

Full-Vehicle Model Development for Prediction of Fuel Consumption

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ABSTRACT

A predictive model of a specific vehicle was modeled in the system-level physical modeling tool, MapleSim, for performance and fuel consumption prediction of a full vehicle powertrain, driving a multi-body chassis model with tire models. The project also includes investigation into overall fuel efficiency and effect on vehicle handling for different drive cycles.

The goals of this project were to investigate: 1) the relationships between the forces at tire/road interfaces during various drive cycles and the fuel efficiency of a vehicle, and 2) the interaction between the powertrain and the chassis of the vehicle. To accomplish these goals, a complete vehicle model was created in the lumped-parameter physical modeling tool, MapleSim. A great deal of effort has gone into using real parameters and to assure that some mathematical rigour has been employed in its development.

This paper presents the development of the model and results, covering: Powertrain (engine, transmission), Chassis (longitudinal, 3D multibody), Steering, and Wheels (tires, brakes). A simple Driver model implementing accelerator/brakes and steering angle was also created, along with some tests to validate the behaviour of the model. Finally, tests are created and executed to elucidate the various relationships under investigation. The results are presented and analyzed.

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INTRODUCTION

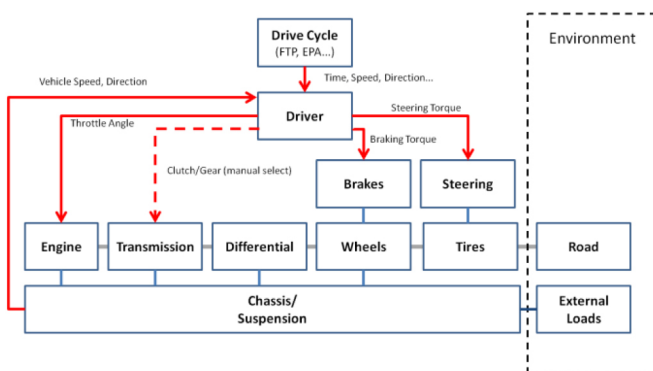


Figure 1. General structure of full-vehicle model

The goal of this project will be to build a predictive model of a complete vehicle in the physical modeling tool, MapleSim, in order to study the forces at tire/road interface

during various drive cycles and investigate any relationship between them and fuel efficiency. This document will summarize the development of the model, covering:

- Powertrain
 - Engine
 - Transmission
- Chassis
 - Longitudinal
 - 3D multibody
- Steering
- Wheels
 - Tires
 - Brakes

The report will cover the use of a simple Driver model for implementing accelerator/brakes and steering angle as well as some basic tests to validate the behavior of the model.

TARGET VEHICLE

2006 Infiniti M35



Horsepower*	280 hp @ 6,200 rpm	Wheelbase(m)	114.2
Torque	270 lb-ft @ 4,800 rpm	Overall length(m)	192.6
Maximum engine speed	6,600 rpm	Overall width(m)	70.8
Gear ratios		Overall height(m)	59.4
1st	3.842	Front track width(m)	60.2
2nd	2.333	Rear track width(m)	60.8
3rd	1.529	Minimum ground clearance(m)	5.1
4th	1.000	Coefficient of Drag(CD)	0.28
5th	0.839	Curb weight (lbs.)	Front 2,052 Rear 1,780 Total 3,832
Reverse	2.765	Weight distribution (% fr/re)	54/46
Final drive ratio	3.538	GVWR (lbs.)	5,053
RWD		Frontal area(A)	24.8 sq ft
Automatic Gearbox		Top speed	225 Km/hr
Tires	245/45R18	0-96 Km	5.9 s

Also consider: $\mu_R = 0.015$, $\eta_T = 0.85$, $I_G = 0.2 \text{ kgm}^2$, $h = 0.36 \text{ m}$

Figure 2. Specification data for the target vehicle

For this project, it was considered important to have a target vehicle in mind so that the model could be based on real specification data to produce as realistic a model as possible. The initial target vehicle is the 2006 Infiniti M35, due to the ready availability of specification data for the chassis and powertrain. However, the model is sufficiently structured to allow data from any passenger vehicle to be used. A summary of the data for the target vehicle is given in Figure 2.

Parameters subsystem default settings

Name	Type	Default Value	Default Units	Description
r_{eff}	Real	0.34		Tire effective radius (m)
m_{veh}	Real	1734		Vehicle mass (kg)
L_{wb}	Real	2.9		Wheel base (m)
L_{cg}	Real	0.54 L_{wb}		Distance between CG and front wheel centre
L_{r}	Real	0.46 L_{wb}		Distance between CG and rear wheel centre
A_{ref}	Real	0.23		Front area (m ²)
GR_1	Real	3.842		Gear Ratio 1
GR_2	Real	2.333		Gear Ratio 2
GR_3	Real	1.529		Gear Ratio 3
GR_4	Real	1.0		Gear Ratio 4
GR_5	Real	3.538		Gear Ratio 5
C_d	Real	0.28		Air drag coefficient of friction
η_T	Real	0.85		Overall transmission efficiency
J_{engine}	Real	0.2		Engine inertia (kg m/s)
h_{CG}	Real	0.36		CoG height(m)
μ_x	Real	0.015		Road contact coefficient of friction
ρ_{air}	Real	1.2		Density of air (kg/m ³)
m_{tire}	Real	28		Mass of tire (kg)
m_{axle}	Real	10		Mass of front axle assembly (kg)
m_{axle}	Real	10		Mass of rear axle assembly (kg)
m_{sprung}	Real	$4 \cdot m_{\text{tire}} + 2 \cdot m_{\text{axle}}$		Unsprung Mass (kg)
m_{sprung}	Real	$m_{\text{veh}} - m_{\text{sprung}}$		Sprung Mass (kg)
GR_{diff}	Real	3.895		Differential Gear Ratio

Figure 3. Example parameter set for the M35

In MapleSim, this data has been made available as a set of Parameter Blocks that allow the user to readily replace the values with parameters for other vehicles. For this project,

Parameter Blocks are used to define the overall parameters for the vehicle, including the engine and chassis. As an illustration, the overall parameter set is shown in Figure 3.

Parameter Blocks are also used for defining different gear-shift settings, such as Sport, for maximum speed studies, and Economy for steady speed and steering studies. This will be covered in the Transmission section.

POWERTRAIN

The goal of the powertrain model is to provide a high-fidelity representation of the vehicle's engine and transmission to reflect changes in torque at the vehicle's wheels. This includes dynamic changes in engine speed, hydraulic coupling in the torque converter and changes in gear ratio, as well as torque split to the driven wheels through a differential.

Engine Model

The target vehicle uses a conventional internal combustion engine (ICE) to convert stored energy in the fuel to energy required to move the vehicle at a desired speed. The rate at which this energy is delivered to the rest of the vehicle, often referred to as the "brake power", is governed by the general equations:

$$P = \tau * \omega \quad (1)$$

where τ is the torque required to turn the drive shaft at ω rad/s, and

$$\tau = \dot{\omega} * J \quad (2)$$

where J is the overall inertia of the vehicle and $\dot{\omega}$ is the angular acceleration of the shaft, from which the longitudinal acceleration of the vehicle can be found.

However, the torque is dependent upon various loads that are applied to the engine from the rest of the vehicle, such as friction and other loads from the tires, gravity (going up and down hills) and aerodynamic drag. Therefore the available torque for accelerating the vehicle is given as

$$\tau = \tau_{\text{ind}} - (\tau_{\text{load}} + \tau_{\text{friction}}) \quad (3)$$

where τ_{ind} is the torque generated by the fuel combustion in the engine. Hence, the acceleration is given by

$$\dot{\omega} = \frac{\tau_{\text{ind}} - (\tau_{\text{load}} + \tau_{\text{friction}})}{J} \quad (4)$$

Two engine models were implemented: a Torque Driver model and a Mean Value model that includes gas flow dynamics in order to compute the fuel being consumed. The use of the Torque Driver model was very useful for making a conceptual connection between the torque-curve characteristic determined from an earlier study and the torque

output from the more physics-based mean value model. In the initial study, a first-approximation for the torque characteristic was derived from the manufacturer's data for maximum power and torque. This was then compared with the torque characteristic from the mean-value model, produced by performing a "virtual dyno test" on it. Each of these was implemented in the vehicle model with the Torque Driver model and the performance results compared. Finally, the mean-value model was implemented and compared with the Torque Driver results.

Torque Driver Engine Model

Figure 4 shows a typical torque/power vs engine speed characteristic curve that is available from the manufacturer or can be produced by loading the engine with a dynamometer over a range of speeds. A Torque Driver engine model implements such a torque-speed characteristic as a lookup table and uses this to compute the maximum available torque to send to a torque driver in the MapleSim environment. Essentially, it is implementing the following equation:

$$\tau(x, \omega_e) = x * Te(\omega_e) \quad (5)$$

where x is the throttle area ratio (0..1), and ω_e is the engine speed.

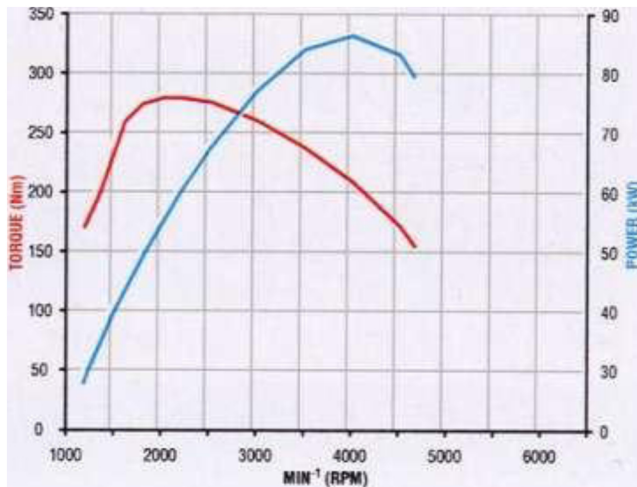


Figure 4. Torque/Power Curve for typical family sedan

In this way, the physical constraints in the design of an engine can be implemented by limiting how much indicated torque can be generated, and this varies over the speed range of the engine. Typically, the indicated torque is low at low and high speeds, with a maximum somewhere in between. In physical terms, this means that, at high speeds, as the indicated torque approaches the load and friction torques, acceleration approaches 0 thus limiting the maximum speed of the vehicle. At low speeds (close to 0), if the indicated torque is less than the load and friction torques, acceleration will be 0 so the vehicle will never start moving. Effectively, the engine stalls.

For this project, the first torque characteristic to be implemented was a simple quadratic estimator, based on the following expressions for torque and power:

$$Te(\omega_e) = A_0 + \omega_e A_1 + \omega_e^2 A_2 \quad (6)$$

$$Pe(\omega_e) = \omega_e(A_0 + \omega_e A_1 + \omega_e^2 A_2) \quad (7)$$

where A_0 , A_1 and A_2 are arbitrary coefficients. Also, it is known that at the point of maximum torque, $\frac{dTe(\omega_e)}{d\omega_e} = 0$, therefore

$$A_1 + \omega_e A_2 = 0 \quad (8)$$

Substituting the values of max torque, power and the engines speeds at which they occur, the values for the coefficients can be found by solving the simultaneous equations. Using the data from the manufacturer:

Max Torque @ 4,800 RPM = 270 lb-ft = 366 Nm,

Max Power @ 6,200 RPM = 280 HP = 21 kW

The resulting quadratic torque curve (QTC) is:

$$Te(\omega_e) = -157.8 + 0.22 \Omega - 0.000023 \Omega^2 \quad (9)$$

where Ω is the speed in RPM. The resulting curve is shown in Figure 5. This is a parabola with a maximum of 366 Nm at 4,800 RPM. For the purpose of this project, the speed range is limited to 2000-6600 RPM.

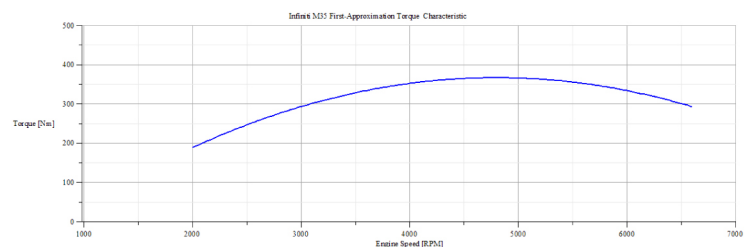


Figure 5. Quadratic Torque Curve (QTC)

As part of the MapleSim Driveline Component Library (DCL), there is an "Engine" component that implements the Torque Driver model, as described above. To use the QTC model, a data set was generated from the function and stored in an Excel spreadsheet. Implementation and testing will be discussed in a later section.

Mean Value Engine Model (MVEM)

This is a physics-based model that determines the overall power output and, hence, output torque and speed of the drive shaft without considering the details of the per-cycle power output. Because they are relatively fast to compute, they are

typically used in simulations that need to run for long durations, such as controller testing and drive-cycle testing.

The MV model that will be used in this project has been proposed by Saeedi [3] that draws on several references to implement a fairly simplified gas-flow and combustion model in MapleSim.

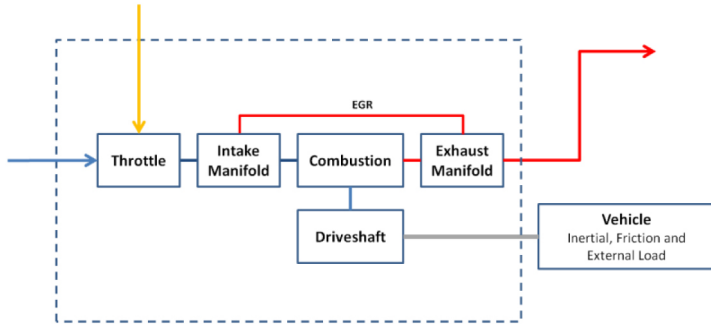


Figure 6. Mean Value Engine Model

The model, shown in Figure 7, provides the overall power/speed/torque/fuel consumption, given the mass flow of air/fuel gas. The gas flow is determined by the position of the throttle valve, which is set by a simplified engine controller that closes the loop between the actual and desired engine speeds.

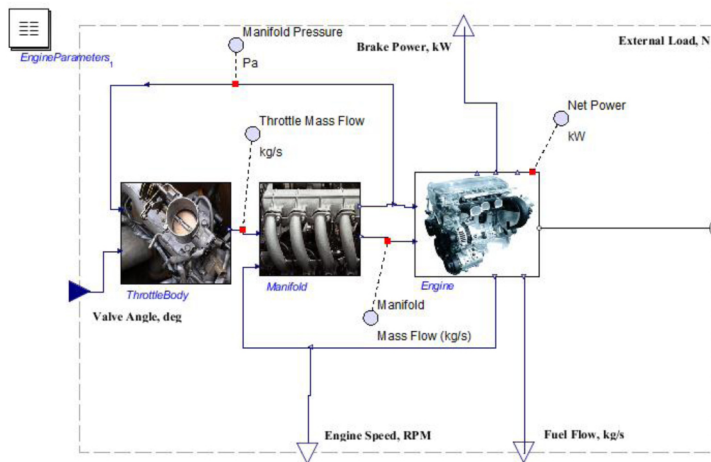


Figure 7. Structure of Mean-value Engine model in MapleSim

The engine model itself is composed of three main subsystems: the throttle, intake manifold and engine power generation from the fuel combustion. Loading on the engine shaft is provided by a model of a dynamometer. For the purpose of this project, the exhaust system and exhaust gas recirculation (EGR) will not be considered.

Throttle Model

The throttle model calculates the air mass flow based on the throttle valve angle. For the purposes of the model the angle is provided by the driver model as an input signal.

Based on the geometry of the throttle and valve, the effective throttle area (A_{thr}) is related to the valve angle (ϕ) from Moskwa [4]:

$$A_{thr} = \frac{-dD}{2} [1 - a^2]^{1/2} + \frac{dD}{2} \left[1 - \left(a \frac{\cos \phi}{\cos \phi_0} \right)^2 \right]^{1/2} + \frac{D^2}{2} \arcsin \left[(1 - a^2)^{1/2} \right] - \frac{D^2}{2} \frac{\cos \phi}{\cos \phi_0} \arcsin \left[1 - \left(a \frac{\cos \phi}{\cos \phi_0} \right)^2 \right]^{1/2} \quad (10)$$

where d = Throttle pin diameter (m), D = Throttle bore diameter (m), ϕ_0 = Throttle plate closed angle (rad), ϕ = Throttle plate angle (rad), $a = d/D$, diameter ratio, and A_{thr} = Effective throttle area (m^2)

The mass flow rate through the throttle is related to effective throttle area and manifold pressure from Heywood [5]:

$$\dot{m}_{thr} = \begin{cases} \frac{C_{D_{thr}} A_{thr} P_0}{\sqrt{RT_0}} \left(\frac{P_m}{P_0} \right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma-1} \left[1 - \left(\frac{P_m}{P_0} \right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2} & \text{if } \frac{P_m}{P_0} > \left[\frac{2}{\gamma+1} \right]^{1/\gamma-1} \\ \frac{C_{D_{thr}} A_{thr} P_0}{\sqrt{RT_0}} \gamma^{1/2} \left(\frac{2}{\gamma+1} \right)^{(\gamma+1)/2(\gamma-1)} & \text{if } \frac{P_m}{P_0} \leq \left[\frac{2}{\gamma+1} \right]^{1/\gamma-1} \end{cases} \quad (11)$$

where R = gas constant (287 kJ/kg), $C_{D_{thr}}$ = throttle discharge coefficient, T_0 = ambient temperature (278.15 K), γ = specific heat ratio (1.4), P_0 = ambient pressure (101.325 kPa = 1 bar), P_m = manifold pressure (kPa), and $\dot{m}_{thr}$ = throttle mass flow rate (kg/s).

There are two phases to this relationship: subsonic and supersonic (or choked), determined by the ratio between the manifold pressure and ambient pressure at the entrance to the throttle.

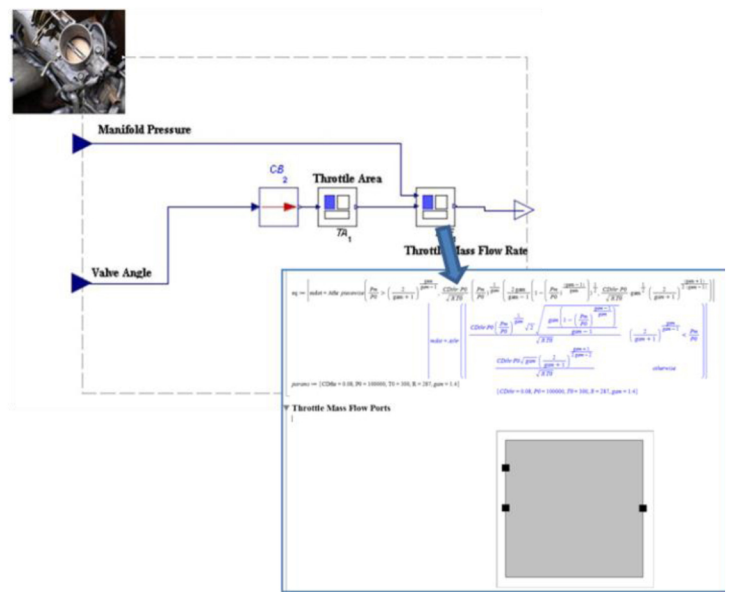


Figure 8. Custom Component creation in Maple

These equations were implemented as Custom Components, using Maple to type in the equations and generate the component, as shown [Figure 8](#).

Intake Manifold

The intake manifold has a significant effect on the gas mass flow and pressure as the air/fuel mixture enters the engine cylinders. There are two physical effects that need to be considered: calculation of the manifold pressure and volumetric efficiency.

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, into the cylinders. The calculation of the manifold pressure (P_m) and mass flow ($\dot{m}_e$) and, according to Hendricks [6], may be based on the ideal gas equation in derivative form by considering isothermal conditions in the intake manifold control volume.

$$P_m = \frac{RT_m}{V_m} (\dot{m}_{thr} - \dot{m}_e) \quad (12)$$

and

$$\dot{m}_e = \frac{\eta_{vol} V_d n}{60 N_{eng} R T_m} P_m \quad (13)$$

where N_{cyl} = number of cylinders, N_{eng} = engine stroke type number (2 for four-stroke engines and 1 for two-stroke), V_d = displaced volume in cylinder:

$$V_d = S \cdot \left(\frac{\pi \cdot B^2}{4} \right) \quad \text{where } S = \text{Stroke (m)}, B = \text{bore diameter (m)}, T_m = \text{manifold temperature, K (engine is considered to be working at steady state condition)}, V_m = \text{manifold volume (m}^3\text{)}, n = \text{engine speed (RPM)}, \text{ and } \eta_{vol} = \text{volumetric efficiency.} \quad (14)$$

diameter (m), T_m = manifold temperature, K (engine is considered to be working at steady state condition), V_m = manifold volume (m^3), n = engine speed (RPM), and η_{vol} = volumetric efficiency.

2. Volumetric Efficiency

Developing analytical models of the pressure losses in the gas through the manifold is extremely difficult, so this is typically modeled as a lookup table of experimental values of volumetric efficiency (η_{vol}) over a range of engine speeds and manifold pressures.

Engine Power Calculation

Given the air/fuel gas mass flow into the engine cylinders, the mean engine power generated from the combusted fuel, and subsequent torque and speed, can be found from the following engine equations from Hendricks [6]:

$$P_{net} = P_{ind} - P_{loss} - P_{load} \quad (15)$$

and

$$\dot{n} = \frac{1}{I_e n} [P_{ind} - P_{loss} - P_{load}] \quad (16)$$

where

$$P_{ind} = \dot{m}_f H_f \eta_{th} \quad (17)$$

$$P_{loss} = n [(a_1 + a_2 n + a_3 n^2) + (a_4 + a_5 n) P_m] \quad (18)$$

$$P_{load} = k_0 n^3 \quad (19)$$

$$\dot{m}_f = \frac{\dot{m}_e}{\lambda L_{th}} \quad (20)$$

H_f = fuel heating value for gasoline is 46000 kJ/kg, λ = air/fuel ratio, L_{th} = stoichiometric normalization factor (=14.67), I_e = engine inertia, η_{th} = thermal efficiency, n = engine speed (rpm or krpm), P_{ind} = indicated power (kW), P_{load} = load power (kW), P_{loss} = loss power (kW), and $\dot{m}_f$ = fuel flow rate (kg/s)

Note that the relationships for the load and loss power are empirical models where a_1 , a_2 , a_3 , a_4 , a_5 , and k_0 are all constants obtained by experiment.

Also given in Hendricks, the thermal efficiency (η_{th}) is modeled using an empirically derived polynomial expression, accounting for speed (n), manifold pressure (P_m), and air-fuel ratio (λ):

$$\eta_{th} = \eta_{thn} \eta_{thPm} \eta_{th\lambda} \quad (21)$$

where

$$\eta_{thn} = b_1 (1 - b_2 n^{b_3}), \quad \eta_{thPm} = c_1 + c_2 P_m + c_3 P_m^2 \quad (22)$$

, and

$$\eta_{th\lambda} = \begin{cases} d_1 + d_2 \lambda + d_3 \lambda^2 & \text{if } \lambda > 1 \\ d_4 + d_5 \lambda + d_6 \lambda^2 & \text{if } \lambda \leq 1 \end{cases} \quad (23)$$

The coefficients, b_n , c_n and d_n are typically generated from manufacturers' testing data and are considered highly proprietary. For the purpose of this project, Hendricks provides some coefficients that can be used.

Implementation in MapleSim

The gas flow model for the MVEM was developed using a combination of signal-flow components, custom components and a lookup table. In order to implement the computed available power as torque and speed, the available torque was calculated from the net, or “available”, power and used as an input to a MapleSim Torque Driver component. In this way, the “causal” (signal-flow) representation can be converted to an “acausal” representation for the mechanical connection to the engine's driveshaft. Implementing the inertia of the engine then becomes a simple task of connecting an Inertia component to the Torque Driver mechanical port, as shown in [Figure 9](#).

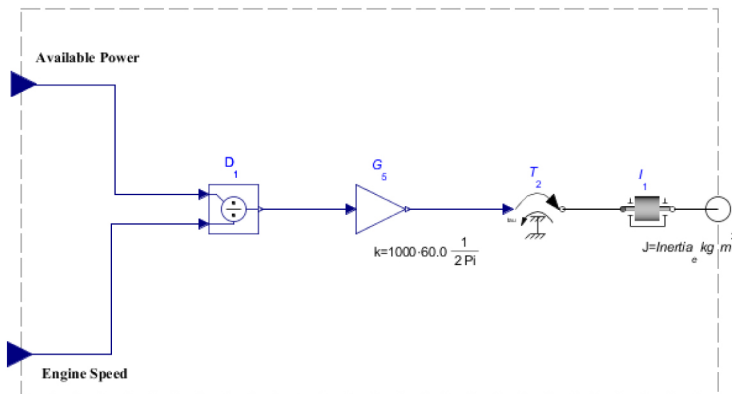


Figure 9. Implementation of calculated net power as available torque

Generating the Torque/Speed Curve from the MVEM: The “Virtual Dyno Test”

Before implementing the MVEM, it is important to compare its performance characteristic with the simple QTC. Assuming all the engine parameters match the real engine, the torque curve should be close to the QTC and should help to make a conceptual connection between this first approximation and the presumably higher fidelity model.

Mean-value IC Engine Model Dynamometer Test

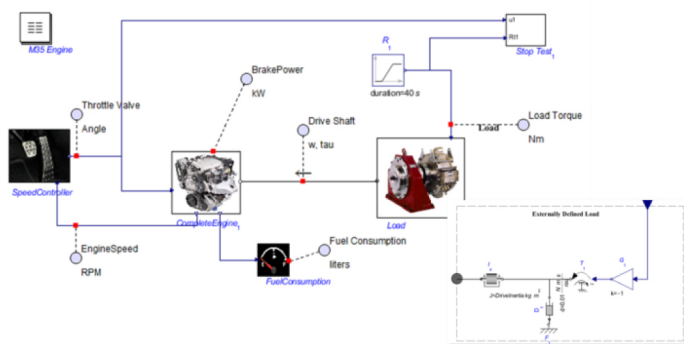


Figure 10. Engine/Dynamometer test in MapleSim. Inset: detail of dynamometer model

To do this, the MVEM was connected to a model of a dynamometer, which is essentially a controlled torque driver that provides a load to the engine. [Figure 10](#) shows the dyno test setup with the detail of the dynamometer model.

The test procedure is then the same for the rest of the engine/dyno test:

- run the engine to a steady speed
- increase the dyno load until the engine stalls, or where the throttle opening is maximized
- record the load torque at which stall occurs
- increase the engine speed to the next value and repeat until the maximum speed of the engine is reached.

In MapleSim, the load test was performed for a value of the engine speed set-point in the controller. In [Figure 11](#) the dyno load is applied after 20 secs. As the load increases, the controller maintains the engine speed by opening the throttle valve to provide more power and compensate for the increased load. Eventually, the throttle valve cannot open any further and the engine begins to stall. The model has logic in it to stop the test when this occurs.

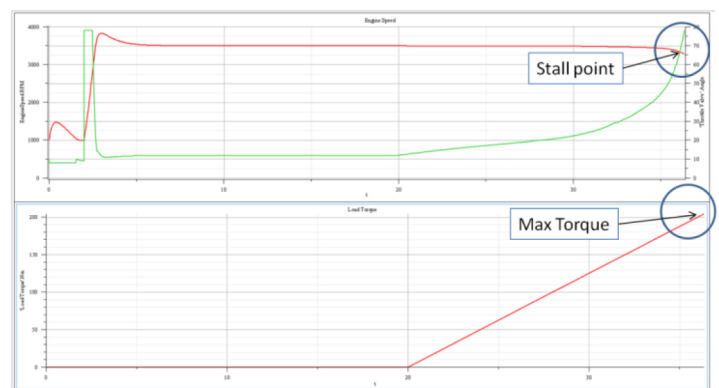


Figure 11. MVEM Engine/Dyno Test @ 3,500 RPM

The whole test could be performed in Maple by looping through the desired engine speeds and the resulting stall points plotted. In [Figure 12](#), the MVEM torque characteristic is plotted with the QTC in order to compare the two. Clearly, there is very little correspondence between the two, but it must be appreciated that the QTC result is very much a simplistic estimation based on two data points. The MVEM curve actually looks closer to the shape of a real dyno test, but the maximum torque is about 327 Nm at 4,000 RPM, compared to the published 366 Nm at 4,800. The differences may also be attributed to the fact that not all the engine parameters were available from the manufacturer, particularly volumetric and thermal efficiency lookups so there should be little surprise that there are discrepancies. In future work, it may be possible to find more realistic parameters to improve the model's fidelity.