Vehicle Propulsion Systems
Lecture 2
Fuel Consumption Estimation & ICE Powertrains

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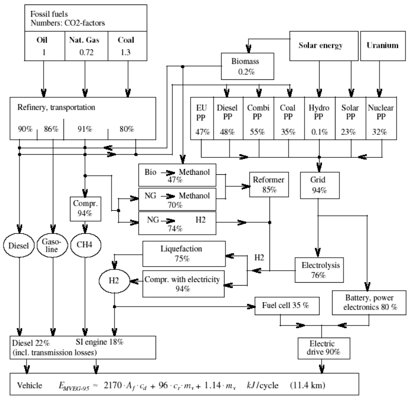
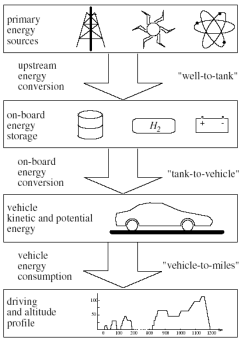
Outline

- Repetition
- Energy demand cont.
  - Energy demand and recuperation
  - Sensitivity Analysis
- Forward and Inverse (QSS) Models
- IC Engine Models
  - Normalized Engine Variables
  - Engine Efficiency
- Gear-Box and Clutch Models
  - Selection of Gear Ratio
  - Gear-Box Efficiency
  - Clutches and Torque Converters
- Analysis of IC Powertrains
  - Average Operating Point
  - Quasistatic Analysis
  - Software tools

W2M – Energy Paths



Energy System Overview



Primary sources

Different options for on-board energy storage

Powertrain energy conversion during driving

Cut at the wheel!

Driving mission has a minimum energy requirement.

Mechanical Energy Demand of a Cycle

Only the demand from the cycle

- The mean tractive force during a cycle

$$\bar{F}_{trac} = \frac{1}{x_{tot}} \int_0^{x_{tot}} \max(F(x), 0)\, dx = \frac{1}{x_{tot}} \int_{t \in trac} F(t) v(t) dt$$

  where $x_{tot} = \int_0^{t_{max}} v(t) dt$.
- Note $t \in trac$ in definition.
- Only traction.
- Idling not a demand from the cycle.

Evaluating the integral

Tractive force from *The Vehicle Motion Equation*

$$F_{trac} = \frac{1}{2} \rho_a A_f c_d v^2(t) + m_v g c_r + m_v a(t)$$

$$\bar{F}_{trac} = \bar{F}_{trac,a} + \bar{F}_{trac,r} + \bar{F}_{trac,m}$$

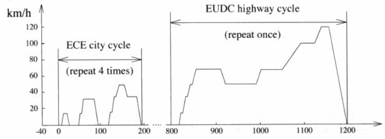
Resulting in these sums

$$\bar{F}_{trac,a} = \frac{1}{x_{tot}} \frac{1}{2} \rho_a A_f c_d \sum_{i \in trac} \bar{v}_i^3 h$$

$$\bar{F}_{trac,r} = \frac{1}{x_{tot}} m_v g c_r \sum_{i \in trac} \bar{v}_i h$$

$$\bar{F}_{trac,m} = \frac{1}{x_{tot}} m_v \sum_{i \in trac} \bar{a}_i \bar{v}_i h$$

Values for cycles



Numerical values for the cycles: {MVEG-95, ECE, EUDC}

$$\bar{X}_{trac,a} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{v}_i^3 h = \{319, 82.9, 455\}$$

$$\bar{X}_{trac,r} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{v}_i h = \{0.856, 0.81, 0.88\}$$

$$\bar{X}_{trac,m} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{a}_i \bar{v}_i h = \{0.101, 0.126, 0.086\}$$

$$\bar{E}_{\text{MVEG-95}} \approx A_f c_d 1.9 \cdot 10^4 + m_v c_r 8.4 \cdot 10^2 + m_v 10 \qquad kJ/100km$$

Tasks in Hand-in assignment

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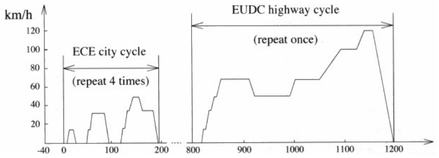
Energy demand again – Recuperation

- Previously: Considered energy demand from the cycle.
- Now: The cycle can give energy to the vehicle.



Recover the vehicle's kinetic energy during driving.

Perfect recuperation

- Mean required force

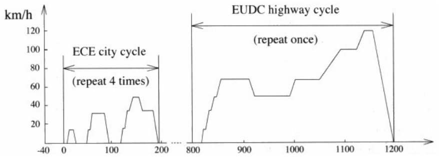
$$\bar{F} = \bar{F}_a + \bar{F}_r$$

- Sum over all points

$$\bar{F}_a = \frac{1}{x_{tot}} \frac{1}{2} \rho_a A_f c_d \sum_{i=1}^{N} \bar{v}_i^3 h$$

$$\bar{F}_r = \frac{1}{x_{tot}} m_v g c_r \sum_{i=1}^{N} \bar{v}_i h$$

Perfect recuperation – Numerical values for cycles



Numerical values for MVEG-95, ECE, EUDC

$$\bar{X}_a = \frac{1}{x_{tot}} \sum_i \bar{v}_i^3 h = \{363, 100, 515\}$$

$$\bar{X}_r = \frac{1}{x_{tot}} \sum_i \bar{v}_i h = \{1, 1, 1\}$$

$$\bar{E}_{\text{MVEG-95}} \approx A_f c_d \, 2.2 \cdot 10^4 + m_v c_r \, 9.81 \cdot 10^2 \qquad kJ/100km$$

Comparison of numerical values for cycles

- Without recuperation.

$$\bar{X}_{trac,a} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{v}_i^3 h = \{319, 82.9, 455\}$$

$$\bar{X}_{trac,r} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{v}_i h = \{0.856, 0.81, 0.88\}$$

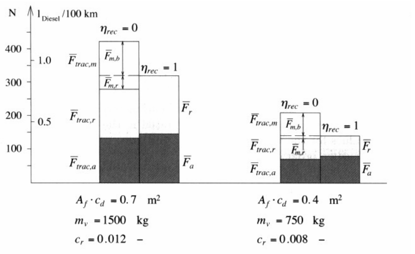
$$\bar{X}_{trac,m} = \frac{1}{x_{tot}} \sum_{i \in trac} \bar{a}_i \bar{v}_i h = \{0.101, 0.126, 0.086\}$$

- With perfect recuperation

$$\bar{X}_a = \frac{1}{x_{tot}} \sum_i \bar{v}_i^3 h = \{363, 100, 515\}$$

$$\bar{X}_r = \frac{1}{x_{tot}} \sum_i \bar{v}_i h = \{1, 1, 1\}$$

Perfect and no recuperation



Mean force represented as liter Diesel / 100 km.

Sensitivity Analysis

- Cycle energy reqirement (no recuperation)

$$\bar{E}_{\text{MVEG-95}} \approx A_f c_d \, 1.9 \cdot 10^4 + m_v c_r \, 8.4 \cdot 10^2 + m_v \, 10 \qquad kJ/100km$$

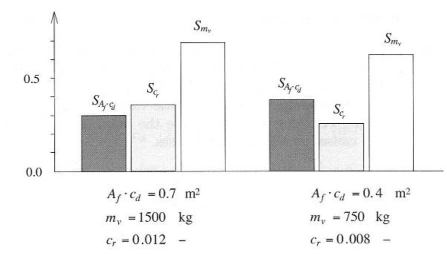
- Sensitivity analysis

$$S_p = \lim_{\delta p \to 0} \frac{\left[\bar{E}_{\text{MVEG-95}}(p + \delta p) - \bar{E}_{\text{MVEG-95}}(p)\right] / \bar{E}_{\text{MVEG-95}}(p)}{\delta p / p}$$

$$S_p = \lim_{\delta p \to 0} \frac{\left[\bar{E}_{\text{MVEG-95}}(p + \delta p) - \bar{E}_{\text{MVEG-95}}(p)\right]}{\delta p} \frac{p}{\bar{E}_{\text{MVEG-95}}(p)}$$

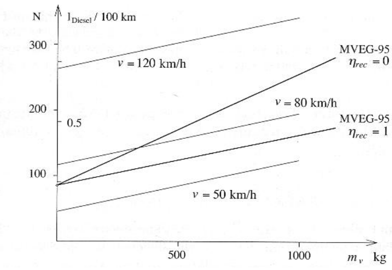
- Vehicle parameters:
  - $A_f c_d$
  - c_r
  - m_v

Sensitivity Analysis

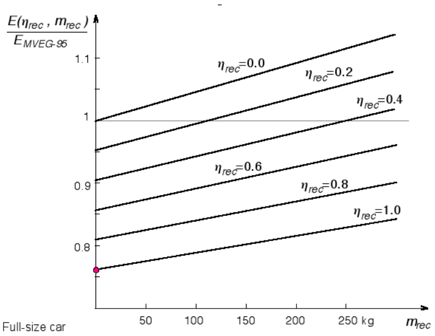


Vehicle mass is the most important parameter.

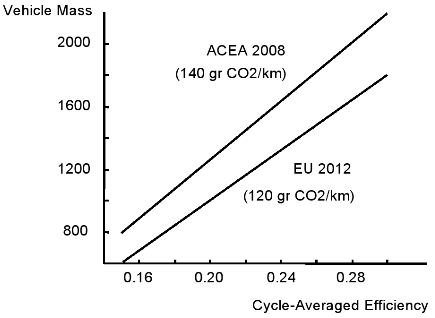
Vehicle mass and fuel consumption



Realistic Recuperation Devices



Vehicle Mass and Cycle-Averaged Efficiency



Outline

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Forward and Inverse (QSS) Models

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Analysis of IC Powertrains

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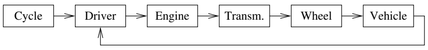
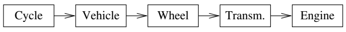
Two Approaches for Powertrain Simulation

- Dynamic simulation (forward simulation)



–“Normal” system modeling direction
–Requires driver model

- Quasistatic simulation (inverse simulation)



–”Reverse” system modeling direction
–Follows driving cycle exactly

- Model causality

Dynamic approach

- Drivers input u propagates to the vehicle and the cycle
- Drivers input $\Rightarrow \ldots \Rightarrow$ Driving force $\Rightarrow$ Losses $\Rightarrow$ Vehicle velocity $\Rightarrow$ Feedback to driver model
- Available tools (= Standard simulation) can deal with arbitrary powertrain complexity.

Quasistatic approach

- Backward simulation
- Driving cycle $\Rightarrow$ Losses $\Rightarrow$ Driving force $\Rightarrow$ Wheel torque $\Rightarrow$ Engine (powertrain) torque $\Rightarrow \ldots \Rightarrow$ Fuel consumtion.
- Available tools are limited with respect to the powertrain components that they can handle. Considering new tools such as Modelica opens up new possibilities.
- See also: *Efficient Drive Cycle Simulation*, Anders Fröberg and Lars Nielsen (2008) . . .

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Causality and Basic Equations

High level modeling – Inputs and outputs

- Causalities for Engine Models

- Engine efficiency

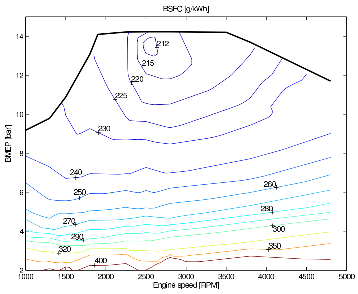
$$\eta_e = \frac{\omega_e T_e}{P_c}$$

- Enthalpy flow of fuel (Power $\dot{H}_{fuel} = P_c$)

$$P_c = \dot{m}_f \, q_{LHV}$$

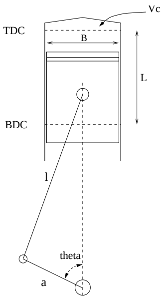
Engine Efficiency Maps

Measured engine efficiency map – Used very often



–What to do when map-data isn't available?

Engine Geometry Definitions



Cylinder, Piston, Connecting rod, Crank shaft

- Bore, B
- Stroke, $S = 2\,a$
- Number of cylinders z
- Cylinder swept volume, $V_d = \frac{\pi\,B^2\,S}{4}$
- Engine swept volume, $V_d = z\,\frac{\pi\,B^2\,S}{4}$
- Compression ratio $r_c = \frac{V_{max}}{V_{min}} = \frac{V_d+V_c}{V_c}$

Definition of MEP

See whiteboard.

Normalized Engine Variables

- Mean Piston Speed ($S_p = mps = c_m$):

$$c_m = \frac{\omega_e\,S}{\pi}$$

- Mean Effective Pressure (MEP=p_{me} ($N = n_r \cdot 2$)):

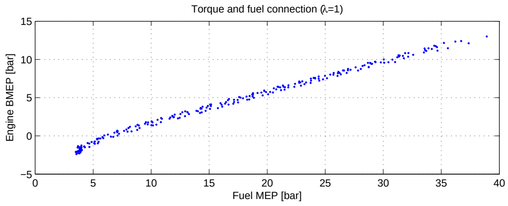
$$p_{me} = \frac{N\,\pi\,T_e}{V_d}$$

- Used to:
  - Compare performance for engines of different size
  - Design rules for engine sizing.
    At max engine power: $c_m \approx 17$ m/s, $p_{me} \approx$ 1e6 Pa (no turbo)
    $\Rightarrow$ engine size
  - Connection:

$$P_e = z\,\frac{\pi}{16}\,B^2\,p_{me}\,c_m$$

Torque modeling through – Willans Line

- Measurement data: x: p_{mf} y: $p_{me} = BMEP$

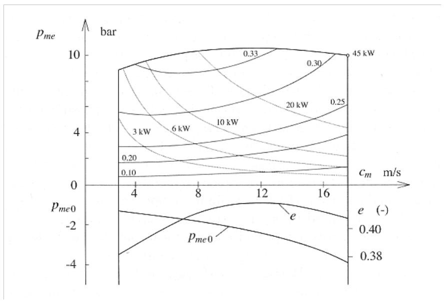


- Linear (affine) relationship – Willans line

$$p_{me} = e(\omega_e) \cdot p_{mf} - p_{me,0}(\omega_e)$$

- Engine efficiency: $\eta_e = \frac{p_{me}}{p_{mf}}$

Engine Efficiency – Map Representation



Willans line parameters: $e(\omega_e)$ $p_{me,0}(\omega_e)$

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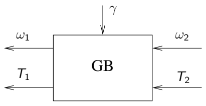
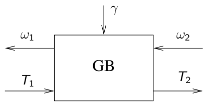
Causality and Basic Equations

- Causalities for Gear-Box Models

Quasistatic Approach



Dynamic Approach



- Power balance – Loss free model

$$\omega_1 = \gamma\omega_2, \qquad T_1 = \frac{T_2}{\gamma}$$

Different Types of Gearboxes

- Manual Gear Box
- Automatic Gear Box, with torque converter
- Automatic Gear Box, with automated clutch
- Automatic Gear Box, with dual clutches (DCT)
- Continuously variable transmission

Connections of Importance for Gear Ratio Selection

- Vehicle motion equation:

$$m_v \frac{d}{dt} v(t) = F_t - \frac{1}{2}\rho_a A_f c_d v^2(t) - m_v g c_r - m_v g \sin(\alpha)$$

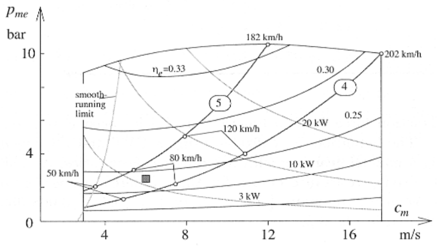
Constant speed $\frac{d}{dt}v(t) = 0$:

$$F_t = \frac{1}{2}\rho_a A_f c_d v^2(t) + m_v g c_r + m_v g \sin(\alpha)$$

- A given speed v will require power $F_t v$ from the powertrain.
- This translates to power at the engine $T_e \omega_e$.
  Changing/selecting gears decouples ω_e and v.
- Required tractive force increases with speed.
  For a fixed gear ratio there is also an increase in required engine torque.

Selection of Gear Ratio

Gear ratio selection connected to the engine map.



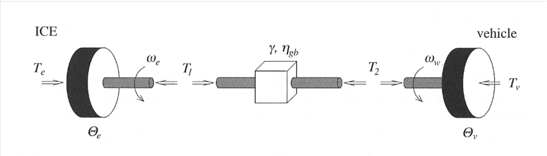
Additionally: Also geometric ratio between gears.
$\frac{i_{g,1}}{i_{g,2}} \approx \frac{i_{g,2}}{i_{g,3}} \approx \frac{i_{g,3}}{i_{g,4}} \approx \frac{i_{g,4}}{i_{g,5}}$

Selection of Gear Ratio

Optimizing gear ratio for a certain cycle.

- Potential to save fuel.
- Case study 8.1 (we'll look at it later).

Gear-box Efficiency



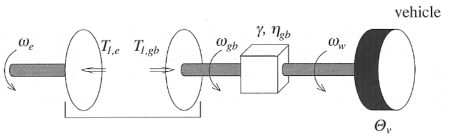
- In traction mode

$$T_2 \omega_w = e_{gb} T_1 \omega_e - P_{0,gb}(\omega_e), \qquad T_1 \omega_e > 0$$

- In engine braking mode (fuel cut)

$$T_1 \omega_e = e_{gb} T_2 \omega_w - P_{0,gb}(\omega_e), , \qquad T_1 \omega_e < 0$$

Clutch and Torque Converter Efficiency

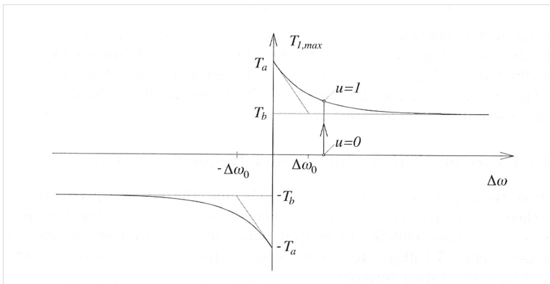


Friction clutch torque:

$$T_{1,e}(t) = T_{1,gb}(t) = T_1(t)\ \forall t$$

Action and reaction torque in the clutch, no mass.

Torque Characteristics of a Friction Clutch



Approximation of the maximum torque in a friction clutch

$$T_{1,max} = \text{sign}(\Delta\omega)\left(T_b - (T_b - T_a)\cdot e^{-|\Delta\omega|/\Delta\omega_0}\right)$$

Main parameters in a Torque Converter

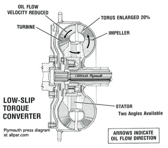
Input torque at the converter:

$$T_{1,e}(t) = \xi(\phi(t))\,\rho_h\, d_p^5\, \omega_e^2(t)$$

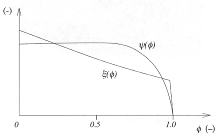
Converter output torque

$$T_{1,gb}(t) = \psi(\phi(t)) \cdot T_{1,e}(t)$$

Graph for the speed ratio $\phi(t) = \frac{\omega_{gb}}{\omega_e}$, and the experimentally determined $\psi(\phi(t))$



Plymouth press diagram at allpar.com



The efficiency in traction mode becomes

$$\eta_{tc} = \frac{\omega_{gb}\, T_{1,gb}}{\omega_e\, T_{1,e}} = \psi(\phi)\,\phi$$

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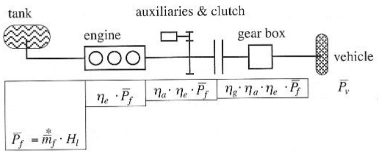
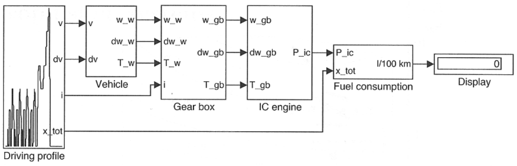
Average Operating Point Method



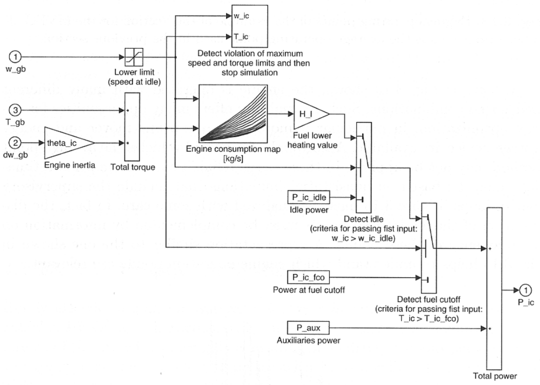
- Average operating point method
  –Good agreement for conventional powertrains.
- Hand-in assignment.

Quasistatic analysis – Layout

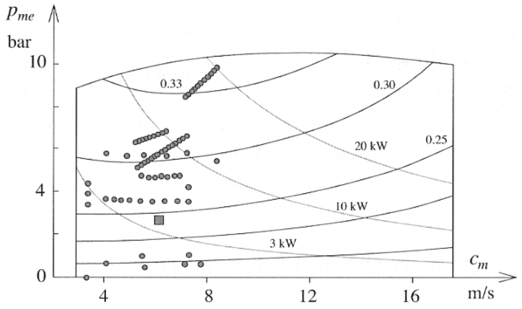


- More details and better agreement (depends on model quality)
  –Good agreement for general powertrains
- Hand-in assignment.

Quasistatic analysis – IC Engine Structure



Quasistatic analysis – Engine Operating Points



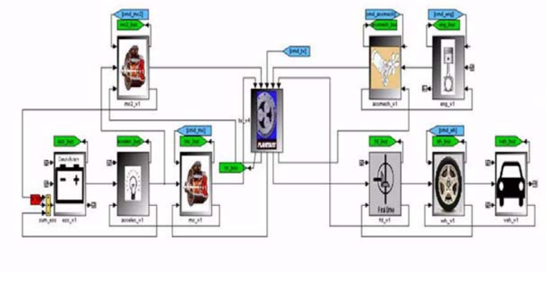
Software tools

Different tools for studying energy consumption in vehicle propulsion systems

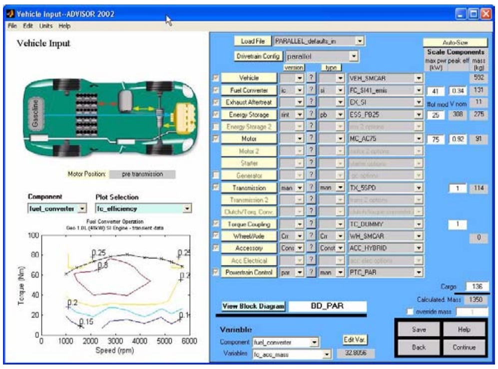
	Quasi static	Dynamic
QSS (ETH)	X	
Advisor, AVL	X	(X)
PSAT		X
CAPSim (VSim)		X
Inhouse tools	(X)	(X)

PSAT

Argonne national laboratory



Advisor



Load File: PARALLEL_defaults_in

Drivetrain Config: parallel

		version	type		max pwr (kW)	peak eff	mass (kg)
☑	Vehicle			VEH_SMCAR			592
☑	Fuel Converter	ic	si	FC_SI41_emis	41	0.34	131
☑	Exhaust Aftertreat			EX_SI			11
					#mod	V nom	
☑	Energy Storage	rint	pb	ESS_PB25	25	308	275
☐	Energy Storage 2			ess 2 options			
☑	Motor			MC_AC75	75	0.92	91
	Motor 2			motor 2 options			
	Starter			starter options			
☐	Generator			gc options			
☑	Transmission	man	man	TX_5SPD		1	114
	Transmission 2			trans 2 options			
	Clutch/Torq. Conv.			clutch/torque converter			
☑	Torque Coupling			TC_DUMMY		1	
☑	Wheel/Axle	Crr	Crr	WH_SMCAR			0
☑	Accessory	Const	Const	ACC_HYBRID			
	Acc Electrical			acc elec options			
☑	Powertrain Control	par	man	PTC_PAR			

Motor Position: pre transmission

Component: fuel_converter

Plot Selection: fc_efficiency

Cargo: 136

Calculated Mass: 1350

☐ override mass: 1

View Block Diagram: BD_PAR

Variable
Component: fuel_converter
Variables: fc_acc_mass
32.8056

Advisor

Information from AVL:

- The U.S. Department of Energy's National Renewable Energy Laboratory (NREL) first developed ADVISOR in 1994.
- Between 1998 and 2003 it was downloaded by more than 7,000 individuals, corporations, and universities world-wide.
- In early 2003 NREL initiated the commercialisation of ADVISOR through a public solicitation.
- AVL responded and was awarded the exclusive rights to license and distribute ADVISOR world-wide.