



Thermochemistry, Air-Fuel Properties, and Mixture Preparation in SI Engine

Nazaruddin Sinaga

Efficiency and Energy Conservation Laboratory

Diponegoro University

Combustion Stoichiometry

Air: Oxygen 21%, Nitrogen (nitrogen + argon) 79%

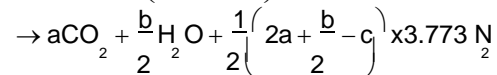
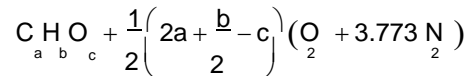
Fuel: Hydrocarbons (C_aH_b), oxygenates ($C_aH_bO_c$)

Examples:

		<u>LHV</u>
Gasoline	$C_nH_{1.87n}$	44 MJ/kg
Diesel fuel	$C_nH_{1.75n}$	43 MJ/kg
Natural gas (mostly methane)	$CH_{3.8}$	45 MJ/kg
Coal	$C_nH_{0.8n}$	30 MJ/kg
Methanol	CH_3OH	20 MJ/kg
Ethanol	C_2H_5OH	26 MJ/kg

(LHV = Energy released per unit mass of fuel without recovery of the heat of vaporization of the water vapor in the combustion products)

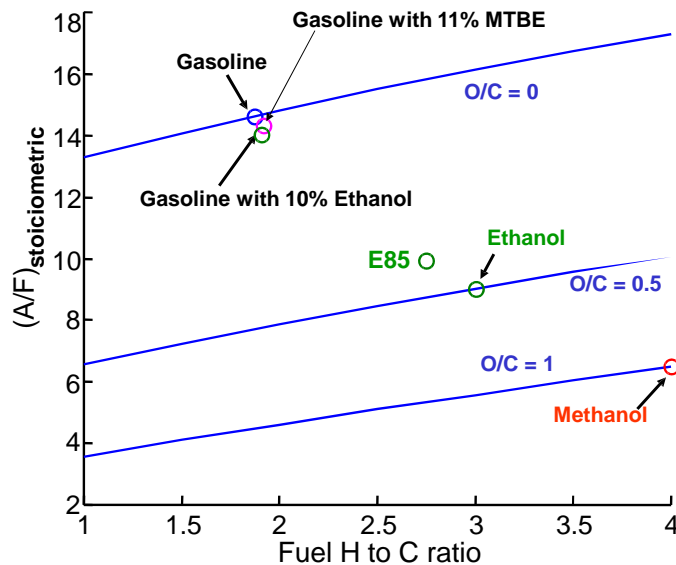
Stoichiometric Combustion



For typical petroleum based fuel ($c=0$):

$$(A/F)_{\text{stoich}} \sim 14.6 \text{ (range 14.2 to 15)}$$

Stoichiometric requirement for different fuels



Lean and rich combustion

Fuel-lean combustion

- major products: CO_2 , H_2O , O_2 , N_2
- minor products: HC , CO , H_2 , NO

Fuel-rich combustion

- major products: CO_2 , H_2O , CO , H_2 , N_2
- minor products: HC , O_2 , NO

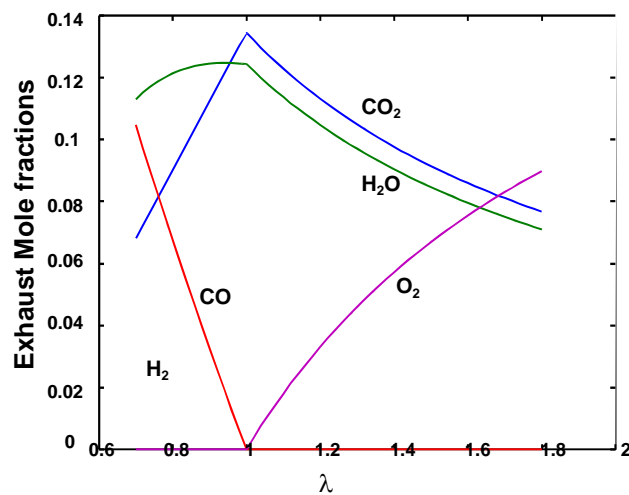
Equivalence ratio: Normalized A/F or F/A ratios:

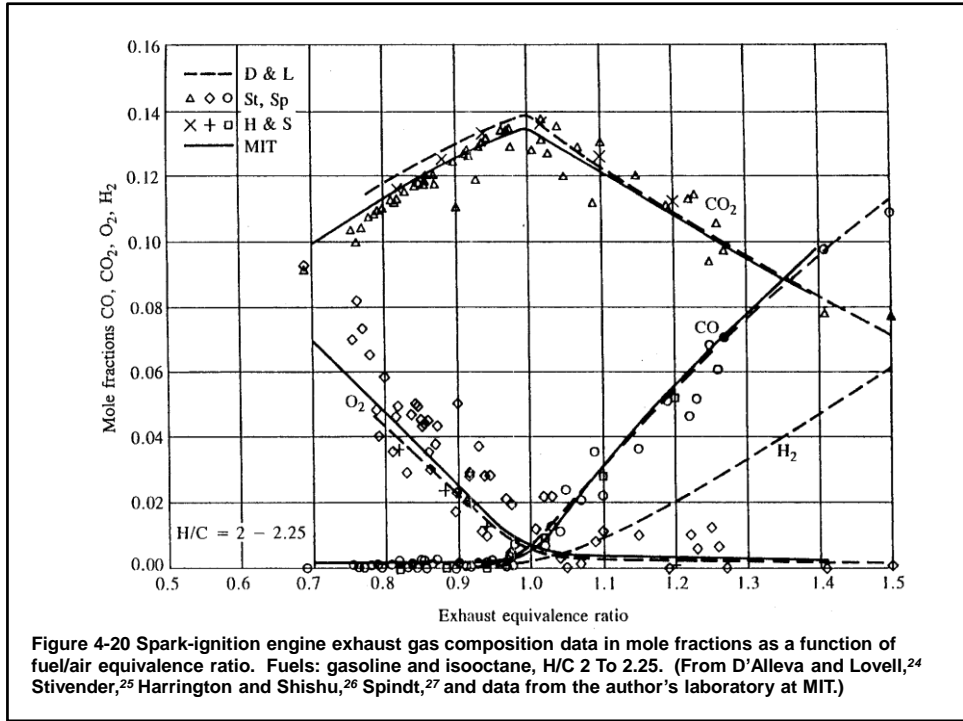
$$\Phi = \frac{F/A}{(F/A)_{\text{stoichiometric}}}$$

$$\lambda = \frac{A/F}{(A/F)_{\text{stoichiometric}}}$$

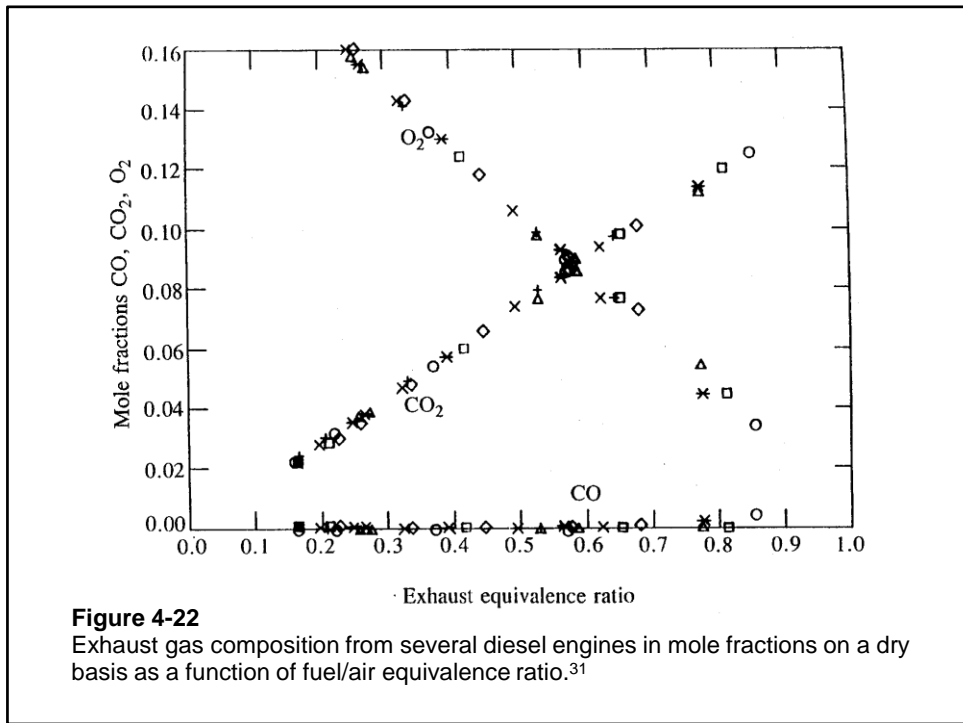
$$\lambda = \frac{1}{\Phi}$$

Exhaust composition (fuel $\text{CH}_{1.85}$)



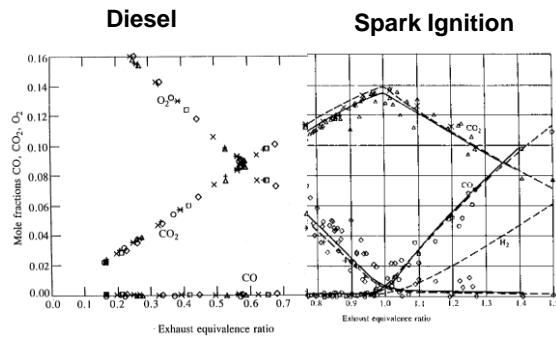


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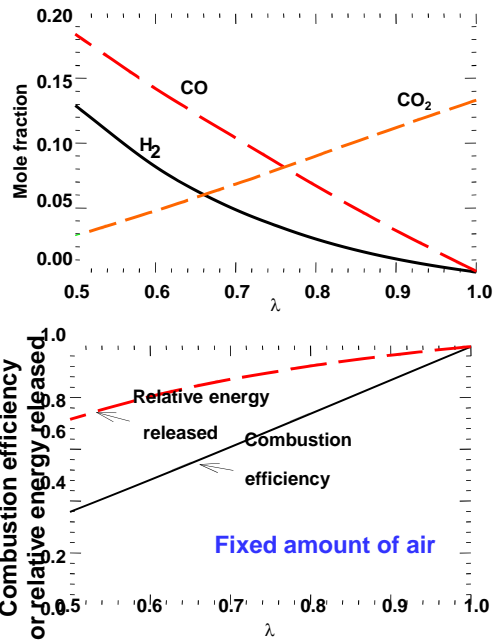
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Φ dependence of exhaust major species



Superposition of Figures 4-20 and 4-22

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Gasoline fuel-rich combustion

For fuel rich combustion, empirically

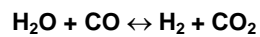
$$\frac{[H_2O][CO]}{[H_2][CO_2]} = 3.5 \text{ to } 3.7$$

$$\frac{[H_2][CO_2]}{[H_2O][CO]}$$

where [] denotes molar concentration

Value corresponds to

equilibrium composition of Water-Gas reaction at ~ 1740°K



Equilibrium combustion products: Dissociation effects

P=30 atmospheres

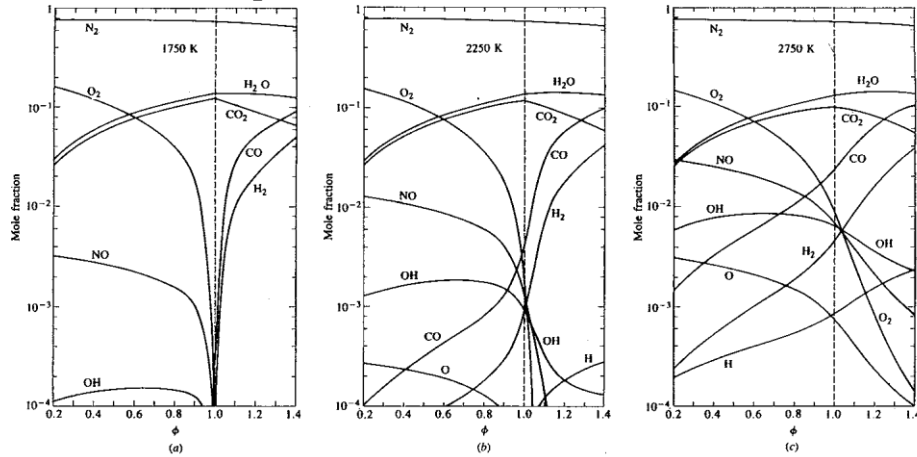


FIGURE 3-10
Mole fractions of equilibrium combustion products of iso-octane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (a) 1750 K; (b) 2250 K; and (c) 2750 K.

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Equilibrium combustion products: Dissociation effects

P=30 atmospheres

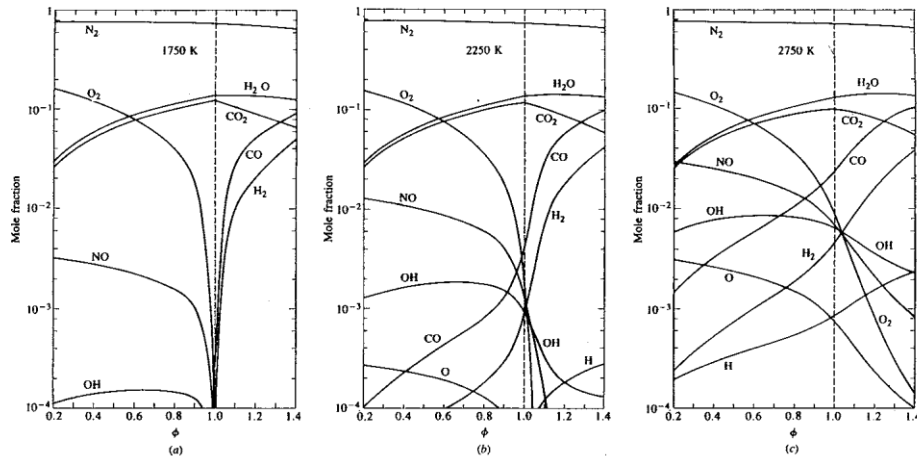


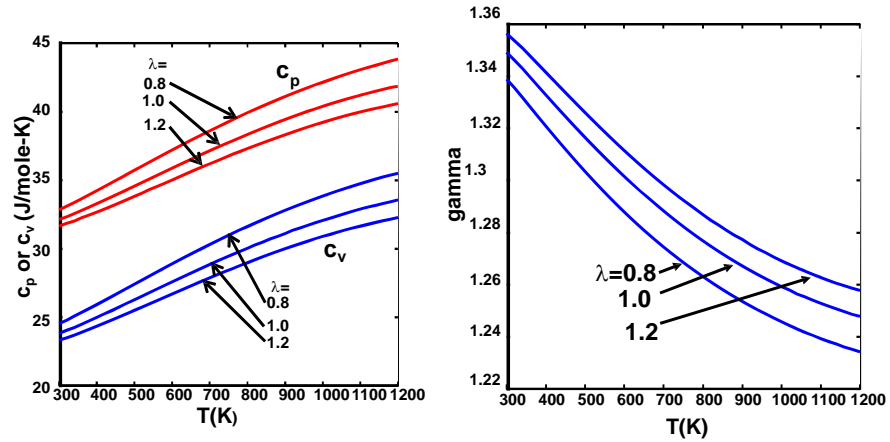
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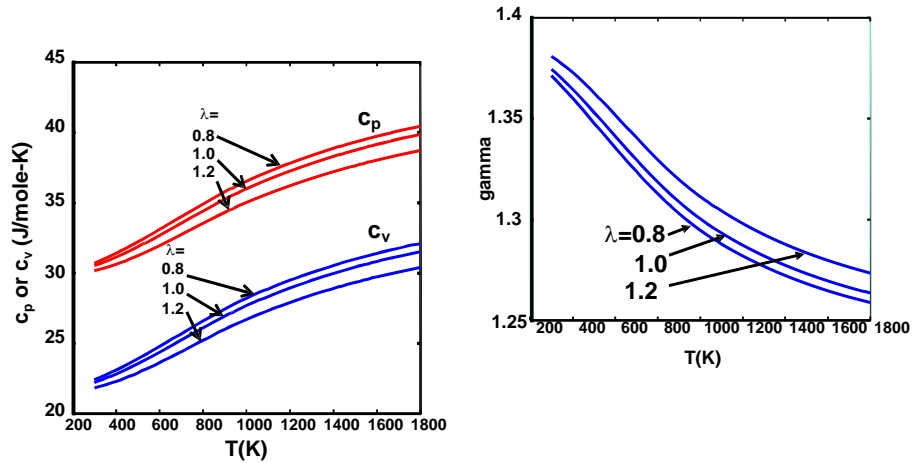
Thermodynamic model of engine charge for heat release process

- Unburned gas
 - Ideal gas of frozen composition
- Burned gas
 - At high temperature ($T > 1740\text{K}$), as equilibrium mixture
 - At low temperature ($T < 1740\text{K}$), as frozen mixture

Unburned gas properties for gasoline ($\text{CH}_{1.85}$)/air



Burned gas properties for gasoline ($\text{CH}_{1.85}$)/air



Composition frozen at 1740K

Fuel-air cycle results

In the Fuel-Air Cycle, the engine processes are still modeled as ideal but the properties of the working fluid (fuel/air/residual gas mixture before combustion, and burned gases in chemical equilibrium after combustion) are described accurately.

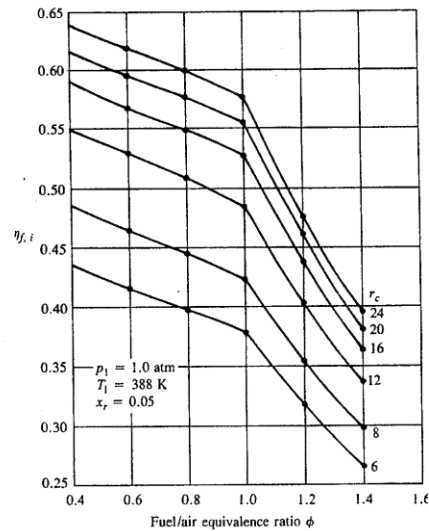
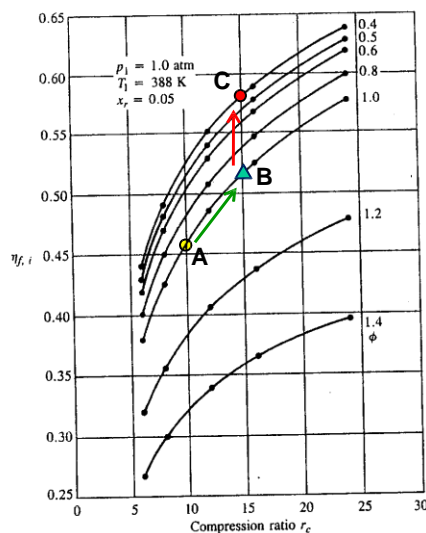
The results from this improved cycle analysis model are useful for estimating, approximately, the effects of compression ratio, fuel/air equivalence ratio, and mixture inlet conditions on engine efficiency and performance. The following approximate relationships are useful.

1. The maximum indicated fuel conversion efficiency of an actual engine is about 0.85 times the efficiency of the equivalent fuel-air cycle.
2. Results from change of engine operating condition can be interpreted in terms of percentage change in output values

Computer codes which accurately simulate the real engine cycle have now been developed and are widely used.

Fuel-air cycle results: $\eta_{f,i}$

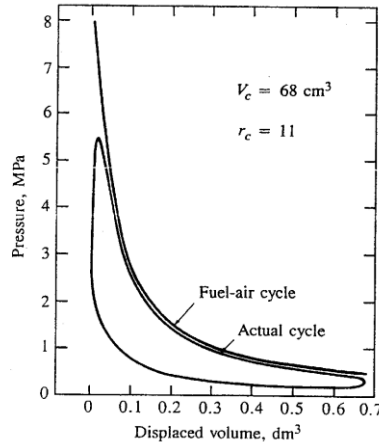
Fuel: octene; $p_1 = 1 \text{ atm}$, $T_1 = 388 \text{ K}$, $x_r = 0.05$ (Fig. 5.9)



A=SI engine at stoichiometric with $r_c=10$; C=Diesel at $A/F=36$ ($\phi=0.4$) with $r_c=15$

Real Cycle Effects

1. **Combustion efficiency** $\eta_c = 1 - \frac{\text{exhaust chemical energy as CO, H}_2, \text{HC, soot}}{\text{chemical energy in inducted fuel}}$
2. **Heat loss, finite combustion time, actual valve timing**

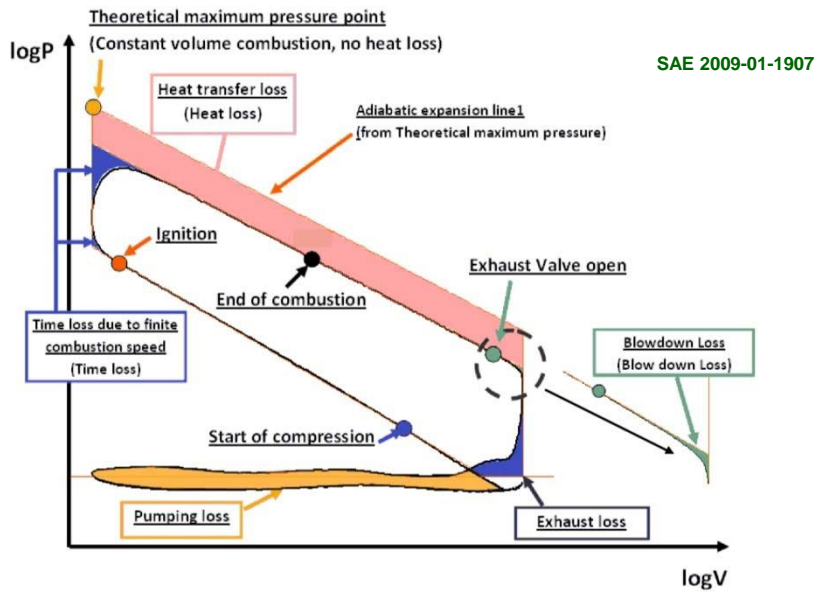


SI engine:
 H₂ and CO ~ 1 to 2% of fuel energy
 HC ~ 1% of fuel energy
 $\eta_c \sim 97-98\%$
 Diesel engine
 Very little unburned gas
 $\eta_c \sim 99\%$

Fig. 5-18
 Pressure-volume diagram for actual SI engine compared with that for equivalent fuel-air cycle; $r_c = 11$.

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Deconstruction of cycle losses

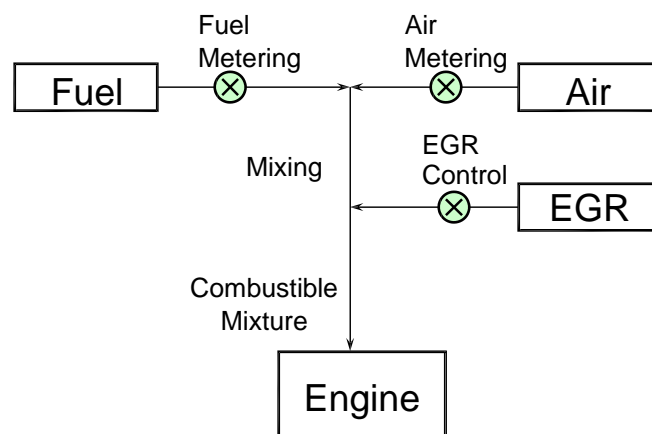


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SI Engine Mixture Preparation

1. Requirements
2. Fuel metering systems
3. Fuel transport phenomena
4. Mixture preparation during engine transients
5. The Gasoline Direct Injection engine

MIXTURE PREPARATION



MIXTURE PREPARATION

Parameters

- Fuel Properties
- Air/Fuel Ratio
- Residual/Exhaust Gas Fraction



Impact

- Driveability
- Emissions
- Fuel Economy

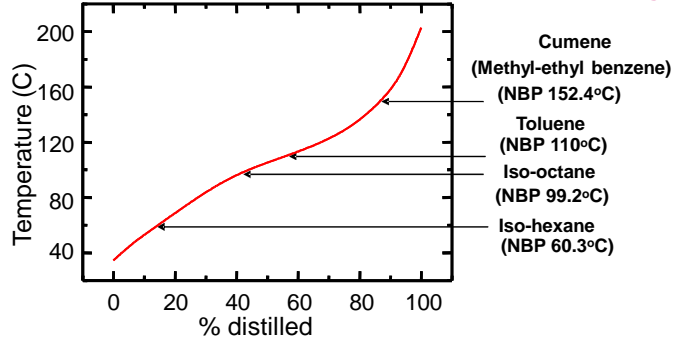
Other issues: Knock, exhaust temperature, starting and warm-up, acceleration/ deceleration transients

Fuel properties (Table D4 of text book)

Fuel	Formula (phase)	Molecular weight	Specific gravity: (density,] kg/m ³)	Heat of vaporization, kJ/kg]	Specific heat		Higher heating value, MJ/kg	Lower heating value, MJ/kg	LHV of stoich. mixture, MJ/kg	A/F _P	P/A ₁	Fuel octane rating	
					Liquid, kJ/kg · K	Vapor c _p , kJ/kg · K						RON	MON
Practical fuels													
Gasoline	C ₈ H _{18.7} (l)	~110	0.72-0.78	350	2.4	~1.7	47.3	44.0	28.3	14.6	0.0685	91-99	82-89
Light diesel	C ₁₂ H _{24.4} (l)	~170	0.78-0.84	370	2.2	~1.7	46.1	43.2	2.79	14.3	0.0690	—	—
Heavy diesel	C ₁₆ H _{34.1} (l)	~200	0.82-0.88	230	1.9	~1.7	45.5	42.8	2.85	14.4	0.0697	—	—
Natural gas	C ₁ H _{1.8-2.4} (g)	~18	(~0.79)	—	—	~2	50	45	2.9	14.5	0.069	—	—
Pure hydrocarbons													
Methane	CH ₄ (g)	16.04	(0.72)	509	0.63	2.2	55.5	50.0	2.72	17.23	0.0580	120	120
Propane	C ₃ H ₈ (g)	44.10	0.51 (2.0)	426	2.5	1.6	50.4	46.4	2.75	15.67	0.0638	112	97
Isooctane	C ₈ H ₁₈ (l)	114.23	0.692	308	2.1	1.63	47.8	44.3	2.75	15.13	0.0661	100	100
Cetane	C ₁₆ H ₃₄ (l)	226.44	0.773	358	—	1.6	47.3	44.0	2.78	14.82	0.0675	—	—
Benzene	C ₆ H ₆ (l)	78.11	0.879	433	1.72	1.1	41.9	40.2	2.82	13.27	0.0753	—	115
Toluene	C ₇ H ₈ (l)	92.14	0.867	412	1.68	1.1	42.5	40.6	2.79	13.50	0.0741	120	109
Alcohols													
Methanol	CH ₃ OH(l)	32.04	0.792	1103	2.6	1.72	22.7	20.0	2.68	6.47	0.155	106	92
Ethanol	C ₂ H ₅ OH(l)	46.07	0.785	840	2.5	1.93	29.7	26.9	2.69	9.00	0.111	107	89
Other fuels													
Carbon	C(s)	12.01	~2]	—	—	—	33.8	33.8	2.70	11.51	0.0669	—	—
Carbon monoxide	CO(g)	28.01	(1.25)	—	—	1.05	10.1	10.1	2.91	2.467	0.405	—	—
Hydrogen	H ₂ (g)	2.015	(0.090)	—	—	1.44	142.0	120.0	3.40	34.3	0.0292	—	—

Gasoline evaporative characteristics

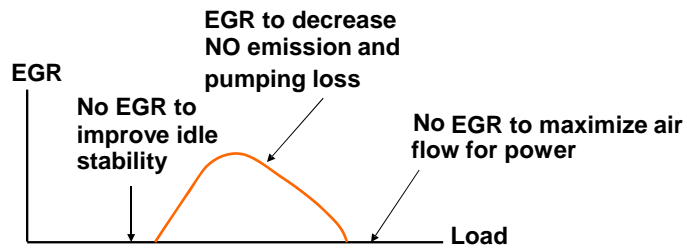
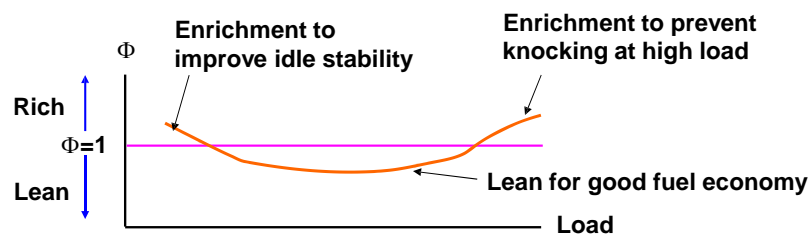
Distillation curve (ASTM D86) of UTG91 (a calibration gasoline)



- Reid Vapor pressure (ASTM D323):
 - equilibrium pressure of fuel and air of 4 x liquid fuel volume at 37.8°C
- T_{10} , T_{50} , T_{90}
 - Temperature at 10, 50 and 90% distillation points
- Driveability Index (DI)
 - For hydrocarbon fuels: $DI = 1.5 T_{10} + 3 T_{50} + T_{90}$ (T in °F)

RVP: winter gasoline ~ 11 psi (0.75 bar); Summer gasoline ~ 9 psi (0.61 bar); California Phase 2 fuel = 7 psi (0.48 bar)
 DI: range from 1100 to 1300; Phase II calibration gasoline has DI=1115; High DI calibration fuel has DI 1275.

Equivalence ratio and EGR strategies (No emissions constrain)



Requirement for the 3-way catalyst

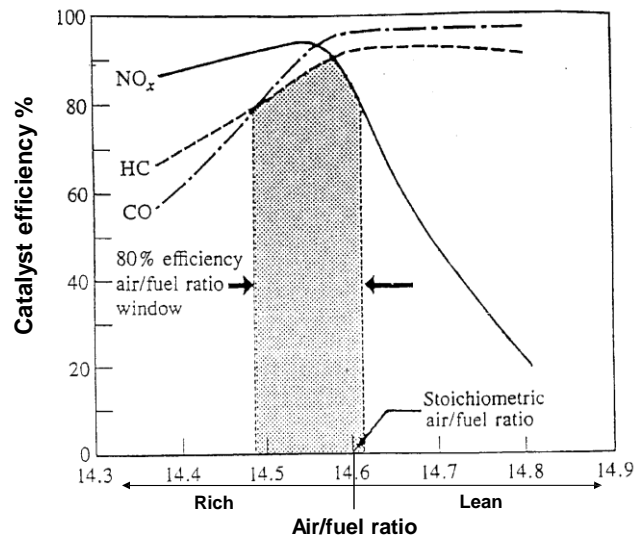


Fig 11-57

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FUEL METERING

- Carburetor
 - A/F not easily controlled
- Fuel Injection
 - Electronically controlled fuel metering
 - ↗ Throttle body injection
 - ↗ Port fuel injection
 - ↗ Direct injection

Injectors

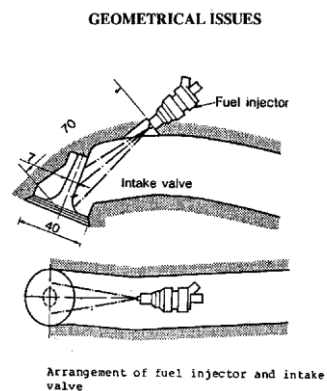
PFI injectors

- Single 2-, 4-,..., up to 12-holes
- Injection pressure 3 to 7 bar
- Droplet size:
 - Normal injectors: 200 to 80 μm
 - Flash Boiling Injectors: down to 20 μm
 - Air-assist injectors: down to 20 μm

GDI injectors

- Shaped-spray
- Injection pressure 50 to 250 bar
- Drop size: 10 to 50 μm

PFI Injector targeting



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INTAKE PORT THERMAL ENVIRONMENT

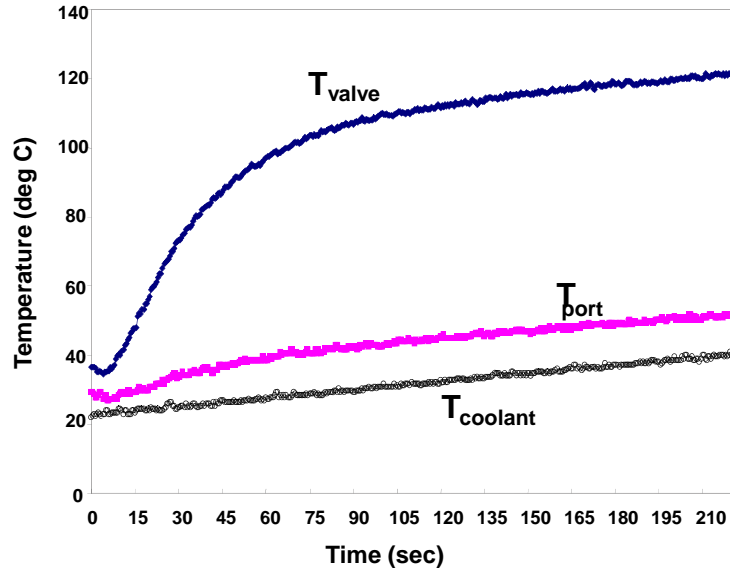
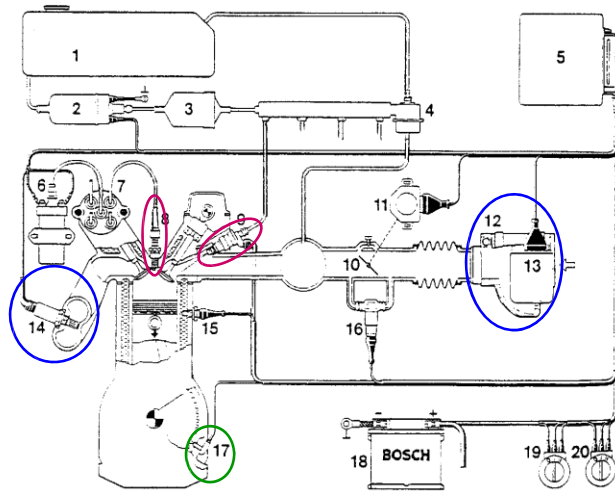


Diagram of a typical Motronic system.

1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 Pressure regulator, 5 Electronic control unit, 6 Ignition coil, 7 High-voltage distributor, 8 Spark plug, 9 Injection valve, 10 Throttle valve, 11 Trottle-valve switch, 12 Air-flow sensor, 13 Air temperature sensor, 14 Lambda sensor, 15 Engine temperature sensor, 16 Idle-speed actuator, 17 Reference-mark and engine-speed sensor, 18 Battery, 19 Ignition/starter switch, 20 A/C switch.



Engine management system

From Bosch Automotive Handbook

Fuel Metering

- **A/F ratio measured by λ sensor (closed loop operation)**
 - feedback on fuel amount to keep $\lambda=1$
- **Feed-forward control (transients):**
 - To meter the correct fuel flow for the targeted A/F target, need to know the air flow
- **Determination of air flow (need transient correction)**
 - Air flow sensor (hot film sensor)
 - Speed density method
 - ↗ Determine air flow rate from MAP (P) and ambient temperature (T_a) using volumetric efficiency (η_v) calibration

$$\dot{m}_a = \rho \frac{V_D}{2} \frac{N}{v} \eta$$

Displacement vol. V_D ,

$$\rho = \frac{P}{RT} (N, \rho)$$

gas constant R

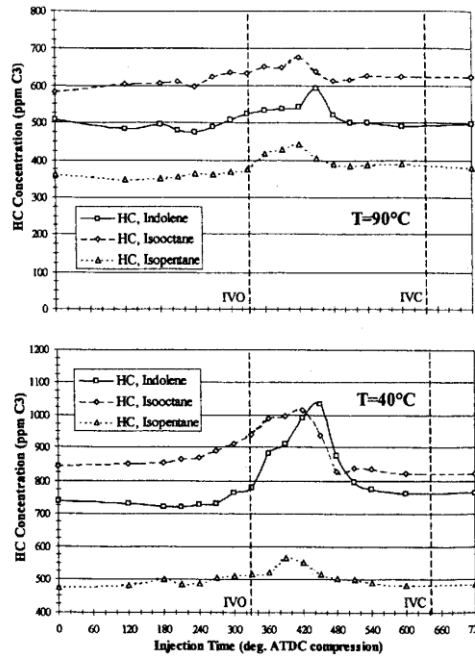
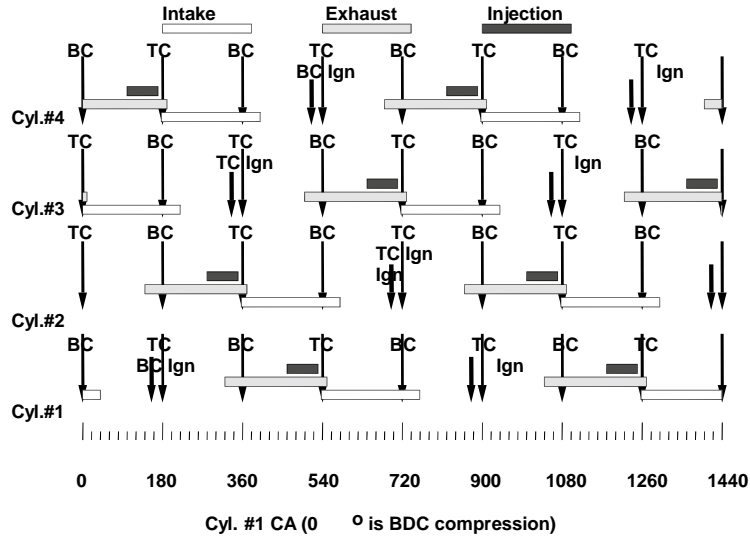
rev. per second N,

a

FEATURES OF ELECTRONICALLY CONTROLLED FUEL INJECTION SYSTEM

- Sensors
 - Air temperature
 - Engine Speed
 - Manifold air pressure (MAP) / air flow rate
 - Exhaust air/ fuel equivalence ratio (Λ): EGO (and UEGO)
 - Coolant temperature
 - Throttle position and throttle movement rate
 - Crank and cam positions
- Controls
 - Injection duration
 - Spark timing
 - Other functions
 - ↗ Idle air, carbon canister venting, cold start management, transient compensation,

ENGINE EVENTS DIAGRAM



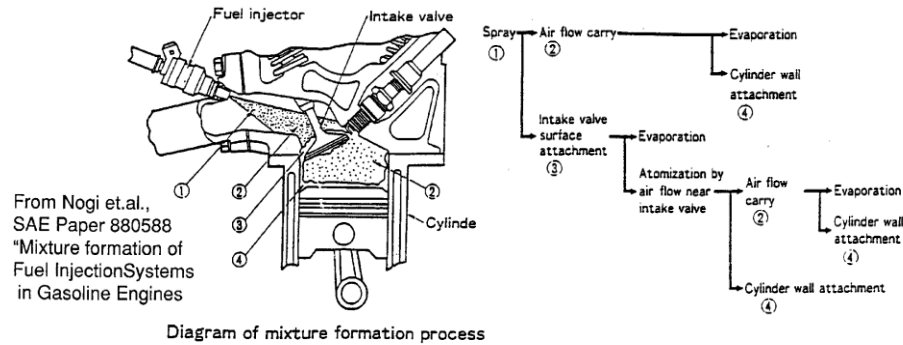
Effect of Injection Timing on HC Emissions

SAE Paper 972981
Stache and Alkidas

Engine at 1300 rpm
275 kPa BMEP

Injection timing refers to start of injection

Mixture Preparation in PFI engine



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Intake flow phenomena in mixture preparation (At low to moderate speed and load range)

Reverse Blow-down Flow

- **IVO to EVC:**
 - Burned gas flows from exhaust port because $P_e > P_i$
- **EVC to $P_c = P_i$:**
 - Burned gas flows from cylinder into intake system until cylinder and intake pressure equalize

Forward Flow

- **$P_c = P_i$ to BC:**
 - Forward flow from intake system to cylinder induced by downward piston motion

Reverse Displacement Flow

- **BC to IVC:**
 - Fuel, air and residual gas mixture flows from cylinder into intake due to upward piston motion

Note that the reverse flow affects the mixture preparation process in engines with port fuel injection

Mixture Preparation in Engine Transients

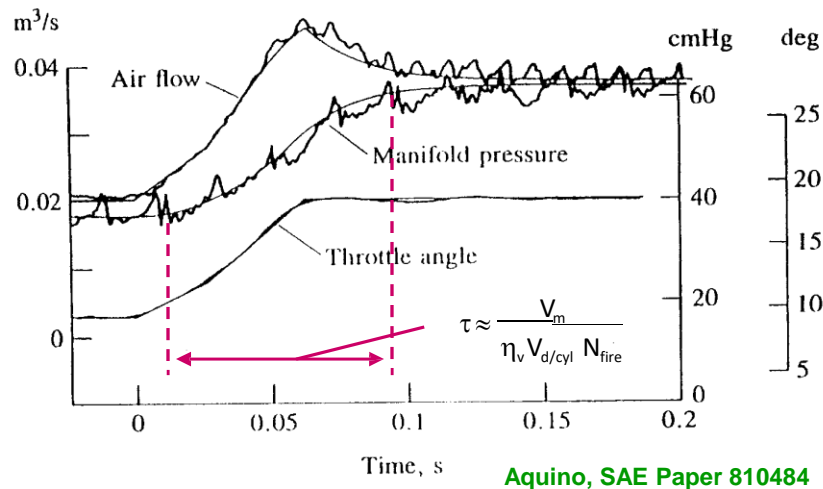
Engine Transients

- Throttle Transients
 - Accelerations and decelerations
- Starting and warm-up behaviors
 - Engine under cold conditions

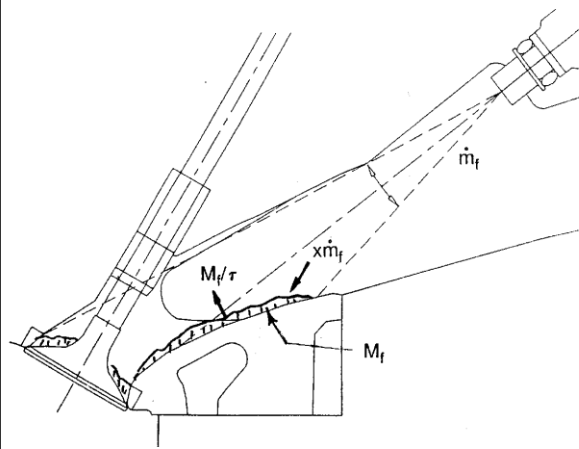
Transients need special compensations because:

- Sensors do not follow actual air delivery into cylinder
- Fuel injected for a cycle is not what constitutes the combustible mixture for that cycle

Manifold pressure charging in throttle transient



Fuel-Lag in Throttle Transient



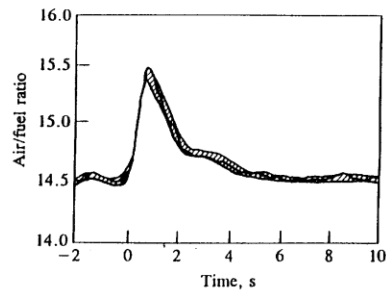
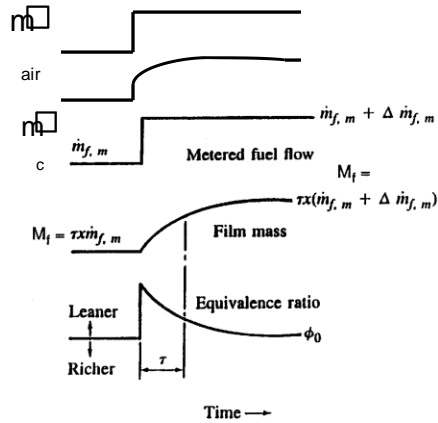
The x - τ Model

$$\frac{dM_f}{dt} = x\dot{m}_f - \frac{M_f}{\tau}$$

$$\dot{m}_c = (1-x)\dot{m}_f + \frac{M_f}{\tau}$$

\dot{m}_f = Injected fuel flow rate
 \dot{m}_c = Fuel delivery rate to cylinder
 M_f = Fuel mass in puddle

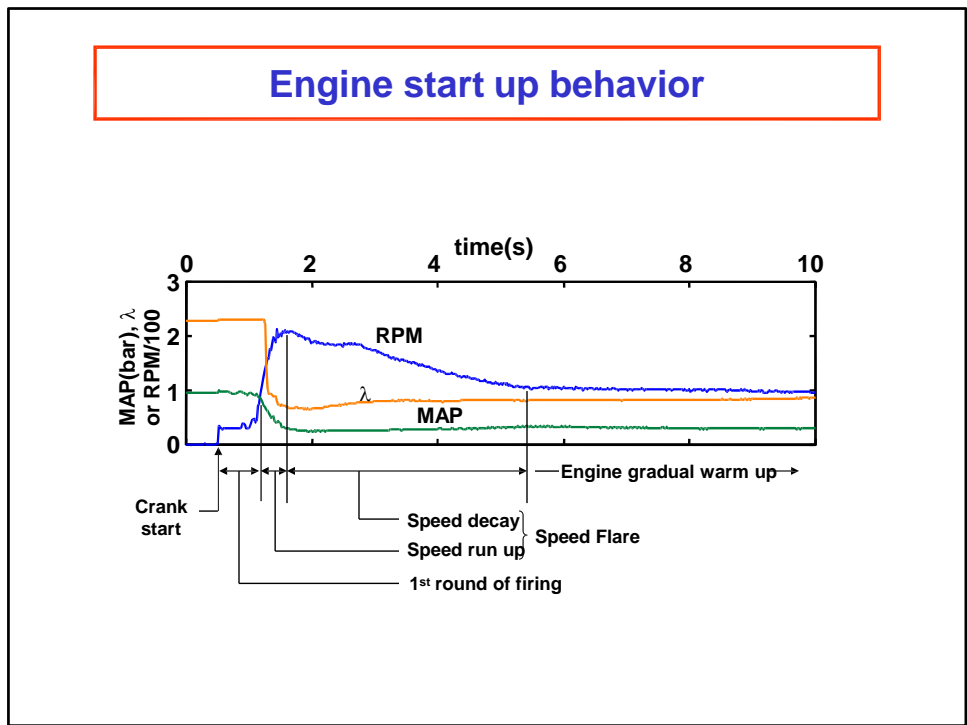
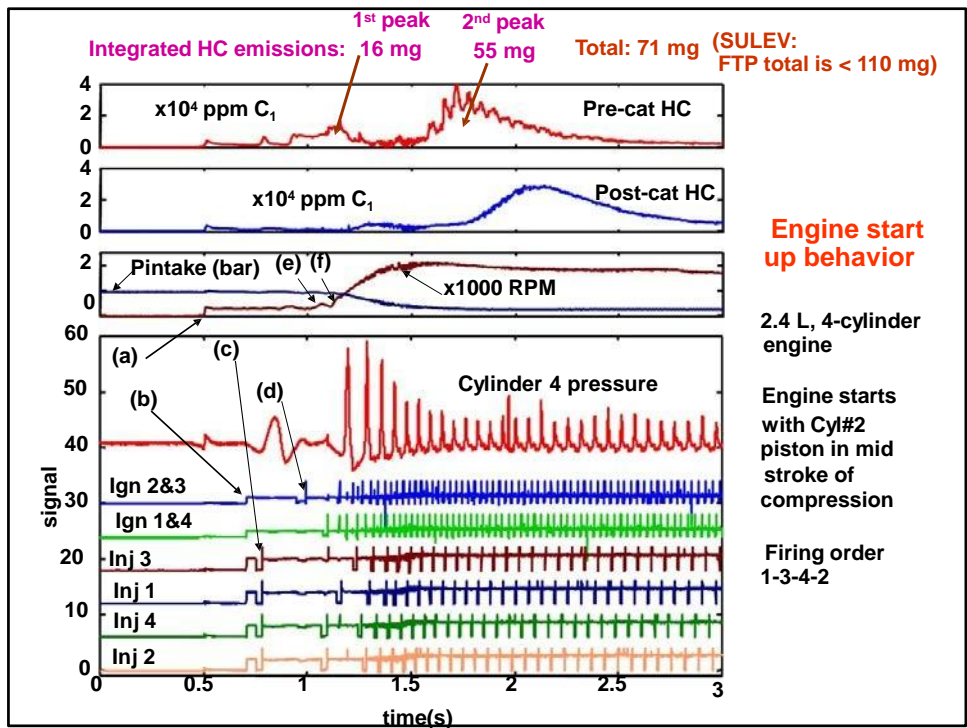
Fuel transient in throttle opening



Model prediction

Observed results

Fig 7-28
 Uncompensated A/F behavior in throttle transient

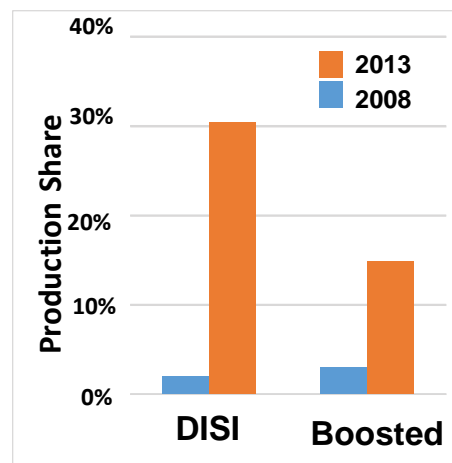


Pertinent Features of DISI Engines

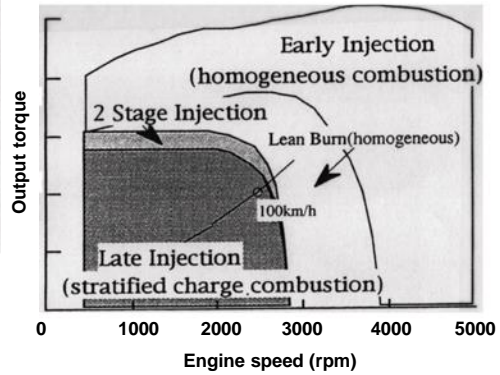
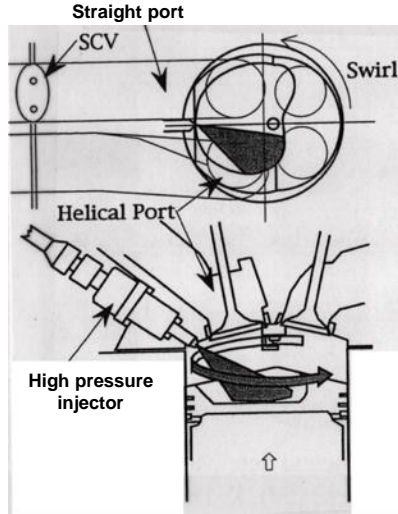
1. Precise metering of fuel into cylinder
 - Engine calibration benefit: better driveability and emissions
2. Opportunity of running stratified lean at part load
 - Fuel economy benefit (reduced pumping work; lower charge temperature, lower heat transfer; better thermodynamic efficiency)
3. Charge cooling by fuel evaporation
 - Gain in volumetric efficiency
 - Gain in knock margin (could then raise compression ratio for better fuel economy)
 - Both factors increase engine output

DISI technology penetration

- Significant market penetration of DISI
 - Homogeneous charge configuration
 - As enabler of the boosted-downsizing strategy

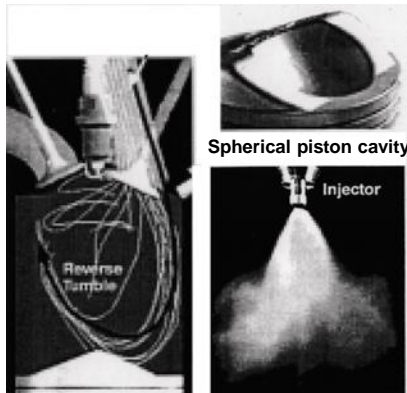


Toyota DISI Engine (SAE Paper 970540)



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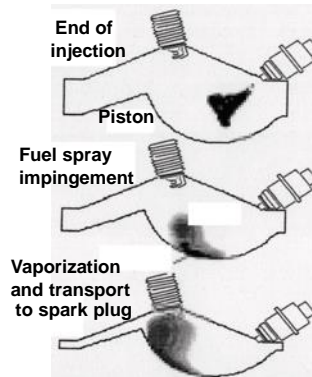
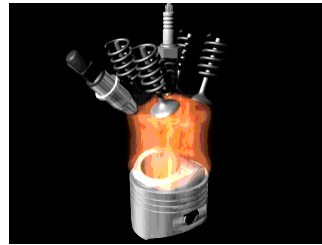
Mitsubishi DISI Engine



Reverse tumble

Swirling spray

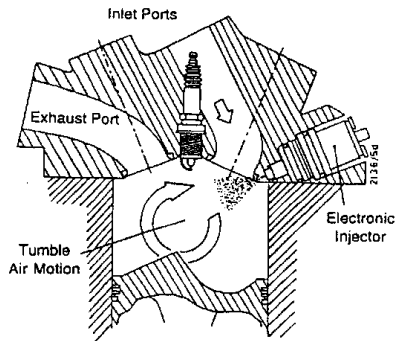
(SAE 960600)



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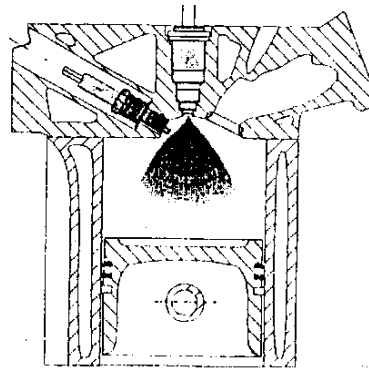
Wall-guided versus spray-guided injection

Wall-guided injection
(injector relatively distant from spark plug)



SAE 970543 (Ricardo)

Spray-guided injection
(injector relatively close to spark plug)

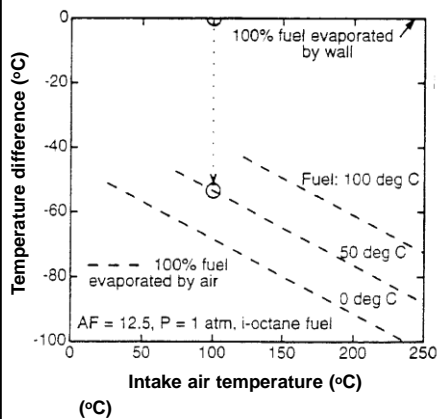


SAE 970624 (Mercedes-Benz)

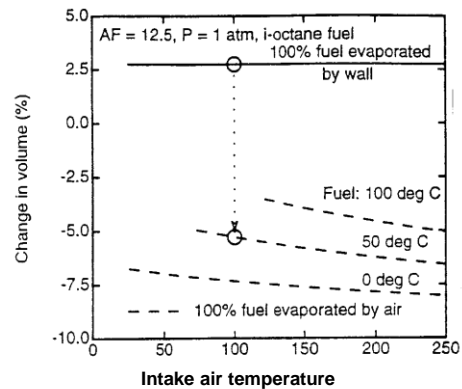
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Charge cooling by in-air fuel evaporation

Charge cooling effect



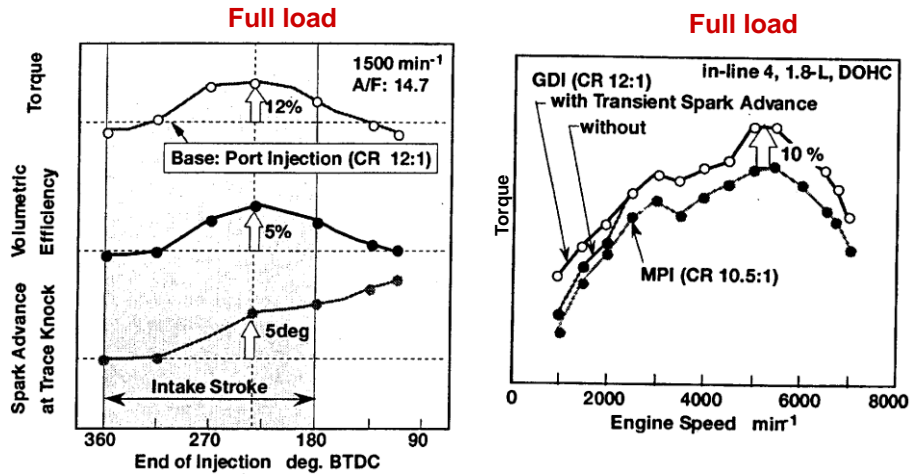
Lowering of intake volume



Anderson, Yang, Brehob, Vallance, and Whiteabker, SAE Paper 962018

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Full load performance benefit



SAE 970541 (Mitsubishi)

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Part load fuel economy gain

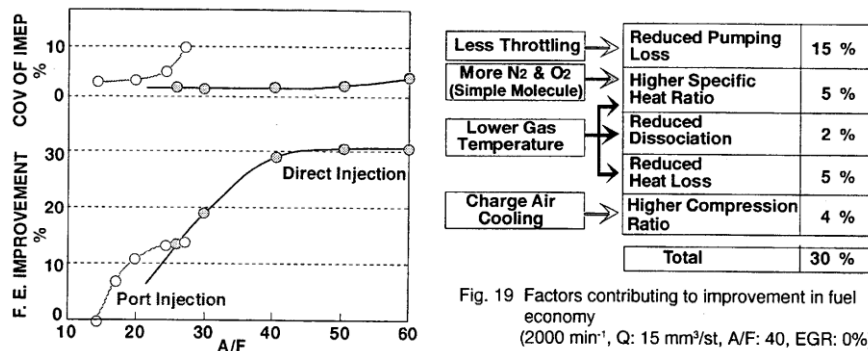


Fig. 18 Improvement in fuel economy by late direct injection (2000 min⁻¹, Q: 15 mm³/st)

Fig. 19 Factors contributing to improvement in fuel economy (2000 min⁻¹, Q: 15 mm³/st, A/F: 40, EGR: 0%)

SAE Paper 960600 (Mitsubishi)

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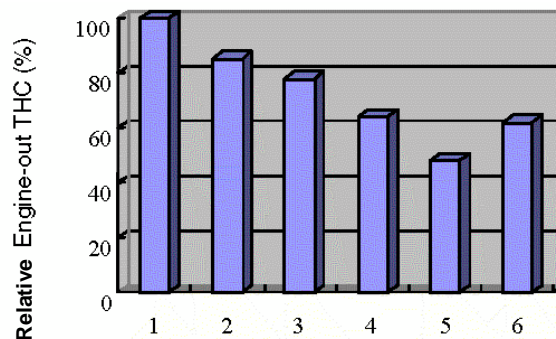
DISI Challenges

1. **High cost**
2. **With the part-load stratified-charge concept :**
 - High hydrocarbon emissions at light load
 - Significant NO_x emission, and lean exhaust not amenable to 3-way catalyst operation
3. **Particulate emissions at high load**
4. **Liquid gasoline impinging on combustion chamber walls**
 - Hydrocarbon source
 - Lubrication problem
5. **Injector deposit**
 - Special fuel additive needed for injector cleaning
6. **Cold start behavior**
 - Insufficient fuel injection pressure
 - Wall wetting

Comparison of cold start HC emissions

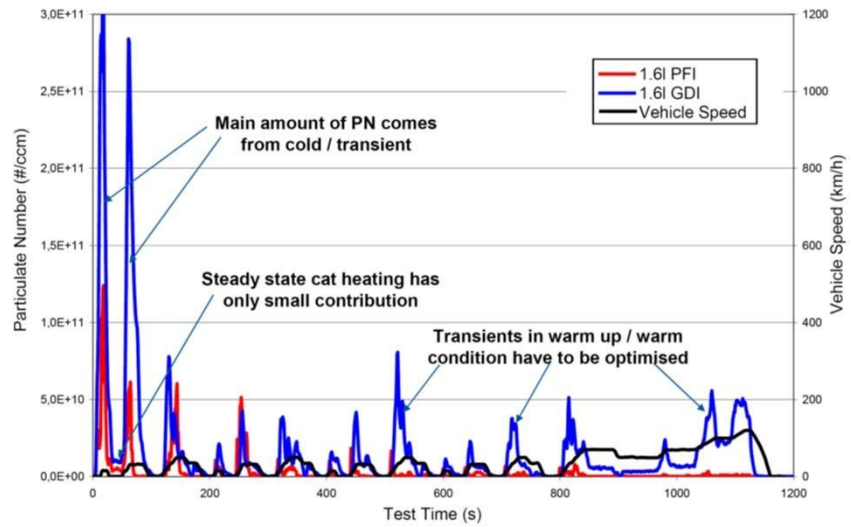
(Koga, Miyashita, Takeda, and Imatake, SAE Paper 2001-01-0969)

Cumulative engine out HC in the first 10 seconds of cold-start



- | | |
|------------------------------|------------------------------|
| 1: Conventional DISI | 4. 2 + late injection |
| 2. Engine Starting W/5MPa | 5: 4 + heated fuel injection |
| 3. Late intake valve opening | 6: MPI |

Significant particle numbers in cold start



SAE 2011-01-1219

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