Diesel injection, ignition, and fuel air mixing

1. Fuel spray phenomena
2. Spontaneous ignition
3. Effects of fuel jet and charge motion on mixing-controlled combustion
4. Fuel injection hardware
5. Challenges for diesel combustion

DIESEL FUEL INJECTION

The fuel spray serves multiple purposes:

- Atomization
- Fuel distribution
- Fuel/air mixing

Typical Diesel fuel injector

- Injection pressure: 1000 to 2200 bar
- 5 to 20 holes at ~ 0.12 - 0.2 mm diameter
- Drop size 0.1 to 10 μm
- For best torque, injection starts at about 20^o BTDC

Injection strategies for NOx control

- Late injection (inj. starts at around TDC)
- Other control strategies:
    - Pilot and multiple injections, rate shaping, water emulsion

Diesel Fuel Injection System

(A Major cost of the diesel engine)

- Performs fuel metering
- Provides high injection pressure
- Distributes fuel effectively
  - Spray patterns, atomization etc.
- Provides fluid kinetic energy for charge mixing

Typical systems:

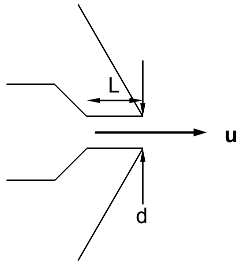
- Pump and distribution system (100 to 1500 bar)
- Common rail system (1000 to 1800 bar)
- Hydraulic pressure amplification
- Unit injectors (1000 to 2200 bar)
- Piezoelectric injectors (1800 bar)
- Electronically controlled

EXAMPLE OF DIESEL INJECTION

(Hino K13C, 6 cylinder, 12.9 L turbo-charged diesel engine, rated at 294KW@2000 rpm)

- Injection pressure = 1400 bar; duration = 40^{o}CA
- BSFC 200 g/KW-hr
- Fuel delivered per cylinder per injection at rated condition
  - 0.163 gm ~0.21 cc (210 mm^3)
- Averaged fuel flow rate during injection
  - 64 mm^3/ms
- 8 nozzle holes, at 0.2 mm diameter
  - Average exit velocity at nozzle ~253 m/s

Typical physical quantities in nozzle flow



- Diesel fuel @ 100°C
  - s.g. ~ 0.78, μ~5×10^{-4} N-s/m^2
- Nozzle diameter ~0.2 mm
- L/d ~ 5 to 10
- Reynolds No. ~ 10^5 (turbulent)
- Pressure drop in nozzle

  ~30 bar << driving pressure
  (~1000 bar)
- Injection velocity

$$u \approx \sqrt{\frac{2\Delta P}{\rho_{fuel}}} \approx 500 \text{ m/s @ } \Delta P \text{ of } 1000 \text{ bar}$$

Fuel Atomization Process

- Liquid break up governed by balance between aerodynamic force and surface tension

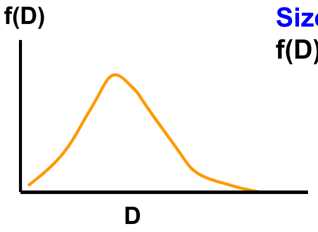
$$\text{Webber Number } (W_b) = \frac{\rho_{gas} u^2 d}{\sigma}$$

- Critical Webber number: $W_{b,critical}$ ~ 30; diesel fuel surface tension ~ 2.5×10^{-2} N/m

- Typical W_b at nozzle outlet > $W_{b,critical}$; fuel shattered into droplets within ~ one nozzle diameter

- Droplet size distribution in spray depends on further droplet breakup, coalescence and evaporation

Droplet size distribution



Size distribution:

f(D)dD = probability of finding particle with diameter in the range of (D, D + dD)

$$1 = \int_0^\infty f(D)dD$$

Average diameter

$$\overline{D} = \int_0^\infty f(D)\,D\,dD$$

Volume distribution

$$\frac{1}{V}\frac{dV}{dD} = \frac{f(D)\,D^3}{\int_0^\infty f(D)\,D^3 dD}$$

Sauter Mean Diameter (SMD)

$$D_{32} = \frac{\int_0^\infty f(D)\,D^3 dD}{\int_0^\infty f(D)\,D^2 dD}$$

Droplet Size Distribution

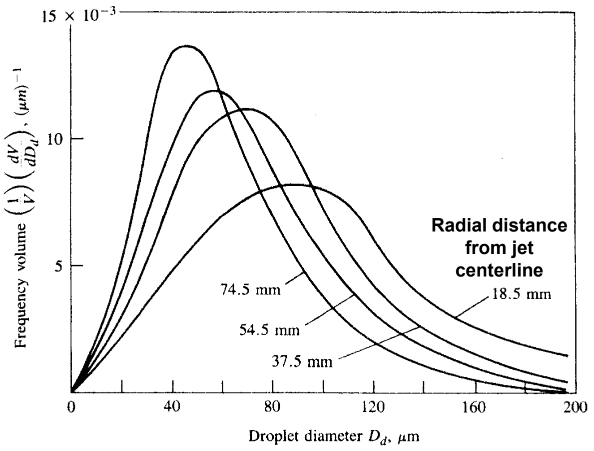
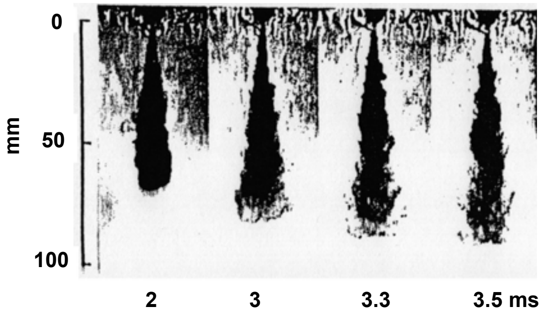


Fig. 10.28 Droplet size distribution measured well downstream; numbers on the curves are radial distances from jet axis. Nozzle opening pressure at 10 MPa; injection into air at 11 bar.

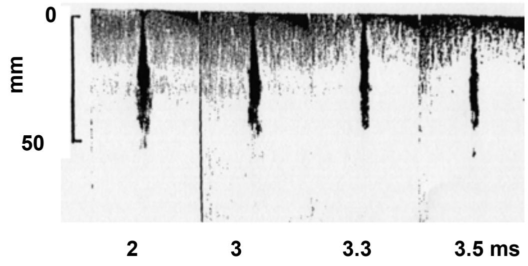
Droplet Behavior in Spray

- Small drops (~ micron size) follow gas stream; large ones do not
  - Relaxation time $\tau \propto d^2$
- Evaporation time $\propto d^2$
  - Evaporation time small once charge is ignited
- Spray angle depends on nozzle geometry and gas density : $\tan(\theta/2) \propto \sqrt{(\rho_{gas}/\rho_{liquid})}$
- Spray penetration depends on injection momentum, mixing with charge air, and droplet evaporation

Spray Penetration: vapor and liquid (Fig. 10-20)



Shadowgraph image showing both liquid and vapor penetration



Back-lit image showing liquid-containing core

Auto-ignition Process

PHYSICAL PROCESSES (Physical Delay)

- Drop atomization
- Evaporation
- Fuel vapor/air mixing

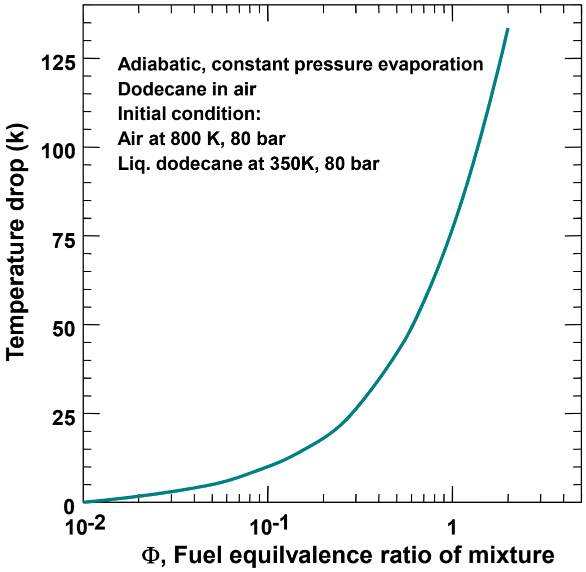
CHEMICAL PROCESSES (Chemical Delay)

- Chain initiation
- Chain propagation
- Branching reactions

CETANE IMPROVERS

- Alkyl Nitrates
  - 0.5% by volume increases CN by ~10

Mixture cooling from heat of vaporization



Adiabatic, constant pressure evaporation
Dodecane in air
Initial condition:
Air at 800 K, 80 bar
Liq. dodecane at 350K, 80 bar

Ignition Mechanism: similar to SI engine knock

CHAIN BRANCHING EXPLOSION

Chemical reactions lead to increasing number of radicals, which leads to rapidly increasing reaction rates

Chain Initiation

$$RH + O_2 \Rightarrow \dot{R} + H\dot{O}_2$$

Chain Propagation

$$\dot{R} + O_2 \Rightarrow R\dot{O}_2, \text{etc.}$$

Formation of Branching Agents

$$R\dot{O}_2 + RH \Rightarrow ROOH + \dot{R}$$

$$R\dot{O}_2 \Rightarrow R'CHO + R''\dot{O}$$

Degenerate Branching

$$ROOH \Rightarrow R\dot{O} + \dot{O}H$$

$$R'CHO + O_2 \Rightarrow R'\dot{C}O + H\dot{O}_2$$

Cetane Rating

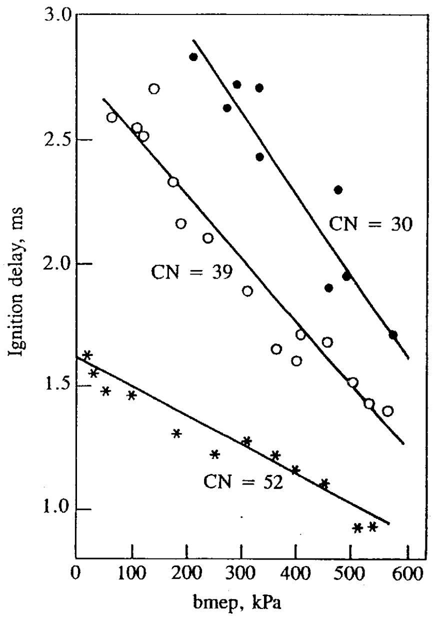
(Procedure is similar to Octane Rating for SI Engine; for details, see10.6.2 of text)

Primary Reference Fuels:

- Normal cetane ($C_{16}H_{34}$): CN = 100
- Hepta-Methyl-Nonane (HMN; $C_{16}H_{34}$): CN = 15
  (2-2-4-4-6-8-8 Heptamethylnonane)

Rating:

- Operate CFR engine at 900 rpm with fuel
- Injection at 13° BTC
- Adjust compression ratio until ignition at TDC
- Replace fuel by reference fuel blend and change blend proportion to get same ignition point
- CN = % n-cetane + 0.15 x % HMN

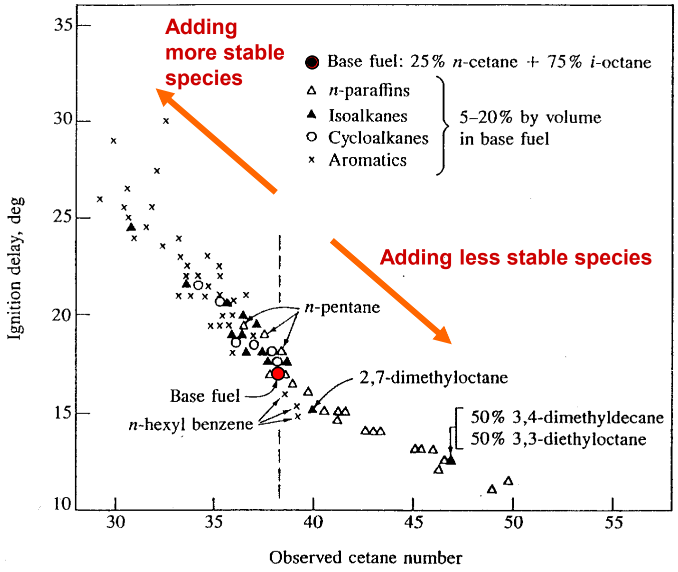


Ignition Delay

Ignition delays measured in a small four-stroke cycle DI diesel engine with r_c=16.5, as a function of load at 1980 rpm, at various cetane number

(Fig. 10-36)

Fuel effects on Cetane Number (Fig. 10-40)



Ignition Delay Calculations

- Difficulty: do not know local conditions (species concentration and temperature) to apply kinetics information

Two practical approaches:

- Use an "instantaneous" delay expression
  $\tau(T,P) = P^{-n}\exp(-E_A/T)$
  and solve ignition delay (τ_{id}) from
  $$1 = \int_{t_{si}}^{t_{si}+\tau_{id}} \frac{1}{\tau(T(t),P(t))}\,dt$$
- Use empirical correlation of τ_{id} based on T, P at an appropriate charge condition; e.g. Eq. (10.37 of text)

$$\tau_{id}(CA) = (0.36 + 0.22\bar{S}_p(m/s))\exp\left[E_A\left(\frac{1}{\tilde{R}T(K)} - \frac{1}{17190}\right) + \left(\frac{21.2}{P(bar) - 12.4}\right)^{0.63}\right]$$

E_A (Joules per mole) = 618,840 / (CN+25)

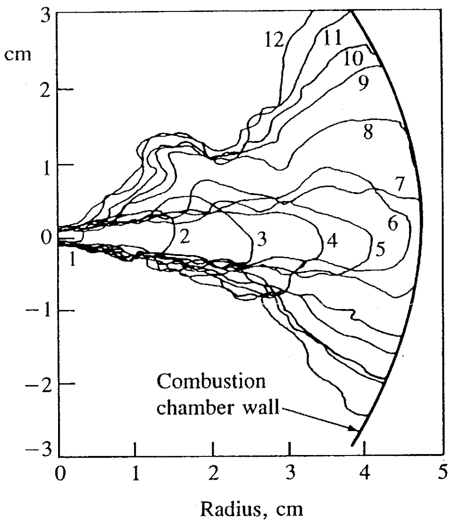
Diesel Engine Combustion
Air Fuel Mixing Process

- Importance of air utilization
  - Smoke-limit A/F ~ 20
- Fuel jet momentum / wall interaction has a larger influence on the early part of the combustion process
- Charge motion impacts the later part of the combustion process (after end-of-injection)

CHARGE MOTION CONTROL

- Intake created motion: swirl, etc.
  - Not effective for low speed large engine
- Piston created motion - squish

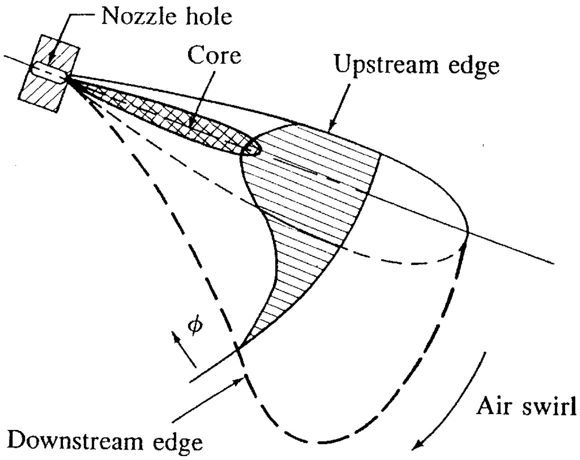
Interaction of fuel jet and the chamber wall



Sketches of outer vapor boundary of diesel fuel spray from 12 successive frames (0.14 ms apart) of high-speed shadowgraph movie. Injection pressure at 60 MPa.

Fig. 10-21

Interaction of fuel jet with air swirl

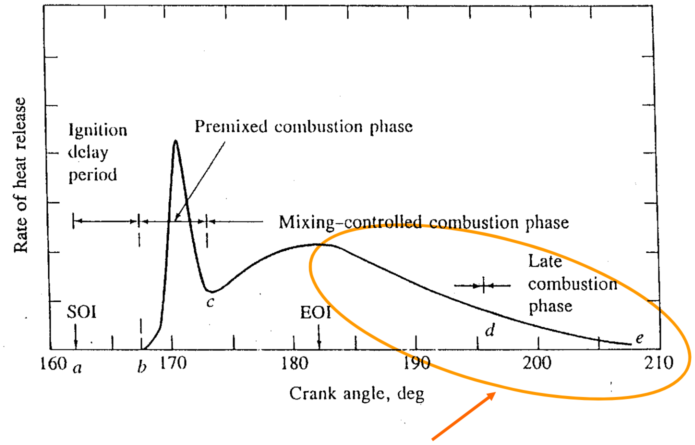


Schematic of fuel jet – air swirl interaction; Φ is the fuel equivalence ratio distribution

Fig. 10-22

Rate of Heat Release in Diesel Combustion

(Fig. 10.8 of Text)



Part of combustion affected most by the charge motion

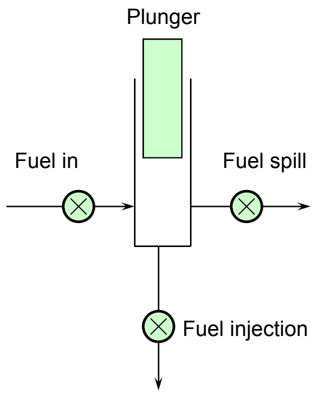
DIESEL FUEL INJECTION HARDWARE

- High pressure system
  - precision parts for flow control
- Fast action
  - high power movements

↓

Expensive system

FUEL METERING AND INJECTION SYSTEM - CONCEPT

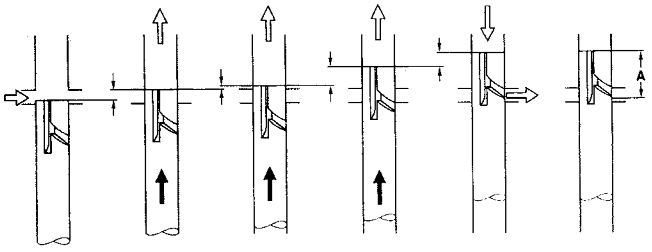


Process:

- Fill
- Pressurize
- Inject
- Spill

Fuel Delivery Control

Fig. 3: Plunger-stroke phases



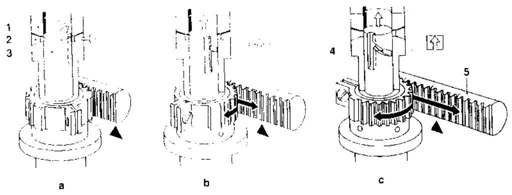
1 *Bottom dead center (BDC)*	2 *Prestroke*	3 *Retraction stroke*	4 *Effective stroke*	5 *Residual stroke*	6 *Top dead center (TDC)*
Fuel flows from the injection pump's fuel gallery and into the high-pressure chamber of the plunger-and-barrel assembly.	Plunger stroke from BDC to closure of the inlet port by the top edge of the plunger (variable depending upon plunger-and-barrel assembly).	Plunger stroke from end of the prestroke until the delivery valve opens (only if a constant-volume valve is used).	Plunger stroke from opening of the delivery valve to opening of the inlet port by the plunger helix (overflow).	Plunger stroke from opening of the inlet port to TDC.	Reversal of plunger travel. **A** Total stroke

From *Diesel Fuel Injection*, Robert Bosch GmbH, 1994

Fuel Rack and In-line Pump

From *Diesel Fuel Injection*,
Robert Bosch GmbH, 1994

Fig. 4: Fuel-delivery control
Using a toothed control rack. a) Zero delivery. b) Partial delivery. c) Maximum delivery.
1 Pump barrel. 2 Inlet port. 3 Pump plunger. 4 Helix. 5 Control rack.



Distributor pump

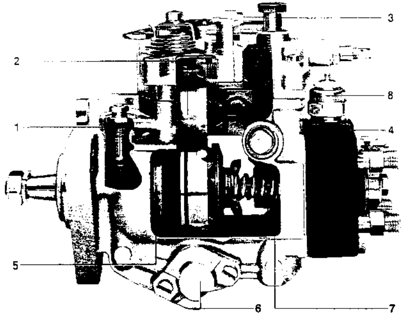
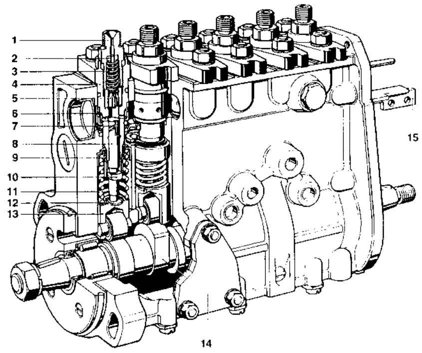


Fig. 1: PES in-line fuel-injection pump
1 Delivery-valve holder, 2 Filler piece, 3 Delivery-valve spring, 4 Pump barrel, 5 Delivery valve, 6 Inlet port and spill port, 7 Control helix, 8 Pump plunger, 9 Control sleeve, 10 Plunger control arm, 11 Plunger spring, 12 Spring seat, 13 Roller tappet, 14 Cam, 15 Control rack.



Diesel Injector

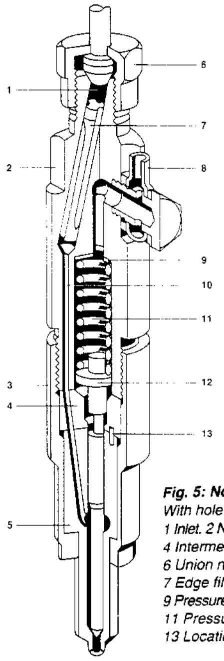
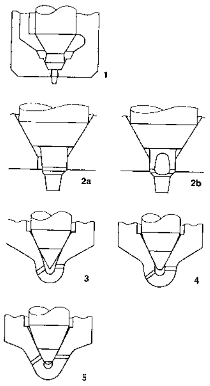
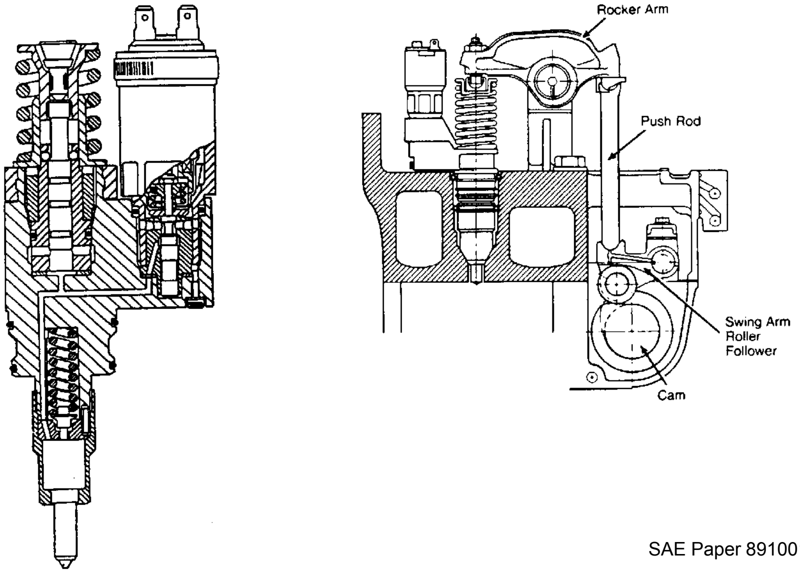


Fig. 5: Nozzle-and-holder assembly
With hole-type nozzle.
1 Inlet. 2 Nozzle-holder body. 3 Nozzle-retaining nut.
4 Intermediate element, 5 Injection nozzle.
6 Union nut with high-pressure line.
7 Edge filter, 8 Leak-off connection.
9 Pressure-adjusting shims, 10 Pressure passage.
11 Pressure spring, 12 Pressure pin.
13 Locating pins.

Fig. 2: Nozzle shapes
1 Throttling pintle nozzle,
2 Throttling pintle nozzle with flat-cut pintle,
2a Side view, 2b Front view,
3 Hole-type nozzle with conical blind hole,
4 Hole-type nozzle with cylindrical blind hole,
5 Seat-hole nozzle.



Electronic Unit Injector

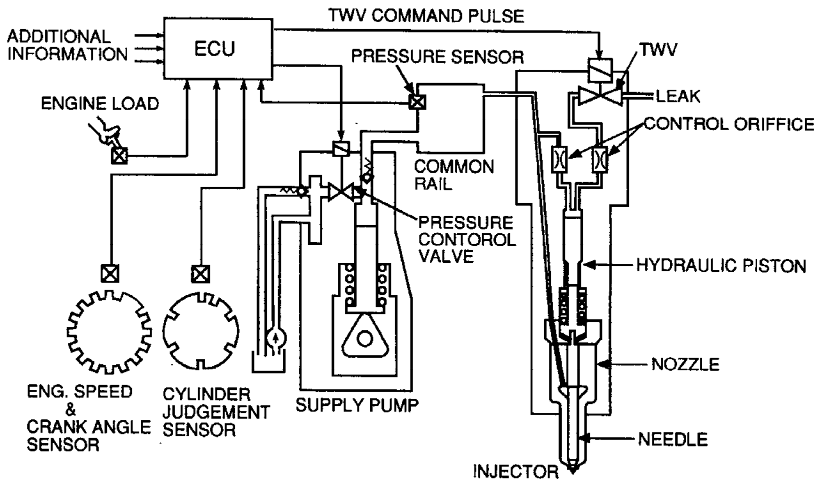


SAE Paper 891001

Injection pressure

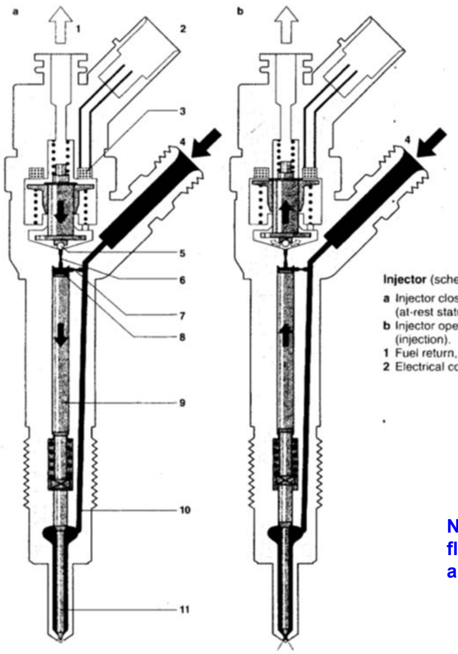
- Positive displacement injection system
  - Injection pressure adjusted to accommodate plunger motion
  - Injection pressure $\propto$ rpm^2
- Injection characteristics speed dependent
  - Injection pressure too high at high rpm
  - Injection pressure too low at low rpm

Common Rail Fuel Injection System



SAE Paper 1999-01-0833

Common Rail Injector



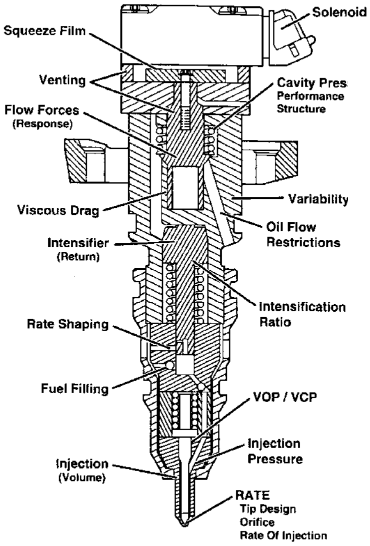
Injector (schematic)

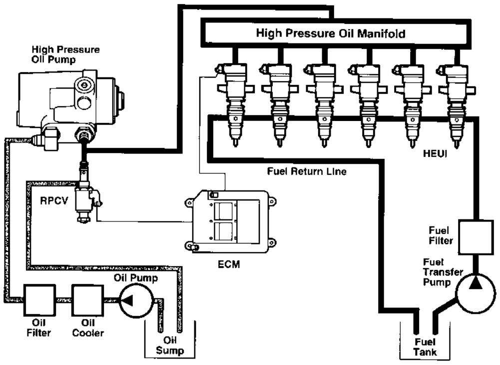
a Injector closed (at-rest status),
b Injector opened (injection).
1 Fuel return,
2 Electrical connection,
3 Triggering element (solenoid valve),
4 Fuel inlet (high pressure) from the rail,
5 Valve ball,
6 Bleed orifice,
7 Feed orifice,
8 Valve control chamber,
9 Valve control plunger,
10 Feed passage to the nozzle,
11 Nozzle needle.

Nozzle opening speed controlled by the flow rate difference between the Bleed (6) and Feed (7) orifices

From Bosch: Diesel Engine Management

Caterpillar Hydraulic Electronic Unit Injector (HEUI)

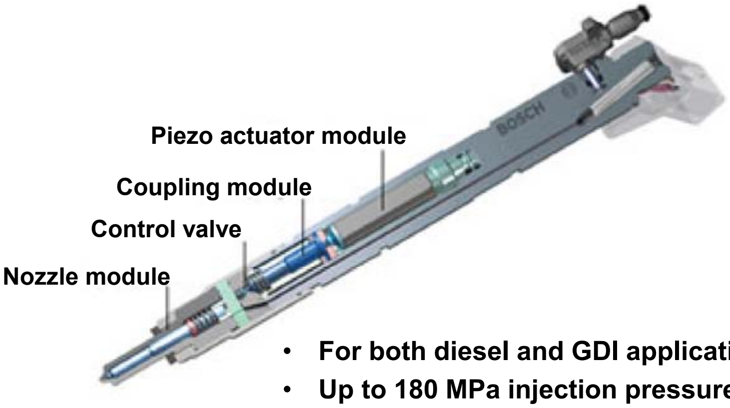




Fuel line: 200kPa; Low pressure oil: 300 kPa;
High pressure oil: up to 23 MPa;
Intensifier area ratio 7:1
Injection pressure up to 150 MPa

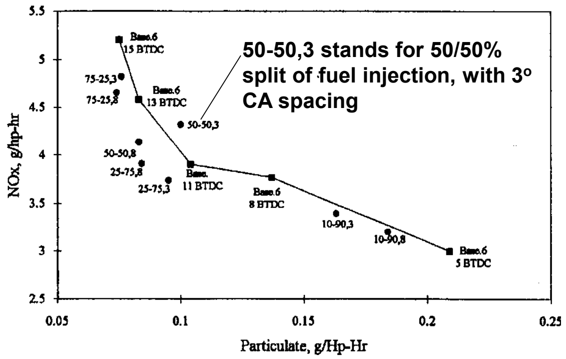
SAE Papers 930270, 930271

Piezoelectric injectors

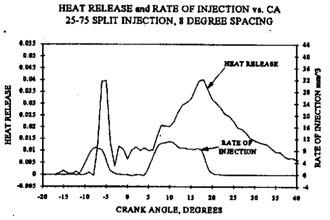


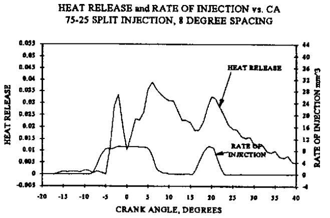
- **For both diesel and GDI applications**
- **Up to 180 MPa injection pressure**
- **5 injections per cycle**
- **In vehicle production already**
- **Suppliers: Bosch; Delphi; Denso; Siemens; …**

Split Injection (SAE Paper 940668)



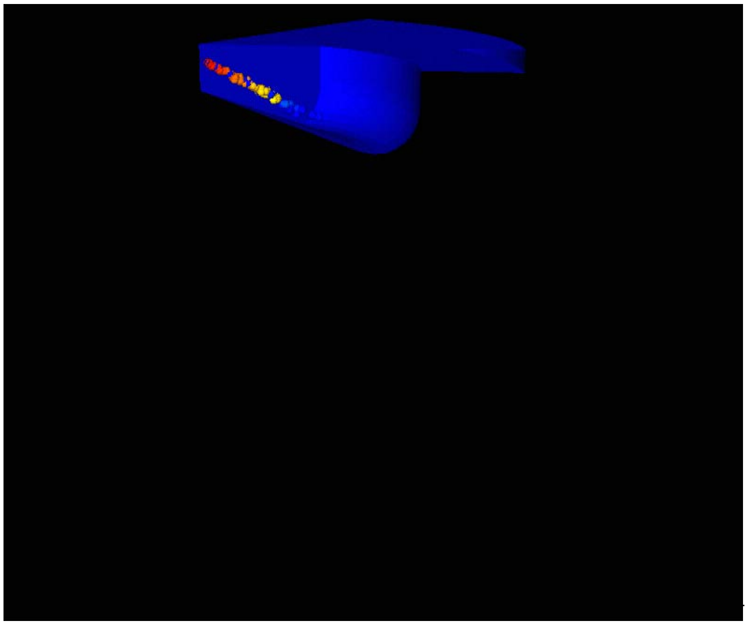
50-50,3 stands for 50/50% split of fuel injection, with 3° CA spacing





1600 rpm, 184 KPa manifold pressure, overall fuel equivalence ratio = 0.45;

Split injection cfd simulation



CHALLENGES IN DIESEL COMBUSTION

Heavy Duty Diesel Engines

- NOx emission
- Particulate emission
- Power density
- Noise

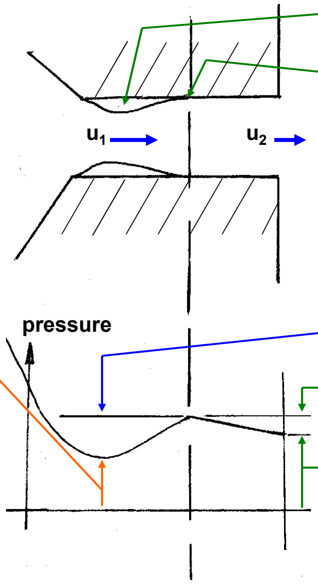
High Speed Passenger Car Diesel Engines

- All of the above, plus
  - Fast burn rate

Cavitation in Injection Nozzle

- Cavitation happens when local pressure is lower than the fluid vapor pressure
- Effects
  - Discharge rate
  - Affects the spray angle
  - Damage to the nozzle passage
- Factors affecting cavitation
  - Combustion chamber pressure
  - Local streamline curvature within the nozzle

Flow process that leads to cavitation



Flow separation (recirculation region)

Flow reattachment

Bernoulli drop

$$\Delta P_b = \tfrac{1}{2}\rho_f (u_1^2 - u_2^2)$$
$$= \tfrac{1}{2}\rho_f u_2^2 [(A_2/A_1)^2 - 1]$$
$$\approx P_{inj} [(A_2/A_1)^2 - 1]$$

further friction drop ΔP_f

Combustion chamber pressure P_c

P_{min} $\approx P_c - \Delta P_b + \Delta P_f$

Cavitation occurs if Pmin $\leq$ fuel saturation pressure

2.61 Internal Combustion Engines
Spring 2017