The development of high-fidelity predictive models of vehicle engines is a major preoccupation of powertrain engineers. By developing virtual prototypes of their engine designs, automotive manufacturers can obtain tremendous insight into the behavior of the engine. This insight is particularly valuable during controller design and development, to maximize vehicle performance while complying with governmental and ecological constraints. Doing this analysis before investing in the physical prototyping stages has been proven to save significant time and substantially reduce costs during the product development process.

This article describes the development of a mean-value model of an internal combustion engine using MapleSim, from the development of a parameterized model using a variety of physical modeling techniques to the final simulation. Mean-value models provide the overall energy consumption/production balance without considering the details of the intake/compression/ignition/exhaust cycles. These models are particularly favored by engine control developers because they deal only with the properties of the system the developers are interested in, and the models are faster to compute.

Note that because this model was created using proprietary customer data, the actual values have been replaced with data from public sources.
The throttle subsystem calculates the air mass flow based on the throttle valve angle. For the purpose of the model, the angle is provided by the engine controller as an input signal. Based on the geometry of the throttle and valve, the effective throttle area \( A_{th} \) is related to the valve angle \( \phi \) as follows:

\[
A_{th} = \frac{dD}{2} \left[ 1 - \left( \frac{\cos \phi}{\cos \phi_0} \right) \right]^{\frac{1}{2}} + \frac{D^2}{2} \left\{ \arcsin \left( \frac{1}{\cos \phi} \right) \right\}^{\frac{1}{2}} - \frac{D^2}{2 \cos \phi} \arcsin \left( \frac{1}{\cos \phi_0} \right)
\]

(Moskwa, 1988. Note all variable definitions are given in the glossary at the end of this article.)

The mass flow rate itself is related to the effective throttle area and manifold pressure as follows:

\[
\begin{align*}
\dot{m}_{\text{thr}} &= \frac{C_{D_0}}{\sqrt{RT_0}} \frac{A_{th}}{P_0} \left( \frac{P_m}{P_0} \right)^{\frac{1}{2}} \left[ 2 \gamma \left( 1 - \frac{P_m}{P_0} \right)^{\frac{\gamma - 1}{\gamma}} \right]^{\frac{1}{2}} \quad \text{if} \quad \frac{P_m}{P_0} > \left[ \frac{2}{\gamma + 1} \right]^{\frac{\gamma - 1}{\gamma}} \\
\dot{m}_{\text{thr}} &= \frac{C_{D_0}}{\sqrt{RT_0}} \frac{A_{th}}{P_0} \gamma \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma}} \quad \text{if} \quad \frac{P_m}{P_0} \leq \left[ \frac{2}{\gamma + 1} \right]^{\frac{\gamma - 1}{\gamma}}
\end{align*}
\]

(Heywood, 1988)

There are two parts to the relationship: subsonic and supersonic, determined by the ratio between the manifold pressure and ambient pressure at the entrance to the throttle.
These equations were implemented as custom components in MapleSim by simply entering the equations into a Maple document template that contains the necessary tools for converting them into a MapleSim block, with the necessary inputs, outputs, and parameters. The equations were implemented as a piecewise function to switch to the choking equation when the supersonic flow condition is exceeded. Note that the mass flow block requires two inputs: the throttle area ($A_{th}$) and the manifold pressure ($P_m$), which is a property fed back from the manifold subsystem. The output signal is the resulting air/fuel mass flow ($\dot{m}_{in}$) through the throttle.

In MapleSim, you can review the Maple document that was used to create the custom components by simply double-clicking on a custom component. The document for computing the effective throttle area is shown in Figure 3.

The Manifold Subsystem

The intake manifold has a significant effect on the gas flow and pressure to the engine cylinders. There are two physical effects that need to be included:

1. Manifold Pressure

Physically, the intake manifold guides the air/fuel mixture into the cylinders but also causes an obstruction that reduces the pressure, and hence the mass flow, at the cylinders. The calculation of the manifold pressure ($P_m$) and mass flow ($\dot{m}_i$) is based on the ideal gas equation in derivative form by considering isothermal conditions in the intake manifold control volume (Hendricks, E., et al., 1996).

$$\dot{P}_m = \frac{RT_m}{V_m} \left( \dot{m}_{in} - \dot{m}_{e} \right) \quad \text{and} \quad \dot{m}_i = \frac{\eta_{vol} V_i n}{60 N_{exp} R T_m} P_m$$

These include the volumetric efficiency ($\eta_{vol}$) which is described below.

Again, these equations, including the differential equation for the manifold pressure, can simply be entered as they appear above into the Maple custom component document. The only additional element required is an initial value for the pressure. This can be entered as a parameter in the custom component block (set at slightly below atmospheric pressure).
Figure 5: Manifold pressure and mass flow equations implemented in Maple as a custom component

The inputs to the block are the engine speed (RPM), mass flow from the throttle, and volumetric efficiency from the lookup table. The outputs are the manifold pressure and mass flow going into the cylinders.

2. Volumetric Efficiency
This is modeled as a lookup table of experimental values of volumetric efficiency ($\eta_{vol}$) over a range of engine speeds and manifold pressures, stored in an Excel® spreadsheet. The actual data values shown here have been changed for confidentiality reasons, but they give a reasonably good approximation to demonstrate the principles of its use.

Figure 6: Volumetric efficiency lookup table
The Engine Subsystem

The engine subsystem considers the power generated through the combustion of the fuel in the gas mixture delivered to the cylinders, and accounts for thermal efficiency, friction and inertial losses in the engine and the inertial load at the drive shaft.

The mean engine power subsystem contains the calculations for the thermal efficiency, indicated power generated from the combustion of the fuel, and lost power due to mechanical and pumping effects in the engine. The brake power – the indicated power without the lost power – is fed out to the engine subsystem. In Figure 7, the net power is computed at the interface between the output torque computation (in the output torque subsystem) and the mechanical flange port at the far right.

1. Thermal Efficiency

The thermal efficiency ($\eta_{th}$) is modeled using an empirically derived polynomial expression (Hendricks, E., et al., 1996), accounting for speed ($n$), manifold pressure ($P_m$), and air-fuel ratio ($\lambda$):

$$\eta_{th} = \eta_{thm} \eta_{thp_m} \eta_{th\lambda}$$

where

$$\eta_{thm} = b_1 (1 - b_2 n^{b_3})$$

$$\eta_{thp_m} = c_1 + c_2 P_m + c_3 P_m^2$$

$$\eta_{th\lambda} = \begin{cases} d_1 + d_2 \lambda + d_3 \lambda^2 & \text{if } \lambda > 1 \\ d_4 + d_5 \lambda + d_6 \lambda^2 & \text{if } \lambda < 1 \end{cases}$$

The coefficients $b_1$, $c_1$, and $d_i$ are typically generated from manufacturers’ testing data and are considered highly proprietary. For the purpose of this example, the coefficients given in Hendricks were used.

A custom component was created using these equations, with air/fuel ratio ($\lambda$), engine speed ($n$), and manifold pressure ($P_m$) as inputs, returning thermal efficiency ($\eta_{th}$) as the output.
2. Engine Power Calculation

Engine power and speed calculations are based on the following engine equations (Hendricks, E., et al., 1996):

\[ \dot{P}_{\text{net}} = \dot{P}_{\text{in}} - \dot{P}_{\text{loss}} - \dot{P}_{\text{load}} \quad \text{and} \quad \dot{n} = \frac{1}{L_c n} \left[ \dot{P}_{\text{net}} - \dot{P}_{\text{loss}} - \dot{P}_{\text{load}} \right] \]

where

\[ \dot{P}_{\text{in}} = \hat{\dot{n}}_j H_j \eta_{\dot{n}} \]
\[ \dot{P}_{\text{loss}} = n \left[ a_1 + a_2 n + a_3 n^2 \right] + \left[ a_4 + a_5 n \right] \eta_{\text{in}} \]
\[ \dot{P}_{\text{load}} = k_i n^2 \]
\[ \hat{\dot{n}}_j = \frac{\dot{\dot{n}}}{\dot{\dot{L}}_j} \]

and where \( a_1, a_2, a_3, a_4, a_5 \) and \( k_i \) are all constants obtained by experiment. (Textbook values were used in this example.)

A custom component was created based on these equations to calculate the net power. However, since the model needs to accommodate external loading, it was desirable to replace the empirical approximation for the load power (\( \dot{P}_{\text{load}} \)) and the differential equation for the engine speed (\( n \)) with the computed load power and speed from the drive shaft.

Load and Net Power Computation

Up to this point in the project, the model has been developed as a signal-flow representation, with engine properties being transported from equation to equation (in custom components) using signal lines. However, one of the strengths of MapleSim is that engineering systems can be represented with connected components where the transfer of properties is implicit in the connections. This “acausal” approach makes it much easier to produce engineering models and has been used to connect the external load from a dynamometer model to the engine model. This means that the load power is implicitly determined and, hence, the engine speed can be computed simply by connecting a speed sensor block to the line that represents the drive shaft.

In effect, this implements the speed equation from above:

\[ \dot{n} = \frac{1}{L \cdot \dot{n}} \left[ \dot{P}_{\text{net}} \right] \quad \text{where} \quad \dot{P}_{\text{net}} = \dot{P}_{\text{brake}} - \dot{P}_{\text{load}} = \dot{P}_{\text{in}} - \dot{P}_{\text{loss}} - \dot{P}_{\text{load}} \]
The main challenge was interfacing the signal-flow model with the acausal mechanical model. This was achieved by computing the brake torque \(((P_{\text{in}} - P_{\text{out}})/n)\) from the engine and converting this to an acausal torque, applied to an inertia block that represents the internal engine inertia, then to a mechanical rotational flange for connection to the external load (see Figure 10).

In Figure 9, the output torque subsystem is connected to the mechanical flange which represents the drive shaft from the engine. Inserted between the two flanges is a power sensor that provides the net power \((P_{\text{brake}} - P_{\text{load}})\) and a speed sensor that provides the engine speed as a signal that is sent back to the engine power block to fulfill the requirements of the power equations.

**External Loading**

*(Dynamometer)*

The engine power and speed are now represented at the drive shaft as available torque, which allows a mechanical load to be attached via a mechanical flange. This acausal connection makes the addition of loads to the system significantly easier than with signal-flow models. In future phases of this project, the transmission and drivetrain models can be very easily connected to the engine in this way.

For the purpose of this phase of the project, a simple dynamometer model is used to apply the external load to the engine (see Figure 11).

The dynamometer model includes rotational inertia to represent the drivetrain loading, vehicle mass for translational inertia and viscous damping to represent drag loading. It also includes the ability to apply arbitrary loading forces from an external signal. During a simulation, this ability will be used to apply a step load increase to the engine at \(t=20\) seconds in order to observe the response.
Speed Control

The engine speed control subsystem is not intended to represent a real Engine Control Unit (ECU). It is simply provided to represent the driver and the overall control law to stabilize the engine response to a desired speed. The driver’s desired speed is set using a step function into a limited PID control block that sets the required valve angle in the throttle, given the difference between the desired and actual speeds.

The limits in the PID block are used to ensure the valve does not exceed the physical limitations of the valve mechanism in the throttle. For this model, the limits are set to 8 and 78 degrees but will vary from engine to engine.

Another advantage of using the PID block to define the limits (instead of, say, a Limit block) is that it includes an anti-windup facility that not only keeps the output at its limit but also stops integrating. This means that when the output returns to within the limits, the integrator does not have to “wind down” from its current state before sending out the angle value, thus sending out the correct value immediately.

Model Parameterization

One of the goals of this project was to provide a general framework that can be modified for different IC engines by adjusting the design parameters and lookup tables. To that end, the model is fully parameterized, allowing the user to adjust values in one place: the engine parameter block.

Parameter blocks are a feature that allows you to include all the parameters in one place. When you click on a parameter block you can view and edit the parameters in the parameter inspector in the MapleSim user interface (see Figure 12). You can also build libraries of parameter blocks for different engines and then implement them simply by dragging and replacing the existing engine parameter block with the new one.

In this way, you can investigate a range of many engine models with the same basic MapleSim model.
**Simulation and Results**
For this phase of the project, a simple test cycle was set up:

**Total duration**: 30 s

**Engine Speed**: Idle (1000 RPM), then step up to 5000 RPM at t = 2 s

**External Load**: 200 N, then step up to 3000 N at t = 20 s

During the simulation, several properties were monitored and plotted. For the purpose of this discussion, only the throttle valve angle, available power, speed, loading and fuel consumption will be considered.

Notice in the results that at t = 2 s, the setpoint speed increases to 5000 RPM and the throttle valve opens but reaches the maximum angle. As the engine speeds up from 1000 RPM to 5000 RPM, the valve angle decreases to a steady angle (~15 deg) to provide enough gas flow to maintain the new speed.

At t=20, the external load increases to 3000 N, the throttle valve opens to provide more power, and the speed drops slightly before settling back to 5000 RPM.

Not surprisingly, the fuel consumption shows an increased rate as the engine increases in speed and then again when the external load increases.
Further Work
This model is Phase 1 of an ongoing project to produce a realistic, parameterized mean-value model for a range of internal combustion engines. Based on feedback from industrial experts, a growing list of enhancements will be made to the model as the project progresses. These include variation of air/fuel ratio (currently assumed constant), effects of ignition and variable valve timing (VVT), as well as the addition of components such as turbo-chargers and catalytic converters.

Work is already underway to convert this model to real-time C code for implementation in HIL testing platforms such as dSPACE® and National Instruments™ LabVIEW™ Real-Time and NI VeriStand™. The next phase of the engine model will include transmission and drivetrain models so that it can be used with published driving cycles; this will allow the model to be fully validated against other engine models and real engine test data.

Conclusion
Phase 1 of this project uses MapleSim and Maple to implement published engine modeling equations and empirical models. This is done through the use of custom components and lookup tables that can be easily connected by signal lines that represent the engine properties. This signal-flow representation is readily complemented by the use of “acausal” model components for the mechanical systems.

The results from the simulation are based on engine parameters from a mixture of public-domain sources, so while we can give an intuitive sense of the “correctness” of the model, we can only validate the results with our customers with their own design data. While we cannot publish these results due to confidentiality reasons, we can report that the correlations so far are very encouraging.

As the project progresses, we will see further details being added to enhance fidelity and extend the scope of the model, especially into real-time testing applications with HIL.

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References


Hendricks, E., Sorenson, S. C., Mean Value Modeling of Spark Ignition Engines, SAE 900616


Glossary of Variable Names
The following variable names were used in the equations in this article, in order of appearance:

- \(d\) = throttle pin diameter (m)
- \(D\) = throttle bore diameter (m)
- \(\varphi_{cp}\) = throttle plate closed angle (deg)
- \(\varphi\) = throttle plate angle (deg)
- \(a = \frac{d}{D}\), diameter ratio
- \(A_{thr}\) = effective throttle area (m²)
- \(R\) = gas constant (287 kJ/kg)
- \(C_{D_{th}}\) = throttle discharge coefficient
- \(T_0\) = ambient temperature (278.15 K)
- \(\gamma\) = specific heat ratio (1.4)
- \(P_o\) = ambient pressure (101.325 kPa = 1 bar)
- \(P_m\) = manifold pressure (kPa)
- \(\dot{m}_{thr}\) = throttle mass flow rate (kg/s)
- \(T_m\) = manifold temperature, (K engine is considered to be working at steady state condition)
- \(V_m\) = manifold volume (m³)
- \(n\) = engine speed (RPM)
- \(\eta_{vol}\) = volumetric efficiency
- \(\dot{m}_c\) = mass flow rate into the cylinders (kg/s)
- \(N_{col}\) = number of cylinders
- \(N_{eng}\) = engine type number: 2 for four-stroke engines and 1 for two-stroke engines.
- \(V_d\) = displaced volume in cylinder:
  \[V_d = S \cdot \left( \frac{\pi \cdot B^2}{4} \right)\]
  where
  - \(S\) = stroke (m)
  - \(B\) = bore diameter (m)
- \(\dot{H}_{fuel}\) = fuel heating value for gasoline (46000 kJ/kg)
- \(\lambda\) = air/fuel ratio
- \(I_{sto}\) = stoichiometric normalization factor (14.67)
- \(I_c\) = engine inertia
- \(\eta_{th}\) = thermal efficiency
- \(P_{ind}\) = indicated power (kW)
- \(P_{load}\) = load power (kW)
- \(P_{loss}\) = loss power (kW)
- \(\dot{m}_f\) = fuel flow rate (kg/s)