literature [12,13,14] as well as by tests on components or by correlation with the overall model

We model the conduction between the masses as linear functions of the temperature differences using an overall heat transfer coefficient $\{UA\}_{bc}$ between the block and crank and $\{UA\}_{cs}$ between the crank and the sump.

Figure (4) shows these energy flows with the three masses highlighted by the darker boundaries. Each mass has some thermal capacity “C” equal to the mass times specific heat. We have also added linear heat loss from each mass to the surrounding air at T_a . The oil and coolant flows are identified as m_o and m_c

TRANSIENT ENERGY EQUATIONS

The three transient conservation of energy equations for the engine are

Block mass:

$$C_b \frac{dT_b}{dt} = \text{Heat from combustion} + \text{valve train friction work} \\ - \text{heat taken away by crank oil} - \text{conduction to crank} \\ - \text{convection to surrounding air} - \text{energy to coolant}$$

$$C_b \frac{dT_b}{dt} = f_h \dot{Q}_{\text{heat}} + f_v \dot{W}_f - f_b \dot{m}_o c_{po} [T_b - T_c] \\ - \{UA\}_{bc} [T_b - T_c] - \{UA\}_{ba} [T_b - T_a] \\ - \dot{m}_c c_{pc} [T_b - T_{ci}] \quad (3)$$

Crank mass:

$$C_c \frac{dT_c}{dt} = \text{Combustion heat} + \text{friction work} + \text{block conduction} \\ + \text{heat loss in oil to block} + \text{heat gain from sump oil} \\ - \text{conduction to sump} - \text{convection to surrounding air}$$

$$C_c \frac{dT_c}{dt} = [1 - f_h] \dot{Q}_{\text{heat}} + [1 - f_v] \dot{W}_f \\ + \{UA\}_{bc} [T_b - T_c] - f_b \dot{m}_o c_{po} [T_c - T_b] \\ + \dot{m}_o c_{po} [T_s - T_c] - \{UA\}_{cs} [T_c - T_s] \\ - \{UA\}_{ca} [T_c - T_a]$$

(4)

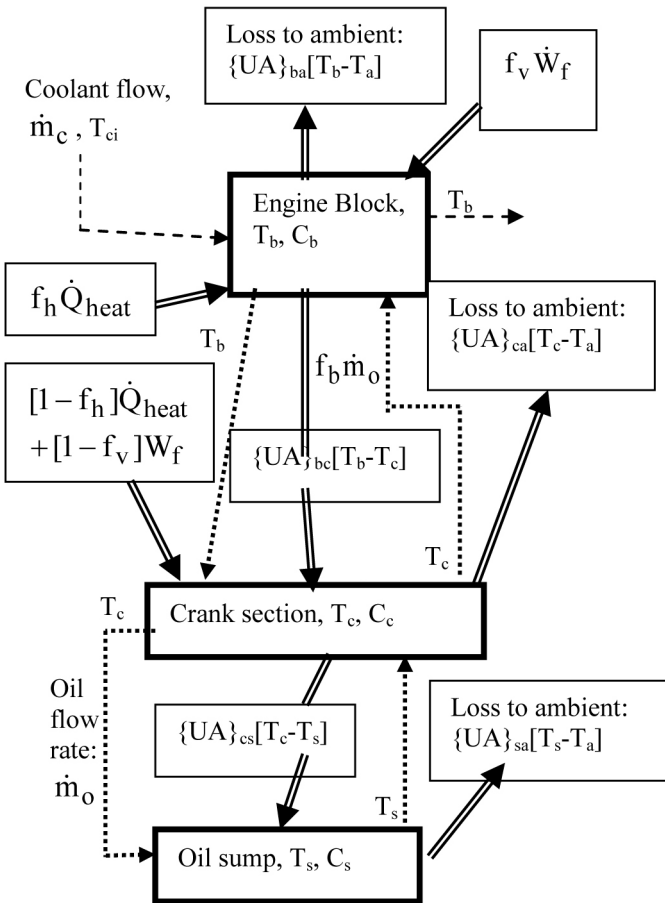


FIG. 4. Energy and Fluid Flows Between Engine Masses

The heat leaving the combustion process goes to the block and the piston. The piston being part of the “crank” mass