Optimisation of Diesel Engine for Hybrid Military Tracked Vehicles using Matlab-Simulink

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ABSTRACT

The demand in the technology requirements for diesel engines is growing keeping hybrid vehicles in mind. In future the diesel engine no longer drives the wheels directly; as a result the engine can be engaged at a limited number of operating points, thus, offering an opportunity to optimise the fuel efficiency and performance at those operating points. The extent to which this optimisation is possible is limited by practical considerations. Also if the positive and negative power peaks in vehicle during mobility (e.g. acceleration and regenerative braking respectively) can be accommodated by high-power batteries, then the size of the engine can be considerably reduced. The engine's operating points depend on the power-control strategy. The consequences of modifications to these operating points will have an effect on performance and efficiency. As in series hybrid only a limited number of operating points are involved and dynamic performance requirements are not imposed on the diesel engine, significant improvements can be achieved by the optimisation of the diesel engine at these operating points. The feasibility of optimisation of the engine at these operating points can be done by modification on the injection systems, the valve timings and other such parameters. This kind of approach requires the use of complex and repeated experimental analysis of the engine which is costly, cumbersome and time consuming. An alternative to this kind of experimental approach is to develop a simulation model of the engine with the generator in Matlab- Simulink.

Keywords: Series hybrid; Diesel engine simulation; Simulink; Generator

1. INTRODUCTION

New developments in electric machines, power electronics and electric batteries boosted the development of alternative drives for military land vehicles. Hybrid-electric commercial vehicles are already bringing benefits in substantial fuel economy and better emission controls, both of these issues being the prime mover of the alternative drives development[9].

Although fossil fuels are facing dead treats, their energy density cannot be replaced by other means[2] and military vehicles (especially combat vehicles) could probably be the last to replace it. But considering numerous advantages of hybrid-electric vehicles such as on-board electric power, improved stealth operation by reducing the thermal & acoustic signatures, it is most likely that the hybrid-electric drives can replace the existing power pack in the military vehicles in future. But still, hybrid-electric drives for the military applications face many challenges such as energy storage, motor/generators, power electronics, reliability and user acceptance, etc. A feasibility study was carried out by CVRDE for adopting hybrid technology for tracked military vehicles and to bring out the pros and cons of parallel and series hybrid. The study reveals that the most apt hybrid configuration for tracked military vehicles is the series hybrid. Hence a detail study on engine behaviour in series configuration was taken.

1.1 Series Hybrid

In series architecture the diesel engine is coupled to integrated starter generator (ISG) and the energy source is a fuel tank. Through a electronic converter (rectifier) control, the output of the ISG is connected to a direct current (DC) bus. The battery connected with the power DC bus through a bilateral, controllable power electronic converter (DC/DC converter), is the bilateral energy source. ISG can be controlled in reverse and forward direction, hence can be used as motor or as generator. Between the Diesel Engine and the output sprocket there is no mechanical connection. Consequently, the Diesel Engine could be made to run at every point on the torque-speed (power) map. This advantage, with a refined power flow with controls, provides the engine with chance to operate in the region with highest efficiency. The control strategy of this configuration is simple, due to its mechanical decoupling between the engine and output sprocket. The typical Series Hybrid architecture is as shown in Fig. 1.

1.2 Methodology

The methodology followed in this study,
Step (i) Calculate Pressure-theta from the mathematical model
Step (ii) Determine the temperature and torque maps in the region of operating points.
Step (iii) Compute the discrete parameters (fuel consumption) for improvement at every working point and energy of the
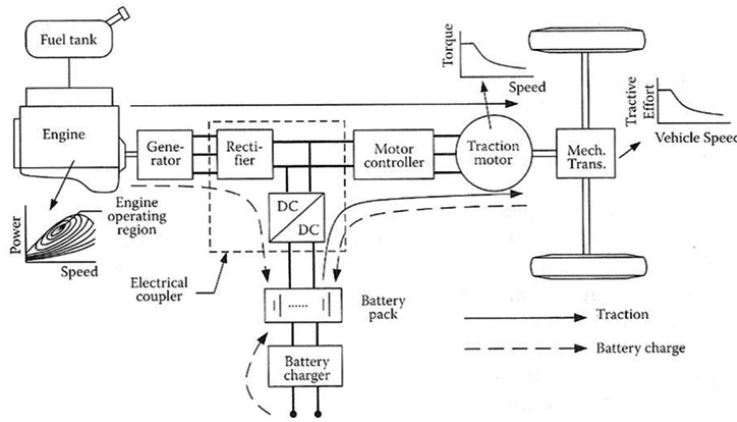
Figure 1. Series hybrid architecture.

fuel that would be used by the diesel engine in various mode as described in optimisation.

Calculate total impact parameters, which is a composition of outcomes for each of the working points as shown in Fig. 4. The main target of this approach is to improve energy usage. Such optimisation involves many varying parameters of the entire operating regime of the diesel engine. This kind of experimental approach would turn out to be very costly and time taking.

Hence an alternative to the experimental approach is to develop a similar simulation model of the engine process using mathematical models. The simulation model for the diesel engine is developed using Matlab-Simulink software[3].

2. ENGINE SIMULATION MODULE
2.1 Engine Specification

The prime mover under study is a 6-cylinder, direct-injection (DI), Naturally Aspirated diesel engine (UTD_20 Engine) with a peak power of 300 hp. The specification of the engine is given in the following Table 1.

Table 1. Engine specification

Parameter	Specification
Engine type	V6
Displacement	15.9 liters
Bore	0.150 m
Stroke	0.150 m
Rated power	300 hp @2600 rpm
Peak torque	1000 Nm @1600 rpm
Compression ratio	15.8
Engine weight	665 kg
Intake and exhaust valve seat diameter	0.048 m and 0.044 m

2.2 Simulation Approach

The approach to the simulation can be divided into three steps. The first step of the approach is to generate the pressure-theta curve (P-θ curve) of the engine. The P-θ curve is then fed as the input for the second step which is a virtual diesel engine subsystem developed in Simulink[3,4]. This engine subsystem can then be connected to the Generator Block/Model in Simulink, to simulate the performance characteristics of the engine and

the generator for given load cycle. The engine simulation is then carried out in a series hybrid power train configuration in Advance Vehicle Simulator.

3. GENERATION OF PRESSURE-THETA CURVE

The P-θ curve can be generated either with the help of thermodynamic 1-D software AVL Boost[10] or using a separate cylinder combustion simulation model developed in Simulink.

3.1 Approach using Cylinder Model

The model can be divided into four parts that interact with each other. The first part is the cylinder geometry[1]. In this part the piston motion, cylinder parameters and crank angle generation are included. The second part describes the mass flow subsystems and includes the effect of mass flow in the intake and exhaust valves. The third part consists of the combustion subsystems and implements the combustion processes. The fourth part works for cylinder thermodynamics including the generation of pressure and temperature of the cylinder[1]. Thus from the fourth part of the model the cylinder pressure with respect to crank angle can be obtained.

The simulink schematics for this step i.e. the cylinder model is given in Appendix-1(a). Figure 3 shows the overall layout of this model. The engine combustion process is determined through the heat release rate that is taken from the cylinder pressure data with respect to crank angle[10].

From Fig. 2 the vertical position of the piston, X can be determined with respect to the crank angle θ. Thus the instantaneous piston position as a function of crank angle can be expressed as:
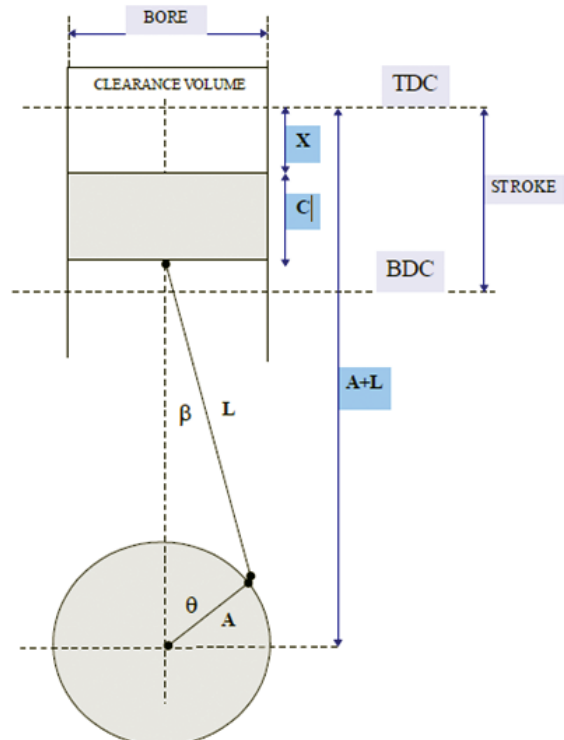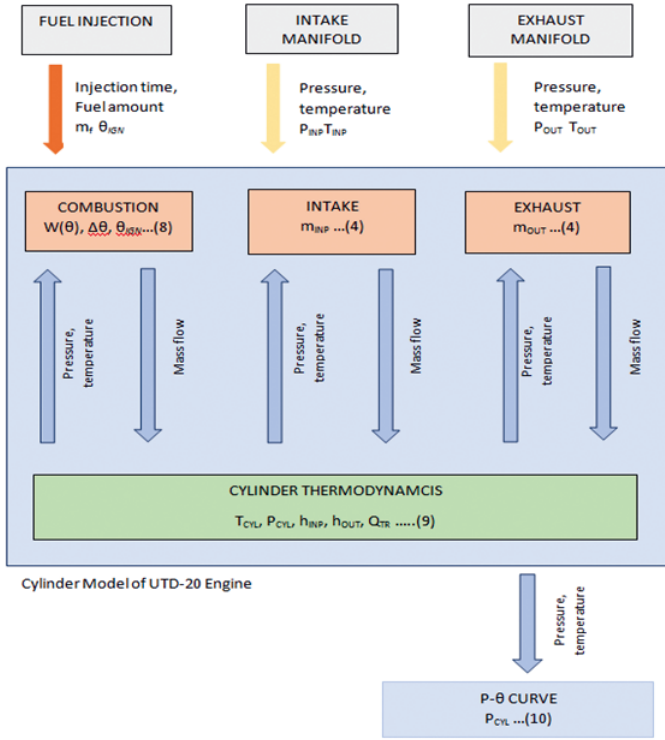

Figure 2. Cylinder geometry.

Figure 3. Cylinder simulation model layout.

$$X(\theta) = A + L - \left(\sqrt{L^2 - A^2 \sin^2(\theta)} \right) - A \cos\theta \qquad (1)$$

where,

A = crank radius in m

L = length of the connecting rod in m

The instantaneous piston speed is given by

$$\frac{dX}{d\theta} = \frac{A^2 \cdot \sin(2 \cdot \theta)}{\left(\sqrt{L^2 - A^2 \cdot \sin^2(\theta)} \right)} + A \cdot \sin(\theta)$$

$$= A \cdot \sin(\theta) \cdot \left(1 + \frac{2A\cos\theta}{\left(\sqrt{L^2 - A^2 \sin^2(\theta)} \right)} \right) \qquad (2)$$

The combustion chamber volume at any crank angle position in other words, the instantaneous cylinder volume in terms of crank angle can be expressed as,

$$V(\theta) = V_C + \frac{\pi.BORE}{4} \cdot X(\theta) \qquad (3)$$

These equations are implemented through the simulink schematics given in Appendix-1(b) and 1(c)

The equation, flow through valve for compressible fluids from thermodynamic relationships for isentropic expansion is as follows:

$$\dot{m}(t) = C_D \cdot A_R(t) \frac{P_{INP}(t)}{\sqrt{R_C T_{INP}(t)}} \cdot \psi\left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right) \qquad (4)$$

where the function

$\psi\left(\dfrac{P_{INP}(t)}{P_{OUT}(t)} \right)$ is defined as

$$\psi\left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right) = \sqrt{K \cdot \left(\frac{2}{K+1} \right)^{\left(\frac{K+1}{K-1} \right)}} \quad \text{for } P_{OUT} < P_{CR}$$

$$\psi\left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right) = \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right)^{\frac{1}{K}} \sqrt{\left(\frac{2.K}{K-1} \right) \cdot \left(1 - \frac{P_{INP}(t)}{P_{OUT}(t)} \right)^{\left(\frac{K-1}{K} \right)}}$$

for $P_{OUT} \geq P_{CR}$

The term $A_R(t)$ refers to the area of the valve. This area can be approximated by

$$A_R(t) = \pi \cdot (ValveDiameter) \cdot (ValveLift(\theta)) \qquad (5)$$

These equations are used for calculating the mass flow through Intake and Exhaust valves, the simulink schematics are given in Appendix-1.(d) and 1.(e). The flow coefficient and discharge coefficient were taken as 0.529 and 0.560 at maximum valve lift.

The combustion block includes the Wiebe function[7,8,10] which gives the amount of fuel combusted at given crank angle. This block also includes the energy released during combustion. The Wiebe function is given by

$$W(\theta) = 1 - \exp\left(-a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta} \right)^n \right) \qquad (6)$$

where a, n are parametric functions , $\Delta\theta$ is the combustion duration in degrees of crank angle and θ_{IGN} is the crank angle at start of combustion. Wiebe function parameters were calculated for premixed and diffusion phase of the diesel engine using double Wiebe function . A Combustion duration of 3 degrees for premixed and 65 degree for diffusion phase in crank angle, and 99 per cent mass fraction burned were considered. The shape parameter and 'a' as 1.5 and 6.9 respectively, were selected for the simulation run[10].

Simulink schematics for the combustion block is given in Appendix-1(f)

The temperature change in the cylinder can be found using the following equation which is derived based on the first law of thermodynamics.

$$\frac{dT_{CYL}}{d\theta} = \frac{\sum_i h_{INPi} \cdot \left(\frac{dm_{INPi}}{d\theta} \right) - \sum_i h_{OUTi} \cdot \left(\frac{dm_{OUTi}}{d\theta} \right) - \sum_i h_i(T) \atop \frac{dm_i}{d\theta} + k_i \cdot T_{CYL}\frac{dm_i}{d\theta} - P_{CYL} \cdot \frac{dV_{CYL}}{d\theta} + \frac{dQ_{TR}}{d\theta}}{\sum_i c_{Pi}(T) \cdot m_i - k_i \cdot m_i} \qquad (7)$$

where,

P_{INP} = intake manifold pressure in Pa

P_{OUT} = exhaust manifold pressure in Pa

T_{CYL} = cylinder temperature in °C

m_{inp} = mass flow through intake manifold kg/s

h_{inp} = specific enthalpy of intake mass flow J/kg

h_{out} = specific enthalpy of exhaust mass flow J/kg

m_{out} = mass flow through intake manifold kg/s

P_{CYL} = cylinder pressure in MPa

Q_{TR} = Heat transferred J/s

c_p = specific heat capacity J/kg.K

k = no. of moles of the gas

B = Bore in m

$u_{P=}$ Mean Piston Speed in m/s

where A_{CYL} is the convention area calculated from the cylinder geometry and T_{WALL} is the average temperature of area surrounding .

$$P_{CYL} = \frac{T_{CYL}}{V_{CYL}} \sum_i r_i \cdot m_i \tag{8}$$

From the equation (7) and (8), the cylinder pressure can be determined with respect to crank angle.

The heat loss to the wall or the heat transfer in the cylinder is generally given by

$$\frac{dQ_{TR}}{dt} = h_C \cdot A_{CYL} \left(T_{CYL} - T_{WALL} \right) \tag{9}$$

The heat transfer in crank angle becomes

$$\frac{dQ_{TR}}{d\theta} = \frac{dQ_{TR}}{dt} \cdot \frac{dt}{d\theta} = \frac{dQ_{TR}}{dt} \cdot \frac{60}{2 \cdot \pi \cdot N} \tag{10}$$

where N is the engine speed in rpm.

The convection heat transfer coefficient hc can be calculated by empirical Woschni heat transfer correlations.

$$h_C = 3.26 \cdot 10^{-3} \cdot B^{-0.2} \cdot P^{0.8} \cdot T^{-0.55} \cdot u_P^{0.8}$$

All these equations are implemented in the simulink block cylinder thermodynamics given in Appendix -1.(g)

3.2 Approach using AVL Boost Software

Alternatively, the pressure-crank angle curve, generated using commercial code AVL boost[10]. In this, the module is run to generate the P-θ curve at one particular speed. This process can then be iterated to generate the P-θ curves for a range of speeds say from 1000 to 2600 rpm.

4. VIRTUAL DIESEL ENGINE SUBSYSTEM

The second step in the approach is to develop a virtual diesel engine module with the help of basic elements from the Simulink library[3]. The six-cylinder diesel engine model will then be used to drive virtual power train consisting of the Generator in the Simulink environment[4-6]. The Simulink schematics for this model is given in Appendix-2.

4.1 Inputs to the Virtual Engine Model

The P-θ curve generated in the previous step is the primary input for this model. The P-θ curve is generated for speeds from 1000 rpm to 2600 rpm. The torque map is then generated from the P-θ curve. Basic engine parameters as given in Engine specification, is used for this calculation.

A matlab script will be used to solve the torque values for the P-θ curve at every engine speed. These values will be made into a 3-D table with torque, engine speed and crank angle in the three axes respectively. If the selected value lies between the three-dimensional matrix specified, then it automatically calculates by means of extrapolation and interpolation methods used between the nearest values. These methods are linear. The first input is the engine speed in the band from 800 rpm to 2600 rpm. The crank angle is the second input. There are other subsystems for each crank throw in the block, which match the

crank angle with the torque map. The throttle position is the third input which indicates with regard to the maximum and minimum fuel supply.

4.2 Model Description

The virtual engine model can be approached based on torsional model including rotating masses in the moments of inertia as the input. Separately rotating masses are interconnected by torsion springs with the stiffness and damping properties. These elements represent the rotational movement, while extended blocks are connected to each of the rotating masses, providing as a torque output from the cylinders.

To each of the blocks, the 3-D look up table that loads, the generated three-dimensional matrix provided with the torque input data calculated from the P-θ curves is connected. The matrix will be shifted for each cylinder separately according to the engine's firing order. Another important feature of the model is the engine speed control.

The final model is with the elements of mechanical losses. For this, rotary dampers are used which increase the torque with respect to speed. This torque counteracts the generated torque by the pressure in the engine cylinders and thus calculates the engine mechanical losses. The torque output of the model should be consistent and in line with the torque measured on a test bed engine.

4.3 Engine Torque Calculation

The force resulting from the pressure in the cylinder

$$F_P = A_p \left(p_{ind} - p_{atm} \right) \tag{11}$$

where, p_{ind} the indicated cylinder pressure in Pa, p_{atm} the atmospheric pressure in Pa.

The inertial forces of dynamic parts are also incorporated in the calculation. The inertial forces of movable parts is given by

$$F_{im} = -m_p R_{crank} \omega_m^2 \left(\cos(\alpha) + \lambda \cos(2\alpha) \right) \tag{12}$$

where m_p is the overall reduced mass of dynamic engine components, R_{crank} crankshaft radius , λ-crankshaft ratio and α crank angle.

The axial direction force acting in the cylinder

$$F_a = F_p + F_{im} \tag{13}$$

The force acting on the connecting rod axis

$$F_{cr} = \frac{F_a}{\left(\cos\beta \right)} \tag{14}$$

where connecting rod angle is given by β.

The tangentially force acting on the crank throw is given by

$$F_t = F_{cr} \sin(\alpha + \beta) \tag{15}$$

The engine torque

$$T = R_{crank} F_t \tag{16}$$

Another approach to determine the torque acting on the piston is by directly calculating the gas pressure torque using the following equation,

$$Gas\ Pressure\ Torque = P * A_p * A * \sin\theta * \left[1 + \frac{A\cos\theta}{L} \right] \tag{17}$$

$$Inertia\ Torque = M_{rec}A^2\omega^2\left[\frac{A\sin\theta}{4L} - \frac{\sin 2\theta}{2} - \frac{3A\sin 3\theta}{4L}\right] \quad (18)$$

where

M_{rec} = Reciprocating mass in kg
θ = Crank angle in deg
P = Cylinder pressure in MPa
A_p = Top area of piston exposing to cylinder gas in m^2

$$Net\ torque = Gas\ pressure\ torque + Inertia\ torque \quad (19)$$

5. COUPLING THE ENGINE MODEL WITH GENERATOR MODEL

In the final step of the approach, the engine module is to be coupled with a generator model from the Simulink library which acts as the Diesel Generator. The generator can be coupled to the engine module and simulated for various load profiles or load cycles to simulate the behaviour of the engine when run with a generator in a series configuration. The generator block can be further linked to 3-phase electric loads to include the effect of drag torque or reactive torque environment[9]. The simulink schematics for this step is as shown in Appendix-3.

6. ENGINE OPTIMISATION

To identify the optimised operating points for the engine, the model is simulated in Advanced Vehicle Simulator written in Matlab-simulink. Initially the advisor file for the engine was created using engmodel function. The simulation is then run for series hybrid configuration along with generator and the power train for the given load cycle. The engine efficiency map is then plotted from the simulation results. The typical engine optimisation map is as shown in Fig. 4.

6.1 Operating Modes

The following operating modes are considered in this study:

Engine with ISG traction mode - The vehicle is powered by the engine, hence the power demand is handled by the diesel engine alone.

Hybrid traction mode - Here, both the diesel engine and electric power from battery, provide traction power to the drive train. This mode should be applied when the charge of the battery is high and then the demand of the peak load can be met.

Motor-alone mode - The diesel engine is shut-off. The vehicle is powered by the electric motor alone, through the battery hence used at low driving velocity.

Battery charge mode -Here, the ISG works as generator and powered by the diesel engine, to charge the battery during propulsion or during idling.

6.2 Fuel Consumption MAP

The fuel map of UTD_20 Engine for series hybrid application is as shown in Fig. 5. The map gives the fuel consumption plots with respect to torque and speed, then the operating points are selected such that the engine always operates in the best efficiency region denoted by Δ.
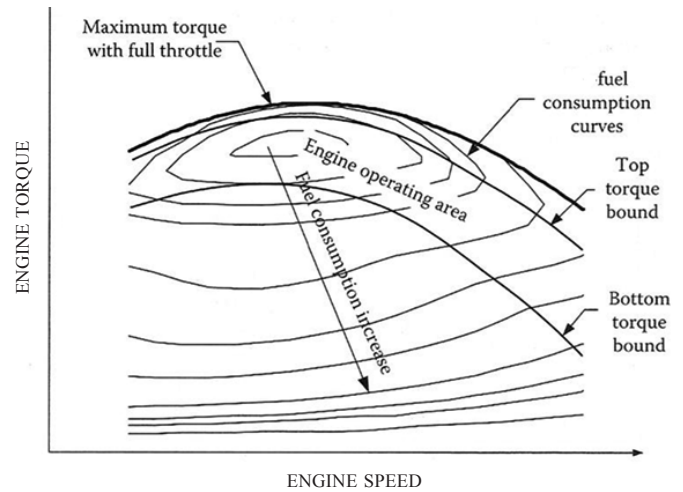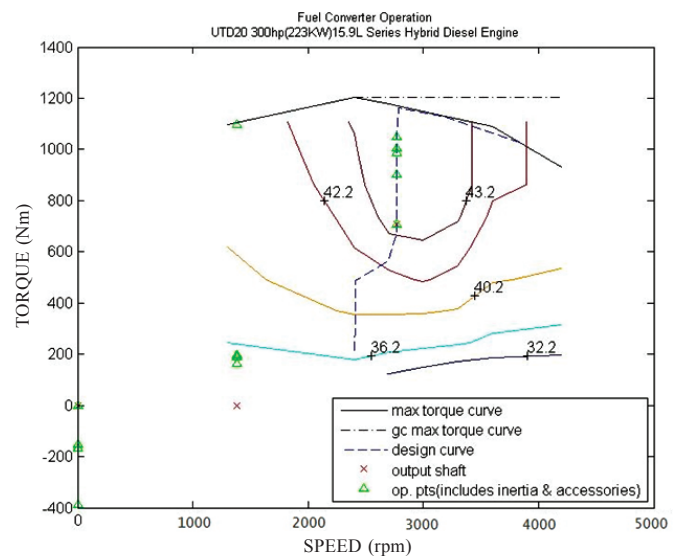

Figure 4. Engine optimisation map.

Figure 5. Engine map of UTD_20 engine.

7. CONCLUSIONS

The approach to the development of a simulation module to analyse the behaviour and performance characteristics of the engine coupled to generator in a series powertrain configuration have been summarised. This type of simulation modules can be a very helpful step in the modelling and development of the total hybrid vehicle. Performance optimisation that represents efficiency over a driving cycle, can be achieved by such a simulation study.

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10. AVL BOOST- Manual.

CONTRIBUTORS

Mr Hari Viswanath has completed MTech (IC Engines) from IITM, Chennai in 2001 and currently working as Scientist `E' at CVRDE, Chennai. His invoved in design of diesel engine for armoured fighting vehicles (AFVs) and his research area includes Development of intake port, study of air motion and combustion process of diesel engine, advanced fuel injection system for diesel engines. Development of in cylinder diesel engine components like piston, liner and piston rings and tribology studies and diesel hybrid power train for AFVs.

Mr A. Kumarasamy, has completed MTech in Mechanical Engg from IITM, Chennai in 1995 and currently working as Scientist `G' at CVRDE, Chennai. He is involved in Development of diesel engine for armoured fighting vehicles (AFVs) for 400 hp to 1500 hp range. His research area includes Compact cooling system for AFVs, Efficient air filtration system for AFVs, Development of advanced and efficient technologies for sub systems of diesel engine and its peripheral systems and diesel hybrid power trains for AFVs.

Dr P. Sivakumar has completed his PhD (Machine Design) from IIT, Madras, in 2011. He is a Distinguished Scientist and is currently Director, Combat Vehicles Research and Development Establishment, Chennai. His research areas include design and development of AFV automatic transmission in the range of 150 hp - 1500 hp, combat aircraft transmission, conceptualization of configuration for main battle tanks both present and future, infantry combat vehicles, armoured repair and recovery vehicles, self-propelled catapult vehicles, carrier command post and unmanned ground vehicles.
His contribution in the current study is overall guidance during the work and conclusion through results.
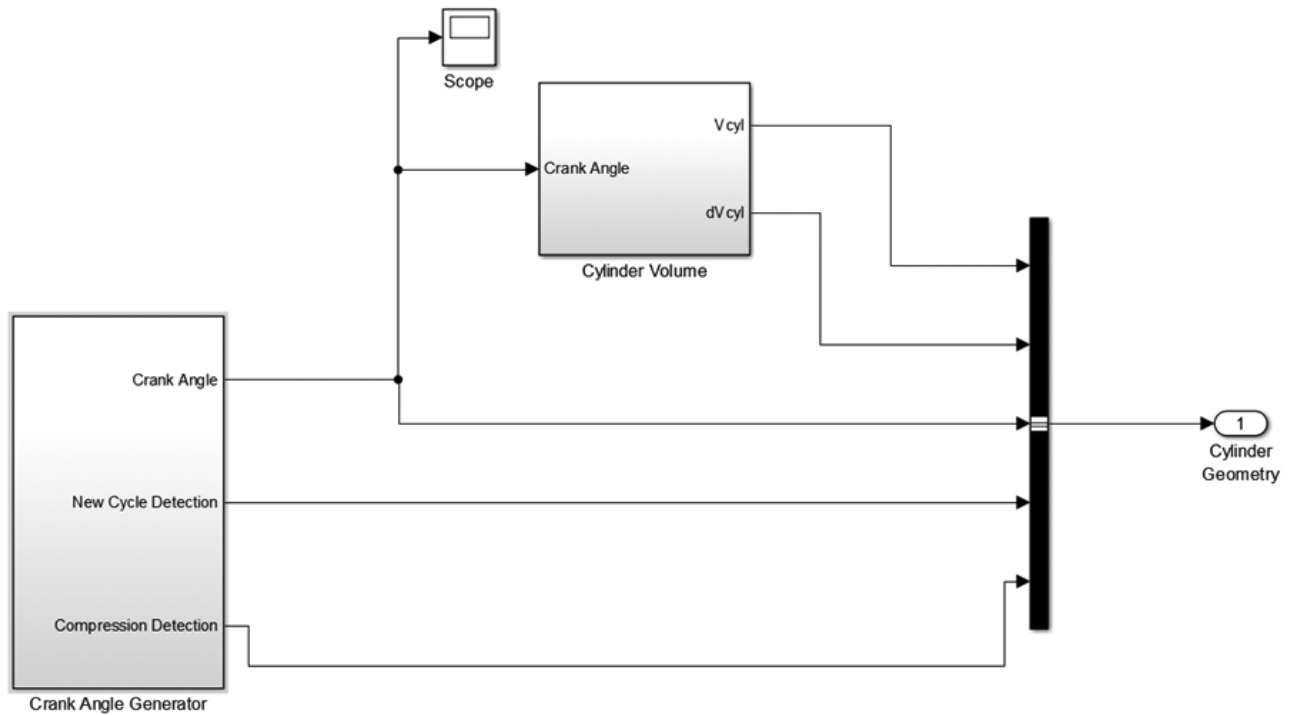
Appendix-1

Simulink Schematics
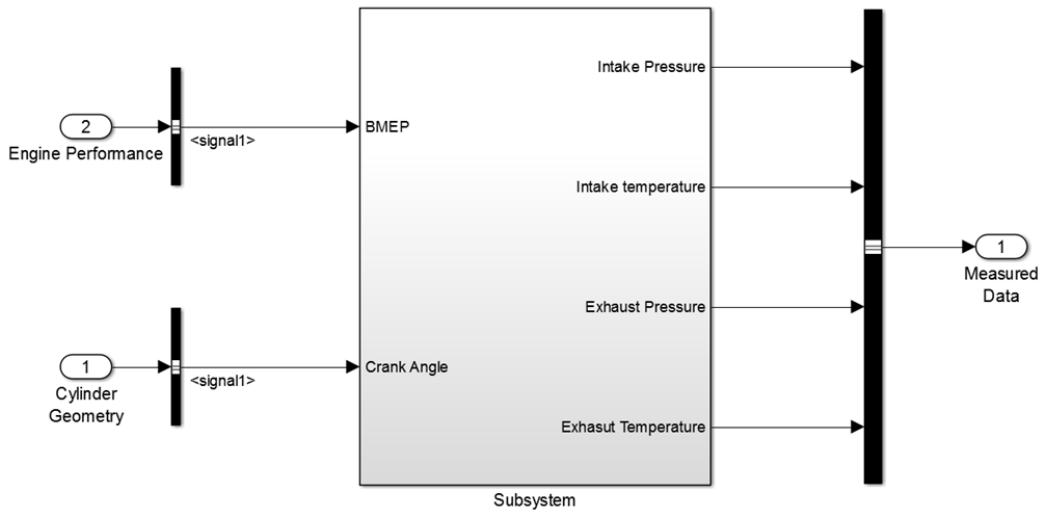*Cylinder model for p-θ generation*
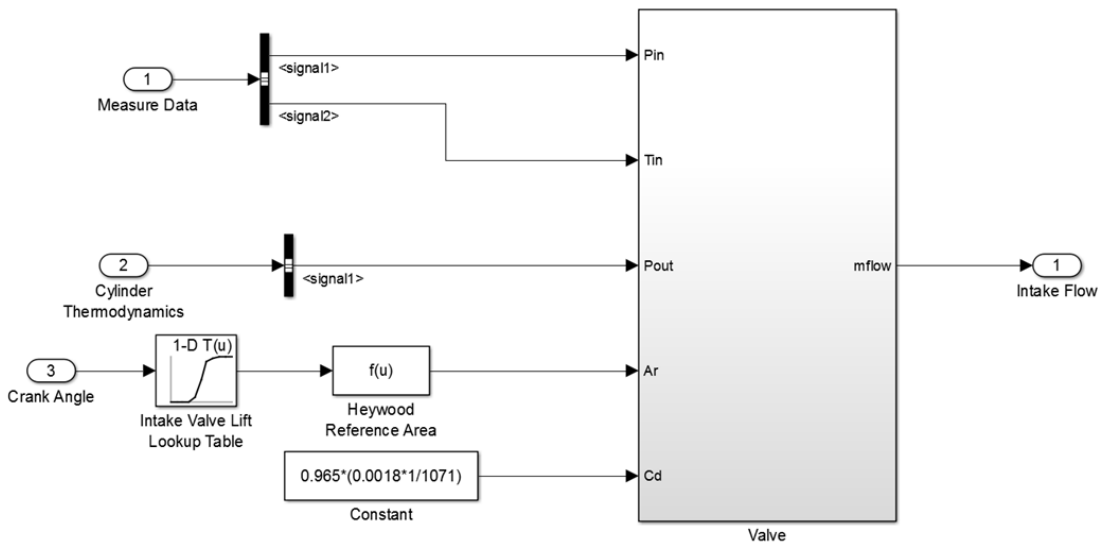
(a)   Full Cylinder Model
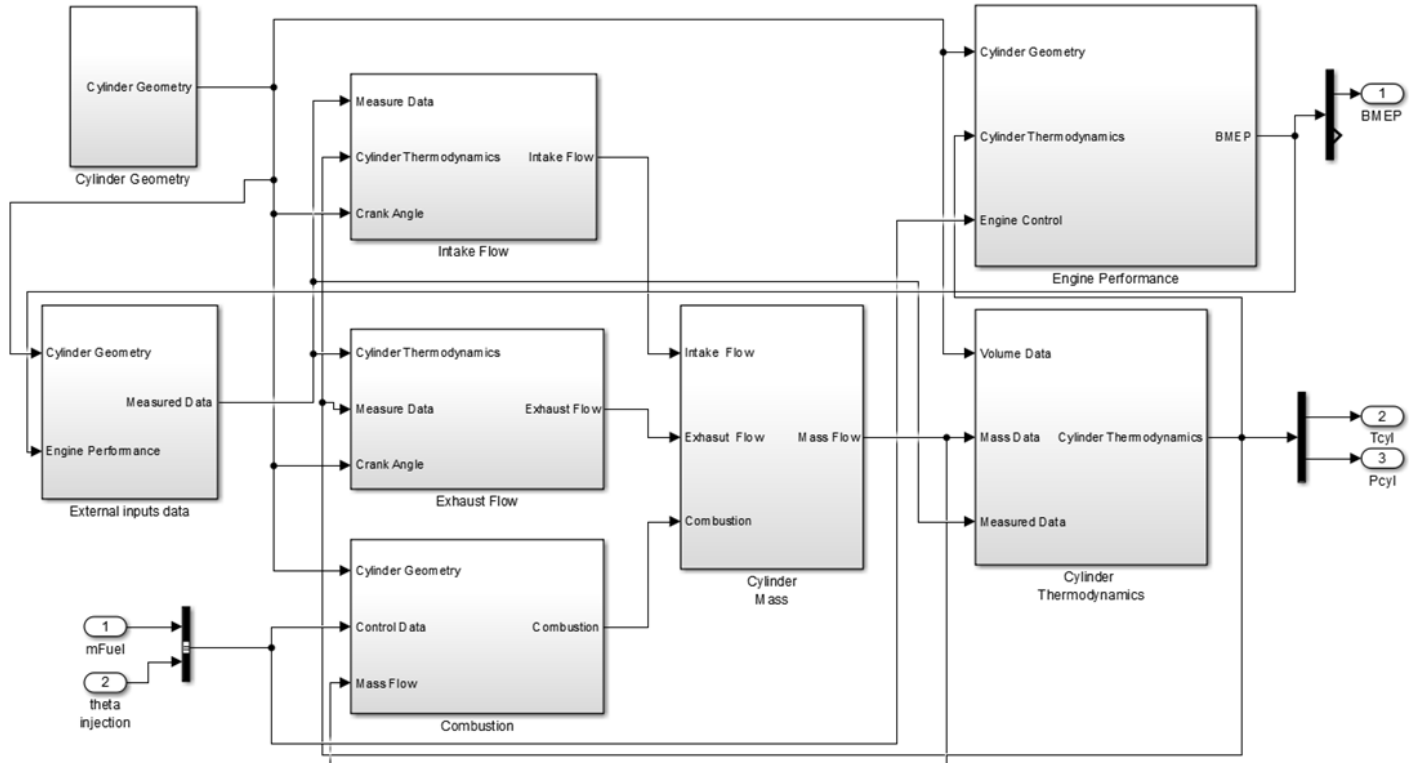
(b)   Cylinder Geometry

(c)   External Inputs
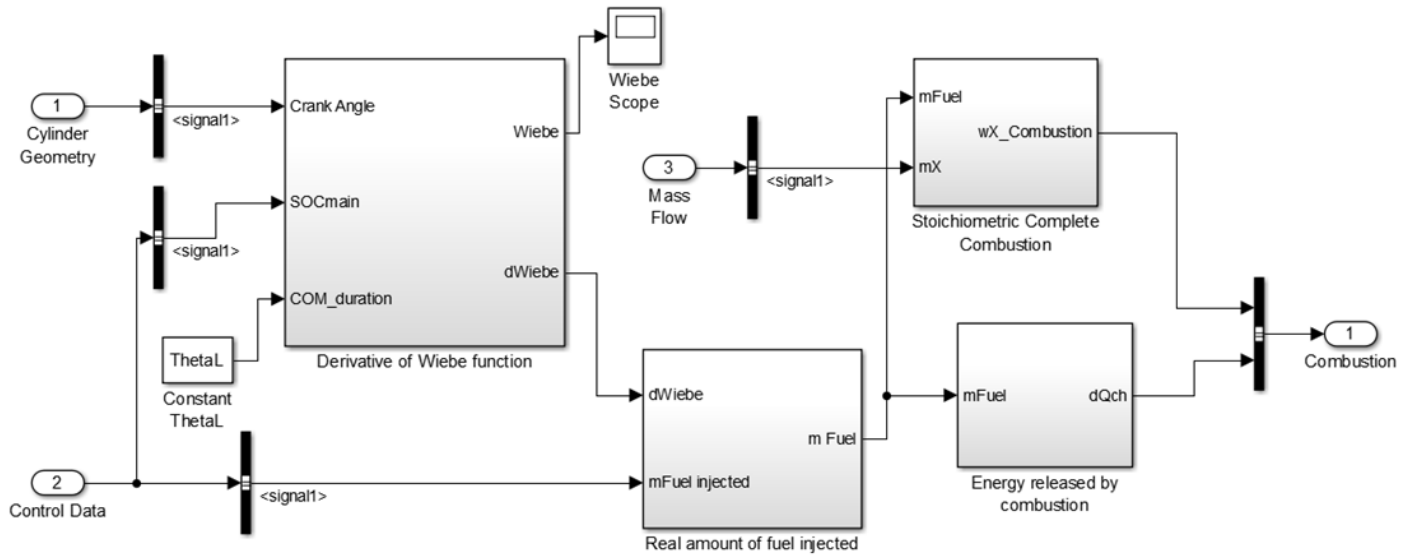

Subsystem

(d)   Intake Valve

Valve

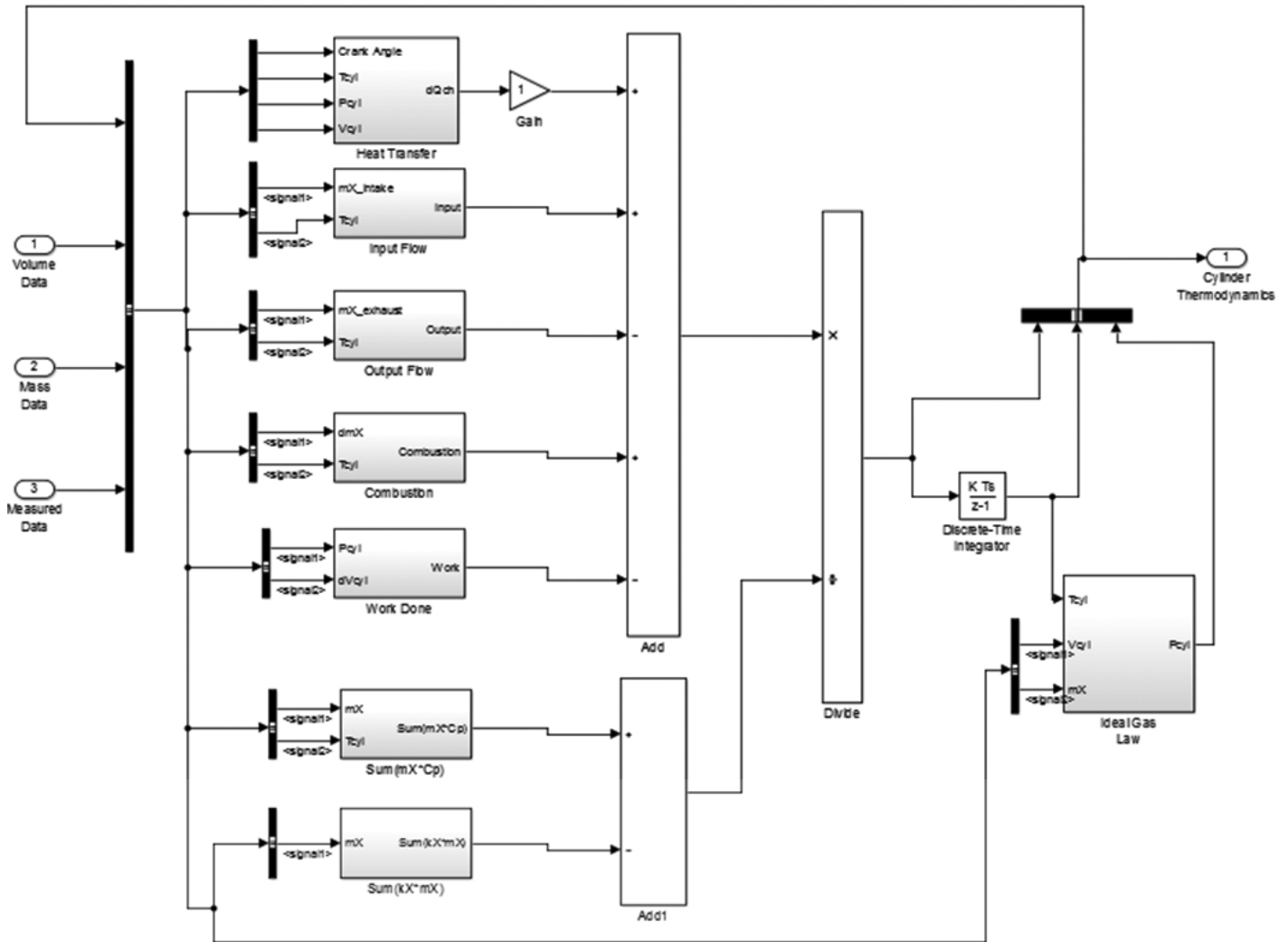*(e)   Exhaust Valve*

Valve
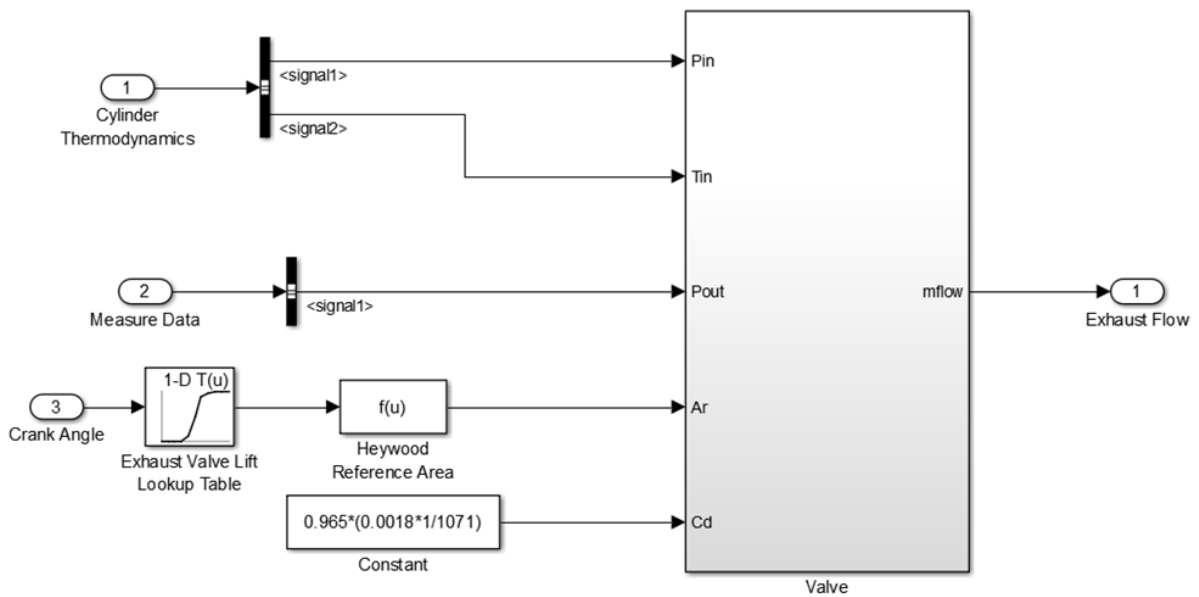
(f)   Combustion

(g)   Cylinder Thermodynamics

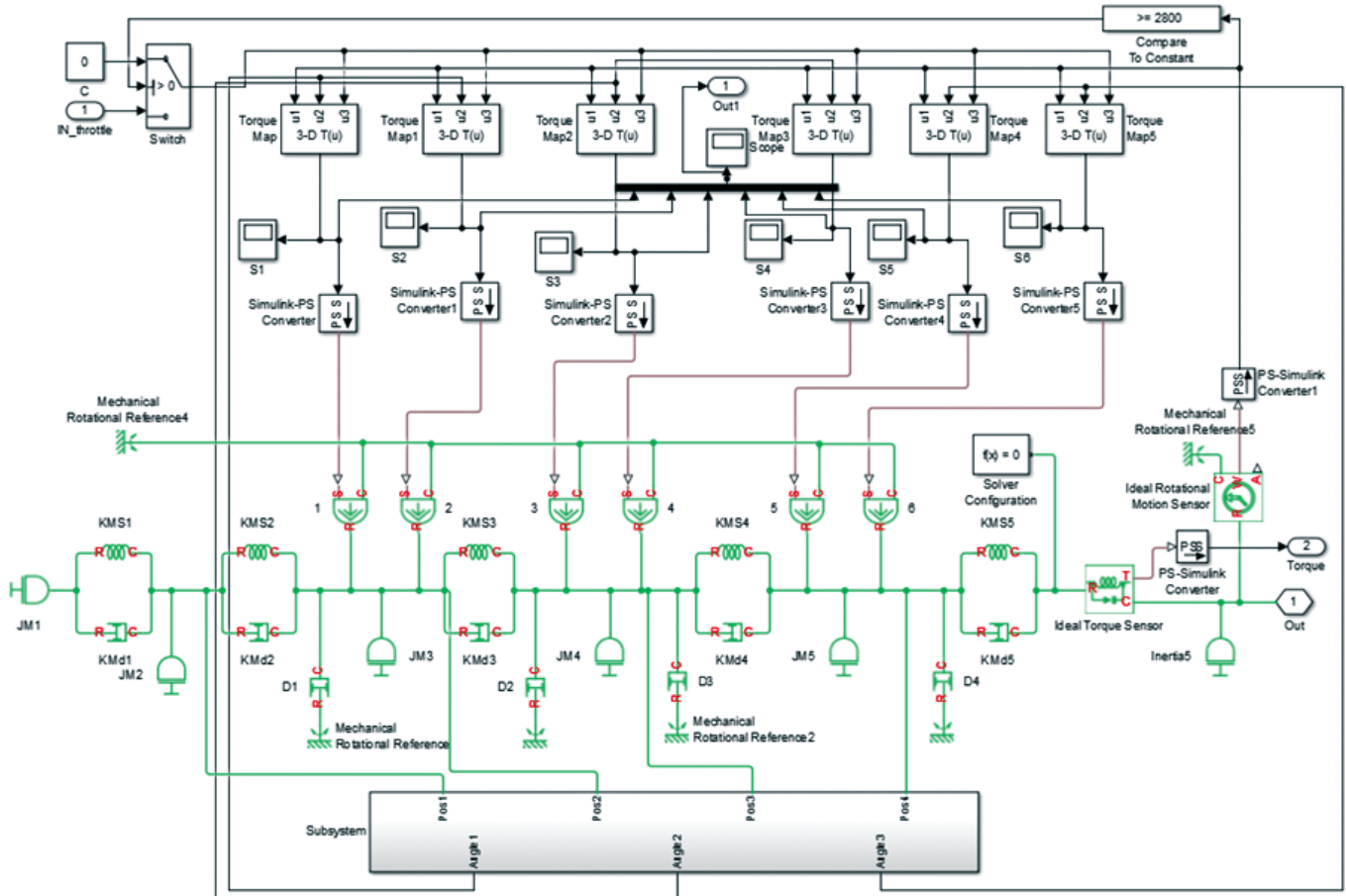# Appendix-2

Virtual Engine Subsystem

Appendix 3

Integrated Model of Engine and PM Generator