University of Nottingham
UK | CHINA | MALAYSIA

# Advanced propulsion systems MMME4066

# Fuel delivery


**Antonino La Rocca**

**Professor in Applied Thermofluids and Propulsion Systems**

**Coates Office C39**

**Email: antonino.larocca@nottingham.ac.uk**

# Content

Fuel delivery in SI and DI engines

Fuel delivery

Fuel injection systems

Gas exchange and turbochargers

# Fuel delivery/gas exchange

Fuel delivery and gas exchange control composition and mass of cylinder charge

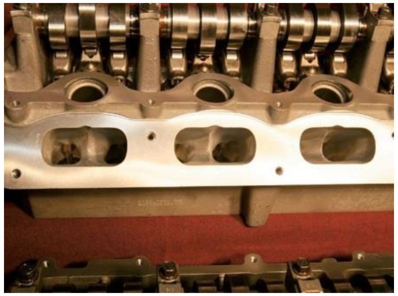
Dictate engine work output, how smooth engine run and level of pollutants

**SI engines**
Air flow through intake manifold controlled by a throttle plate (butterfly valve) at upstream position.
Lower volumetric efficiency (pressure drop at throttle and air displacement by fuel)

**CI engines**
Operate unthrottled (speed/power controlled by fuel injection)
High volumetric efficiency (no flow restriction and displacement of air by fuel)

# Components of fuel and gas exchange systems

Intake manifold/ runners



Exhaust manifold

Intake valves

Exhaust valves

Throttle body

turbocharger

Fuel injectors/carburetors

Supercharger

# Fuel delivery

Prepare an optimum air-fuel ratio mixture which gives:

- the required power output
- lowest fuel consumption and emissions,
- smooth and reliable operation
- operate over a range of speed and load conditions.

Various delivery and injection systems specific to the operation of the engines

**SI port injected**
Well-mixed homogeneous Air & Fuel inducted

**SI direct injection:**
Operates in homogeneous mode (early injection during induction)

**CI direct injection:**
Diesel delivered to individual cylinders

# Fuel injection systems

## Spark ignition engines:

SI carburetor

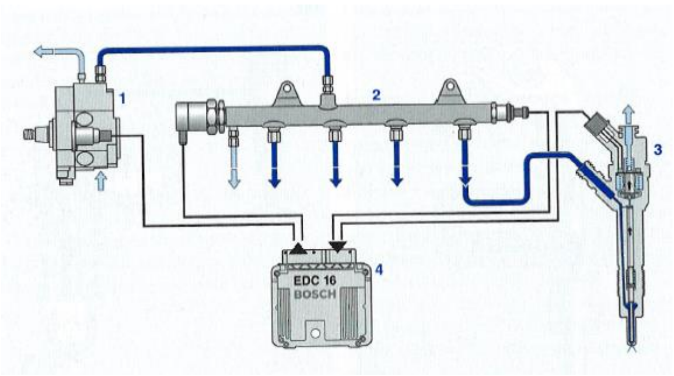
SI multipoint/single-point injection

$p_{inj}$ = 3-6 bar (<100 PSI)
$m_{inj}$ ~ 1-10 mg/stroke
$t_{inj}$ = 1-10 ms

SI direct injection

$p_{inj}$ = 50-70 bar (~1000 PSI)
$m_{inj}$ ~ 14-20 mg/stroke
$t_{inj}$ = 1- 4 ms

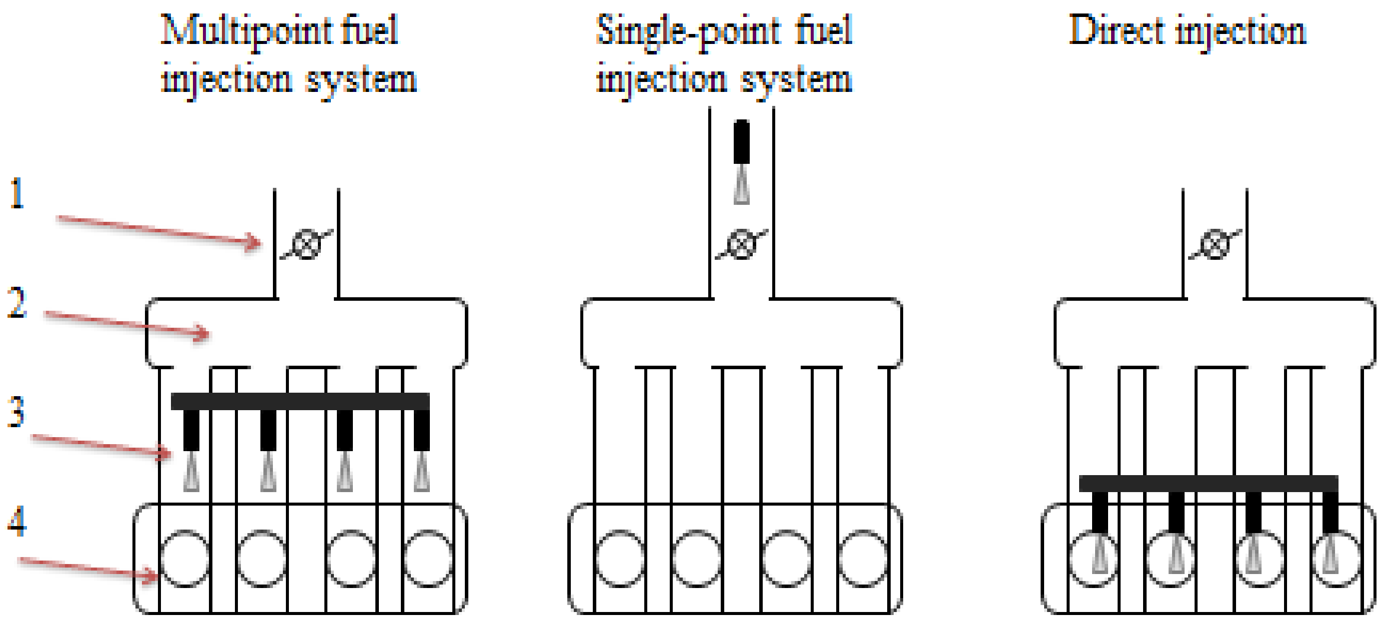
## Compression ignition engines:




$p_{inj}$ = 1400-2000 bar (30000 PSI)
$m_{inj}$ ~ 50-300 mg/stroke
$t_{inj}$ = 1-2 ms

# Gasoline engine – fuel delivery

## Gasoline Fuel injection systems




1) Throttle valve

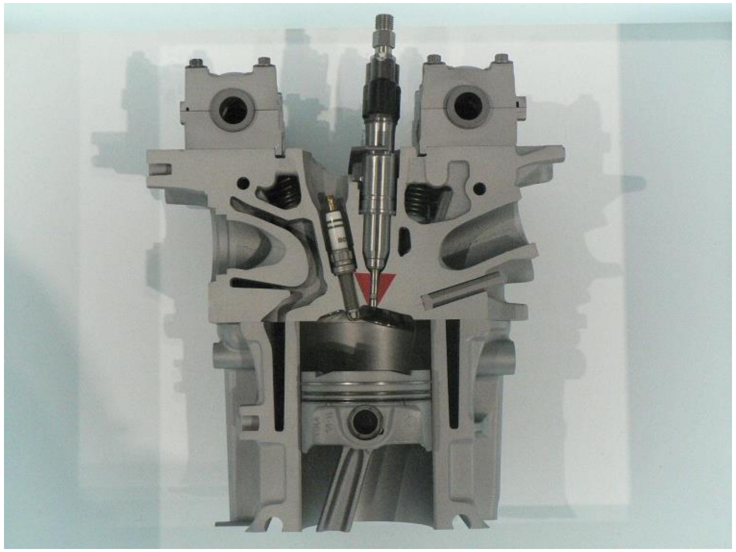
2)Intake manifold

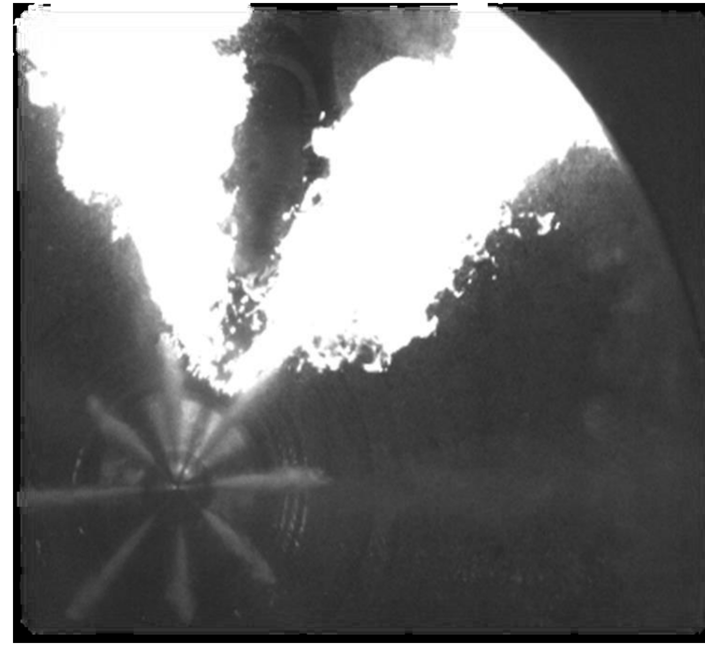
3) Injector

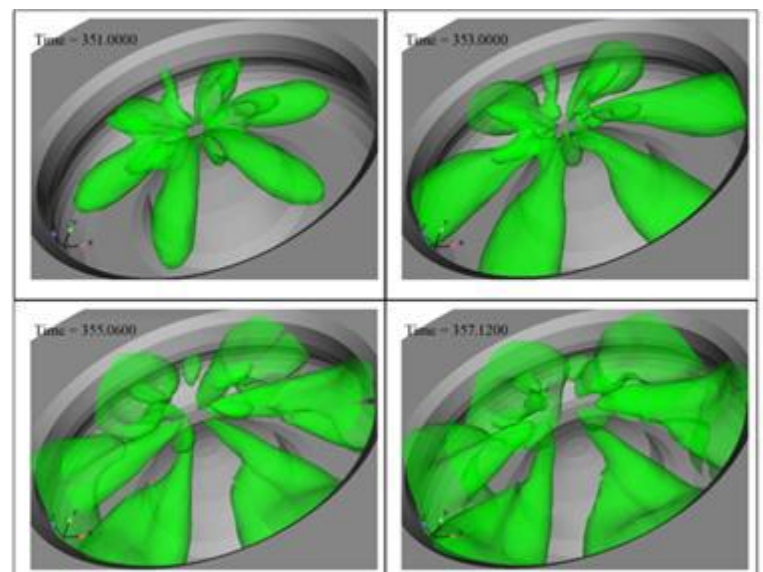
4) Engine

Also relevant for dual fuel operations ($CH_4$, $H_2$, $NH_3$ )

# Electronic control of injectors is nowadays available.



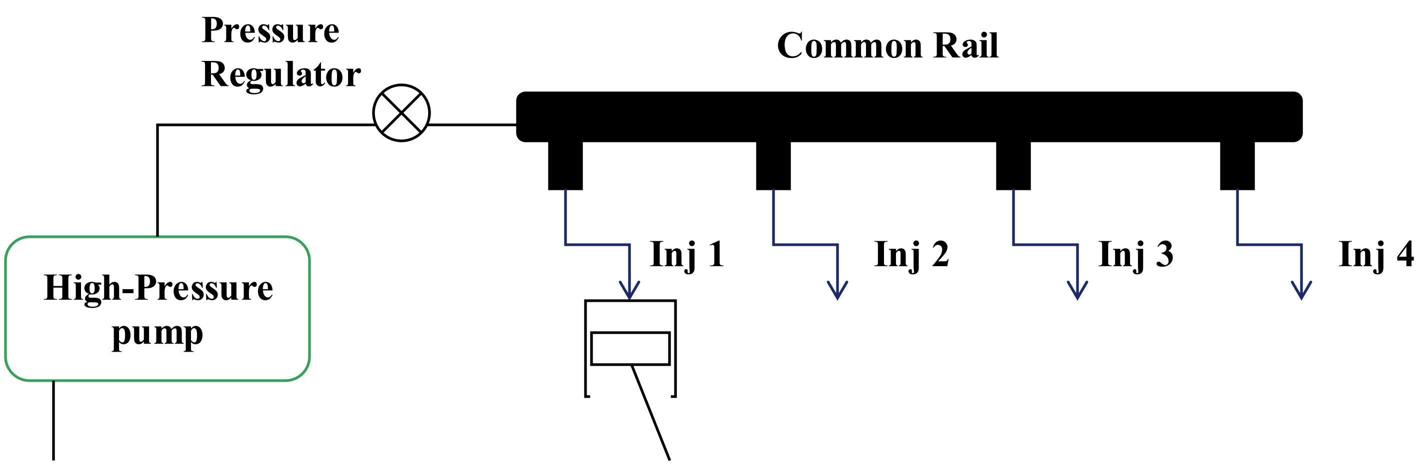




# Diesel Fuel injection systems

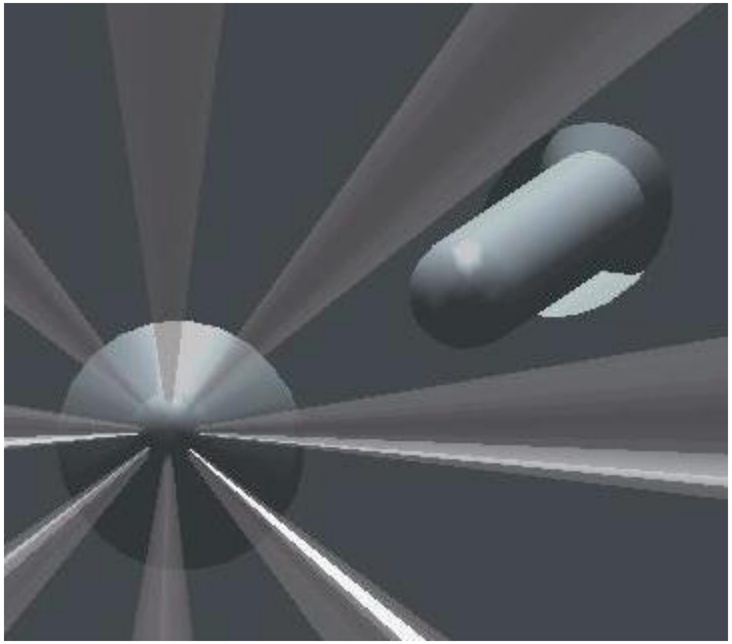
Functions of pressure generation and fuel injection is separated – via accumulator volume composed of common rail and injectors.

Injection pressure generated by **high-pressure pump is independent of engine speed/injected fuel quantity** – offers great injection flexibility.

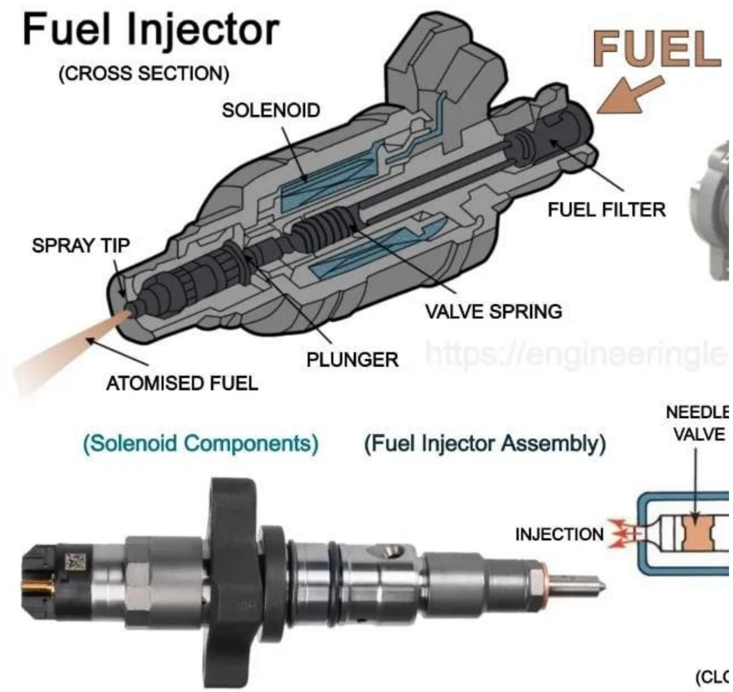



**HPCR** can inject fuel up to 180 MPa and provides a five-time multiple injection function – reducing particulates and $NO_x$, improving output torque, fuel economy and reducing noise/vibration.

# Single and multi hole injectors



Multi hole nozzles are used in most direct-injection engines

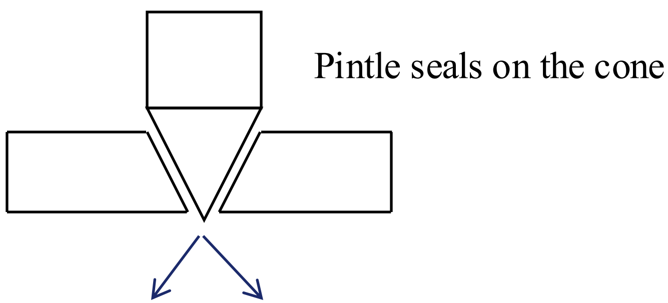



Single hole nozzle suffer from narrow spray angle so no so good mixing unless higher velocities are used.

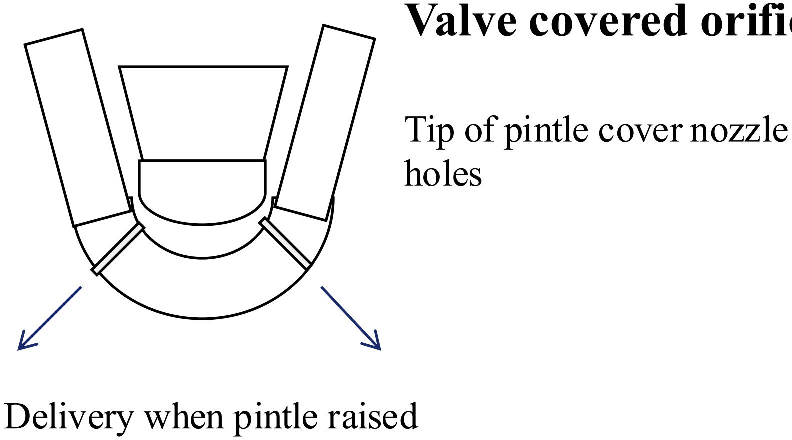
# Injector Nozzle types

Injector nozzle determines the efficiency of mixture formation – plays an essential role in injection rate shaping, optimum atomisation/distribution of fuel and sealing off the fuel-injection system from the combustion chamber.

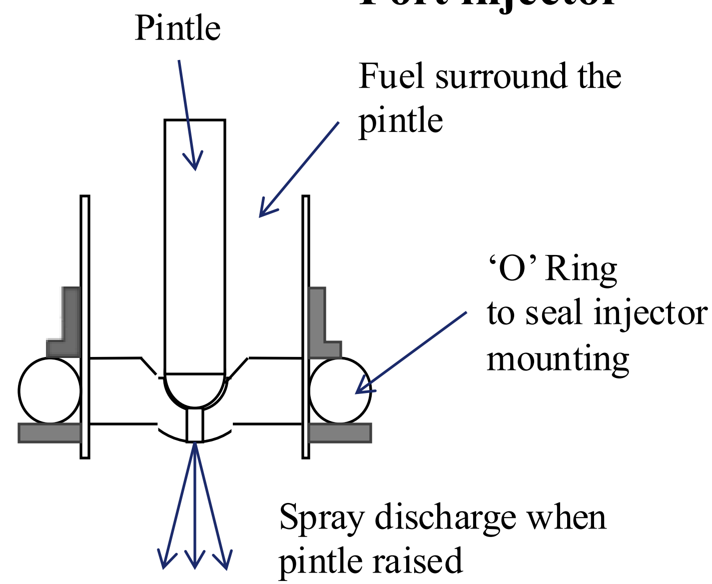
## Throttling Pintle

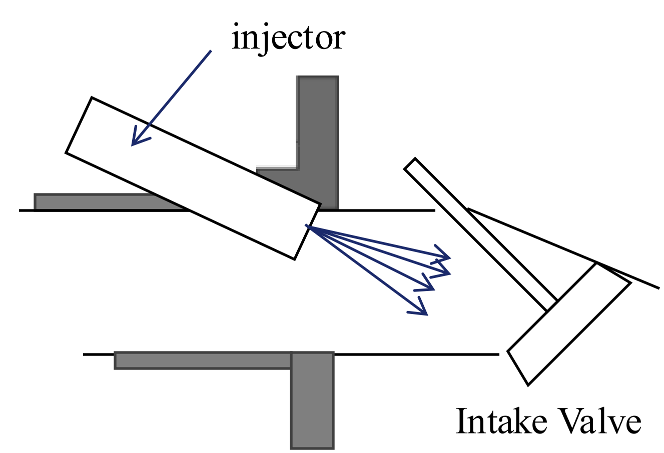



## Valve covered orifice



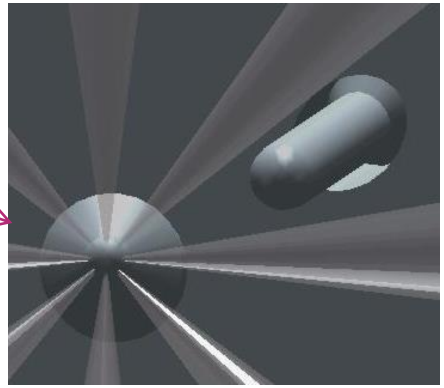

## Port injector



# Direct injection in diesel engines

Injector



Glow plug

Fuel plume

Front view

Side view

Read pages 590- 596 (new Heywood)
Topic is "multiple injection diesel combustion".

Experiment - Engine Research Group

# Fuel injection –spray mass flow rate

A statement of law of conservation of energy is the Bernoulli equation. In a simplified form:

$$\frac{p_1}{\rho} + \frac{v_1^2}{2} = \frac{p_2}{\rho} + \frac{v_2^2}{2}$$

Pressure energy the energy needed to move the flow against the pressure.

Kinetic energy: The energy provided by the movement.

Bernoulli equation can be used if the following assumption are valid:

Flow through each nozzle is quasi steady, incompressible and 1-dimensional.

Then mass flow rate of fuel injected through nozzle can be calculated.

# Fuel injection –spray mass flow rate

The mass flow rate of fuel injected through nozzle can be calculated by calculating the fuel velocity at the nozzle

Bernoulli's equation

$$\frac{p_1}{\rho} + \cancelto{0}{\frac{v_1^2}{2}} = \frac{p_2}{\rho} + \frac{v_2^2}{2}$$

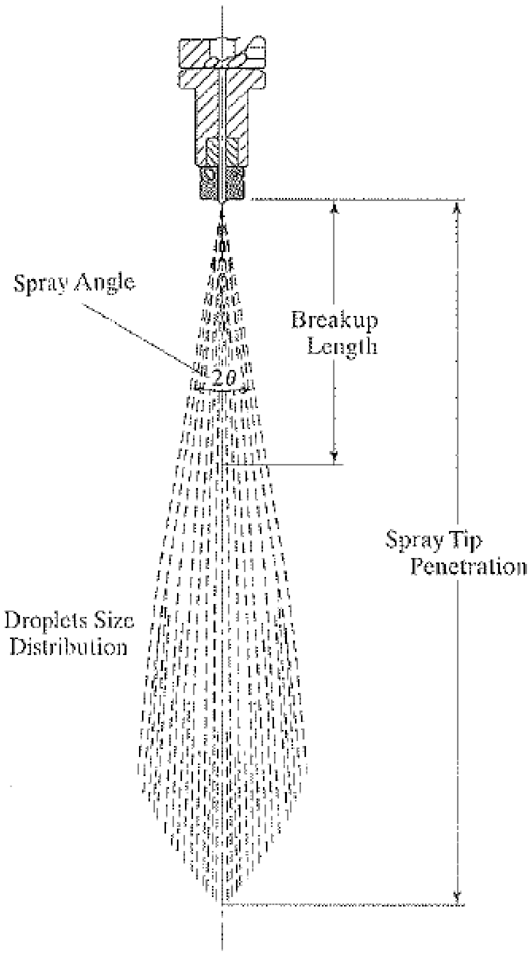
Then

$$\frac{p_1}{\rho} = \frac{p_2}{\rho} + \frac{v_2^2}{2}$$

$$v = \sqrt{\frac{2\Delta p}{\rho}}$$

# Fuel injection – spray mass flow rate

Assuming the flow through each nozzle is quasi steady, incompressible and 1-dimensional, mass flow rate of fuel injected through nozzle is




From Bernoulli's equation

$$v = \sqrt{\frac{2\Delta p}{\rho}}$$

$$\dot{m}_f = \rho C_D A \sqrt{2\ \Delta p / \rho}$$

$C_D$ = discarge coeff

$A$ = total area of nozzle holes

$\rho$ = fuel density

$\Delta p$ = pressure drop

# Worked example

The $Q_{LHV}$ of diesel and gasoline are similar, so the mass of fuel injected to produce a required gross indicated amount of work is similar.

Injection durations are also quite similar (for manufacturing cost, control and performance reasons): typically, a small number of milliseconds (ms).

Thus, each injector might deliver 20mg/stroke (or cycle) in 2 ms at a moderate load and speed for both diesel and gasoline engines.

**Using typical values for the variables, which must be specified, compare the fuel discharge velocities and implied nozzle hole diameters of a single hole gasoline injector and a six-hole diesel injector.**

# Worked example

typical values for the variables which must be specified,

**Mass flow rate**

$$\dot{m} = \frac{20 \cdot 10^{-6}}{2 \cdot 10^{-3}} = 10 \cdot 10^{-3} \frac{kg}{s}$$

**diesel**

$\rho = 870\, kg/m^3$

$C_D = 0.6$

$p_{rail} = 1400 bar$

$p_{deliv} = 40 bar$

$holes = 6$

$m = 20 mg$

$\tau = 2 ms$

Common rail direct injection diesel

Vs

Port injected gasoline

**gasoline**

$\rho = 760\, kg/m^3$

$C_D = 0.6$

$p_{rail} = 3.5 bar$

$p_{deliv} = 1 bar$ (port)

$holes = 1$

$m = 20 mg$

$\tau = 2 ms$

# Worked example -Velocity of the fuel

**Diesel case**

Rail pressure is 1400bar while in-cylinder pressure at time of injection is 40bar

$$v = \sqrt{\frac{2\Delta p}{\rho}} = \sqrt{\frac{(1400 - 40) \cdot 2 \cdot 10^5}{870}} = 559 \frac{m}{s}$$

**gasoline**

fuel pressure is 3.5bar while air pressure in the intake port is 1bar

$$v = \sqrt{\frac{2\Delta p}{\rho}} = \sqrt{\frac{(3.5 - 1) \cdot 2 \cdot 10^5}{760}} = 25.6 \frac{m}{s}$$

# Worked example –Area and diameter

## Hole (s) area

**diesel** $$A = \frac{\dot{m}}{\rho C_D v} = \frac{10^{-2}}{870 \times 0.6 \times 559} = 3.427 \cdot 10^{-8} m^2$$

**gasoline** $$A = \frac{\dot{m}}{\rho C_D v} = \frac{10^{-2}}{760 \times 0.6 \times 25.6} = 8.566 \cdot 10^{-7} m^2$$

**diesel**

$$d = \sqrt{\frac{A}{6} \times \frac{4}{\pi}} = 0.085mm$$

**gasoline**

$$d = \sqrt{A \times \frac{4}{\pi}} = 1mm$$

# Summary

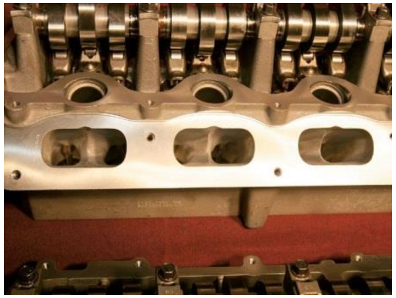
1. Introduction to gas exchange and fuel delivery

2. How fuel is delivered in IC engines

3. Fuel injection systems

# Components of gas exchange systems

Intake manifold/ runners



Exhaust manifold

Intake valves

Exhaust valves

Throttle body

Supercharger

turbocharger

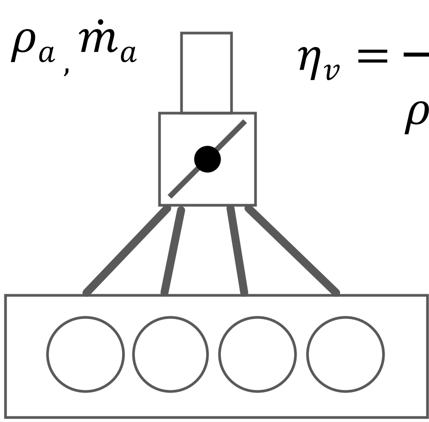
# Volumetric efficiency

Overall measure of the effectiveness of a 4-stroke engine and its intake/exhaust systems as an air pump device.

$$\eta_v = \frac{\dot{V}}{(nV_s)\frac{N}{2}}$$

Indicates: How much of engine swept volume will be filled by air

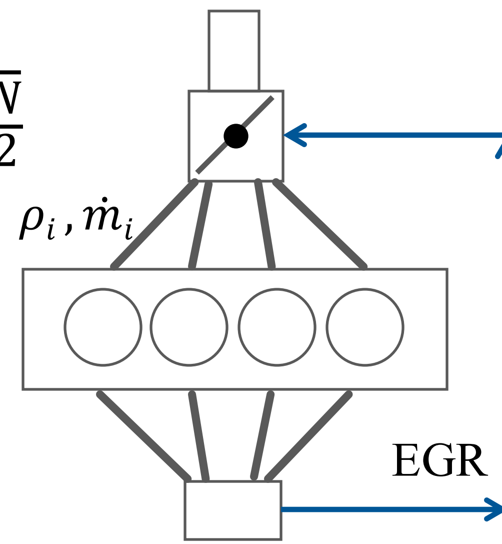
Overall volumetric efficiency




$$\eta_v = \frac{\dot{m}_a}{\rho_a (nV_s)\frac{N}{2}}$$

Volumetric efficiency based on intake port conditions

$$\eta_v = \frac{\dot{m}_i}{\rho_i (nV_s)\frac{N}{2}}$$




Air and exhaust gas can be treated as the same perfect gas

# Residual gases and EGR

Some of the burned gas from previous cycle dilute the new charge – residual gas

Residuals concentration varies with operating condition

| | | |
|---|---|---|
| SI | 20% idle | 7% high loads |
| CI | few % | |

| PRO | CONS |
|---|---|
| Lower combustion temp<br>Reduces NOx | Reduces volumetric efficiency |

Dilution can be increased using External Gas Recirculation (EGR)

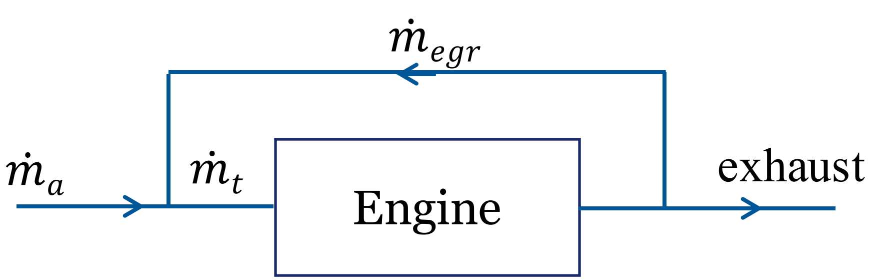
# External Gas Recirculation (EGR)

EGR: route a fraction of the exhaust gas back to intake

| | |
|---|---|
| SI | 15 - 20% |
| CI | up to 60 % |

$\dot{m}_{egr}$ Mass flow rate of recirculated exhaust

$\dot{m}_{a}$ Mass flow rate of air




$$EGR = \frac{\dot{m}_{egr}}{\dot{m}_{t}}$$

$$EGR = \frac{\dot{m}_{egr}}{\dot{m}_{egr} + \dot{m}_{a}}$$

# Example (Notes pag20)

Calculate the overall volumetric efficiency for a spark ignition engine operating at part load with an EGR rate of 15% if the volumetric efficiency based on intake port conditions is 90%, ambient air is at 1.0 bar and 300K, and in the intake ports pressure is 0.6 bar and temperature is 320 K. Assume the exhaust gas has the same specific gas constant as air.

$$\eta_v(overall) = \frac{\dot{m}_a}{\rho_a (nV_s)\frac{N}{2}}$$

Gas: Charge and EGR

$$EGR = \frac{\dot{m}_{egr}}{\dot{m}_{egr} + \dot{m}_a}$$

$$\eta_v(port) = \frac{\dot{m}_g}{\rho_g (nV_s)\frac{N}{2}}$$

$$\dot{m}_g = \dot{m}_a + \dot{m}_{egr}$$

$$\dot{m}_g = \frac{\dot{m}_a}{1 - EGR}$$

$$\eta_v(port) = \frac{\dot{m}_a}{(1 - EGR)\rho_g (nV_s)\frac{N}{2}}$$

$$\eta_v(overall) = \frac{\dot{m}_a}{\rho_a (nV_s)\frac{N}{2}}$$

# Example (Notes pag20)

$$\eta_v(port) = \frac{\dot{m}_a}{(1-EGR)\rho_g\,(nV_s)\,\frac{N}{2}}$$

$$\eta_v(overall) = \frac{\dot{m}_a}{\rho_a\,(nV_s)\,\frac{N}{2}}$$

$$\dot{m}_a = \eta_v(port)(1-EGR)\rho_g\,(nV_s)\,\frac{N}{2}$$

$$\eta_v(overall)\rho_a = \eta_v(port)(1-EGR)\rho_g$$

$$\eta_v(overall) = (1-EGR)\,\frac{\rho_g}{\rho_a}\eta_v(port)$$

$$\frac{\rho_g}{\rho_a} = \frac{p_g}{RT_g}\frac{RT_a}{p_a} = \frac{0.6}{1.0}\times\frac{300}{320} = 0.5625$$

$$\eta_v(overall) = (1-0.15)\,\times 0.5625\times 0.9 = 0.43 = 43\%$$

# Supercharging and turbocharging

Best way to increase power is to increase intake air pressure above ambient using an air pump that forces air into the engine at $P_{in} > P_{ambient}$

**Turbocharging**: Divert exhaust gas through a turbine & use shaft power to drive air pump. use of high pressure gas (otherwise wasted during blowdown)

**Supercharging:** air pump driven directly from engine rather than separate turbine.

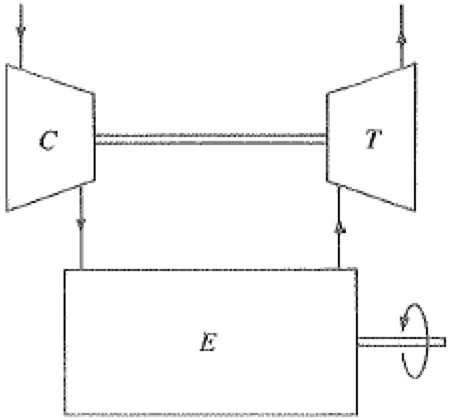
**Limitations / problems**

- Need time to overcome inertia of rotating parts & fill intake manifold with high-pressure air ("turbo lag")
- Turbochargers: moving parts in hot exhaust system - not durable
- Cost and complexity

**Turbo Lag**: sudden throttle change – (e.g when throttle is quickly opened to accelerate the vehicle) the turbocharger will not respond as quickly as a supercharger. It can take several engine revolutions to change the exhaust flow rate and to speed up the turbine

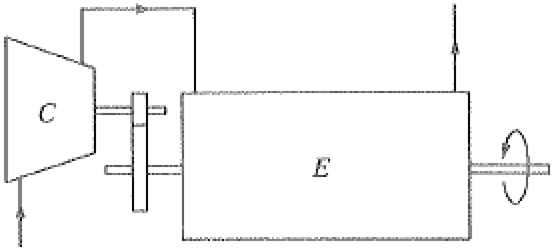
# Supercharging and turbocharging configurations

## Turbocharger

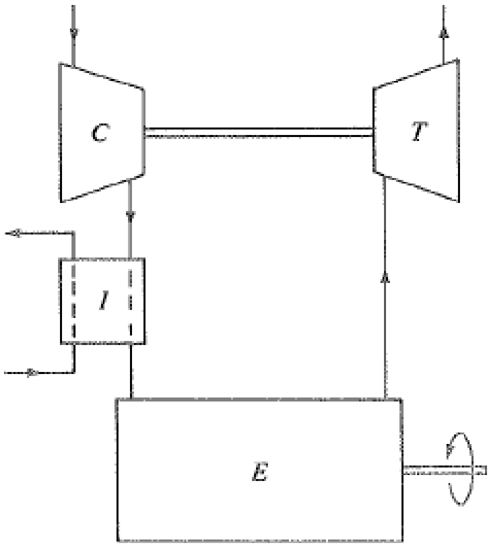



These units run at 120,000 rpm or more

## Mechanical supercharger




Jaguar V8 gasoline XJS - high performance




## Turbocharger with intercooler

"intercooler" cools intake air after compression but before entering the engine

# Supercharging

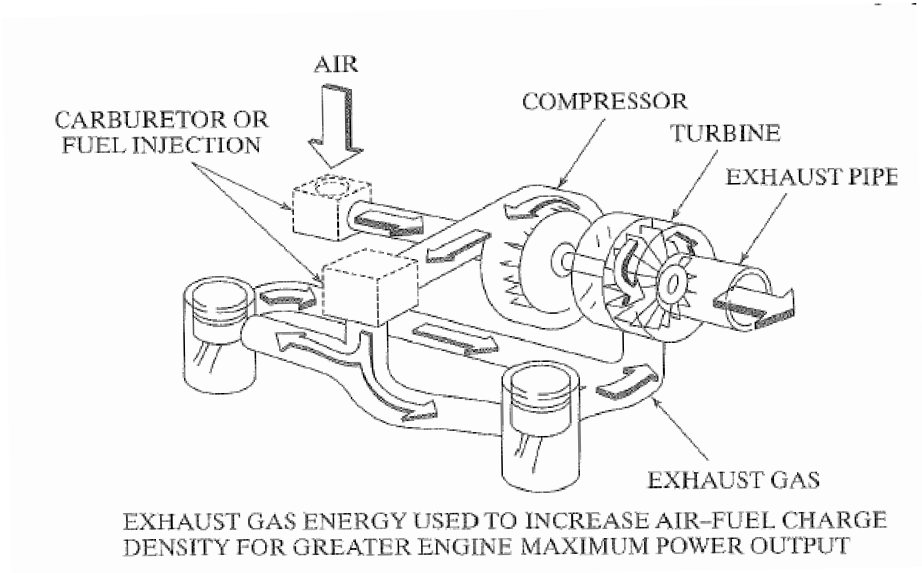
Compressors mounted in the intake system to raise the pressure of incoming air/charge – increase in-cylinder trapped mass and hence, higher net power output.

Superchargers are compressors mechanically driven directly off the engine crankshaft at the same speed of the engine.

Although superchargers are more powerful and have extremely quick dynamic response towards throttle/engine speed changes, they are more expensive to run (parasitic load on engine output), heavier and noisier.

# Turbocharging

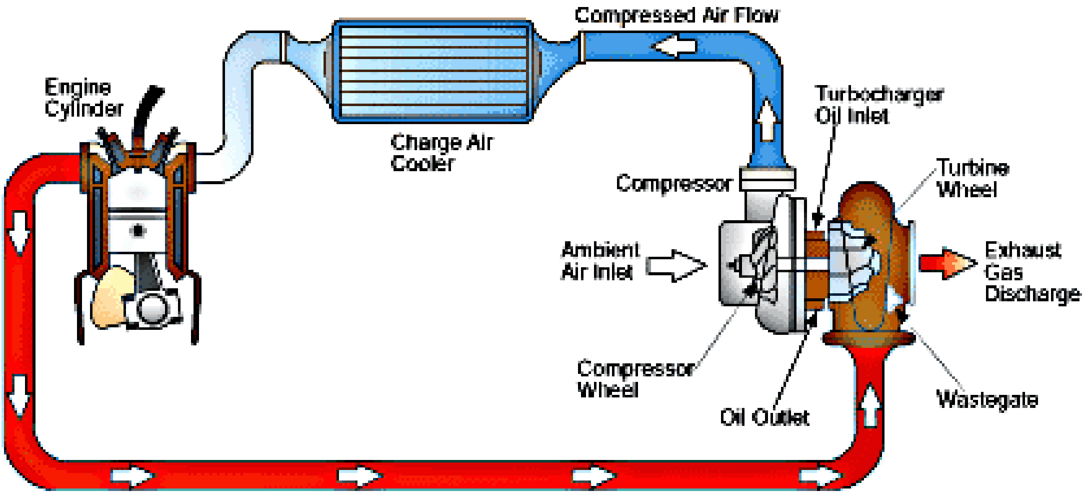
Compressor of turbocharger is powered by turbine mounted as close as possible to the cylinder exhaust ports (pressure/temperature/KE highest here).

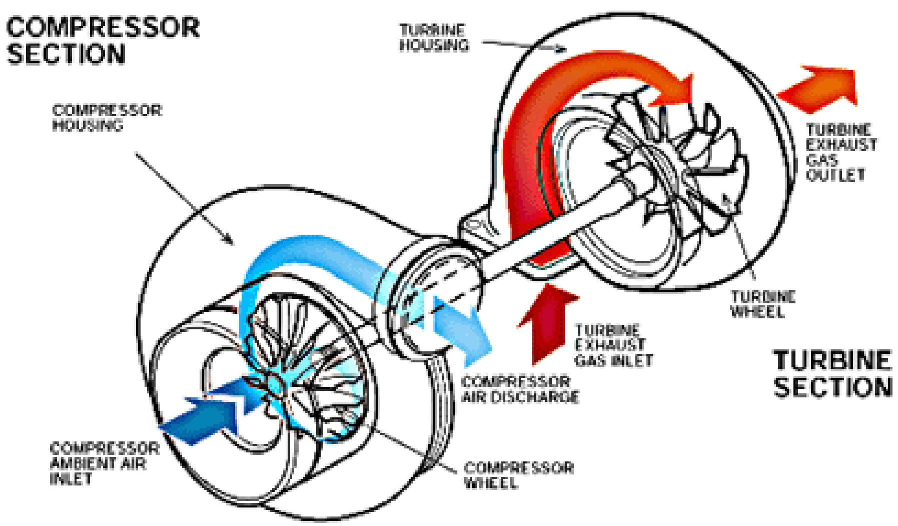



**Advantage:**
includes recovery of exhaust energy to drive turbocharger,
**Disadvantages**:
restrictive exhaust flow and turbo lag (bad dynamic response towards throttle/speed changes).

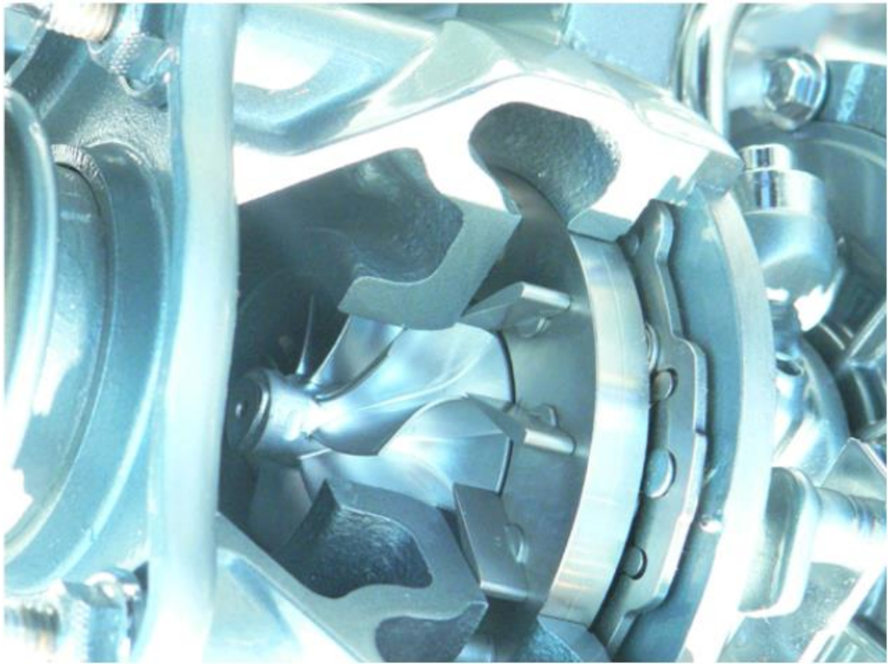
**Aftercooler** is used with turbochargers to lower the temperature of compressed intake air as in superchargers. Raising the density at constant pressure increses trapped mass of fresh charge. Lowering intake temp also reduces danger of knock (in SI engines)

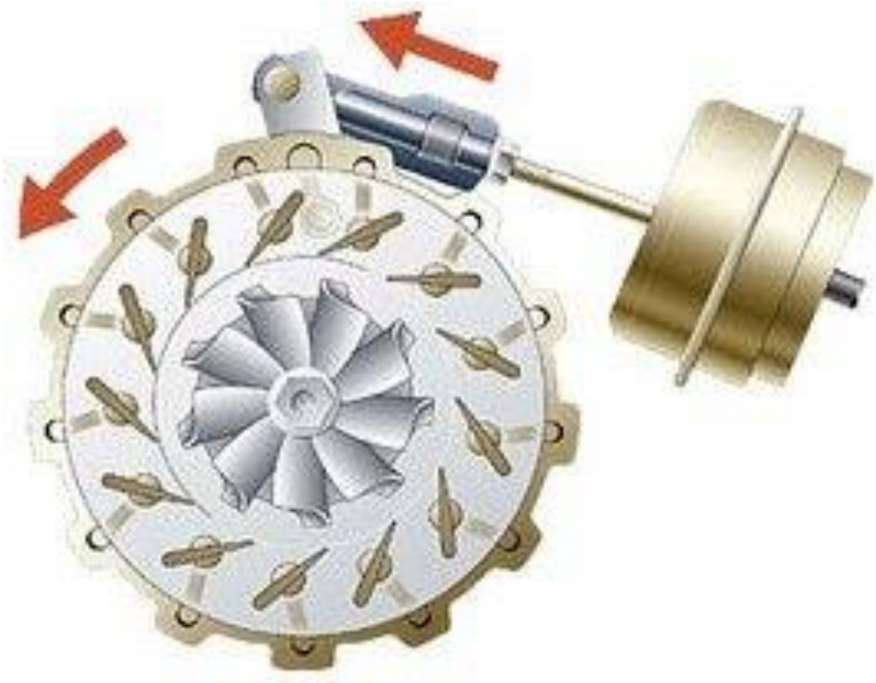
# Turbocharging



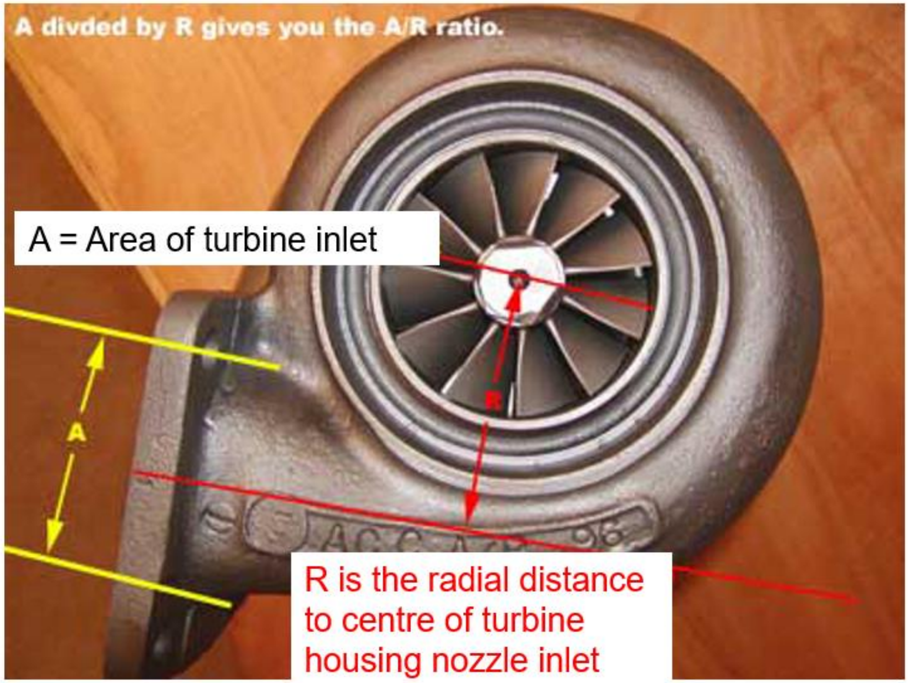

# Variable Geometry Turbochargers (Diesels)



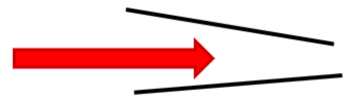


- Variable geometry at the entry to the turbine allows higher boost pressures at lower mass flows by reducing the aspect ratio to mimic a smaller turbocharger
- Better suited to turbo-diesels:
    - Higher air mass flow
    - Lower peak temperatures (VGT mechanism limited to 900°C, ok for diesels but gasoline engines >950°C)

# Turbine Aspect Ratio

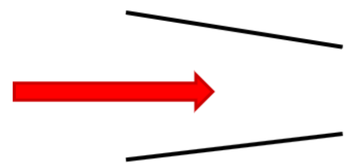



Aspect Ratio = A / R



**LOW ASPECT RATIO**

- Higher gas velocities and high boost at low flow
- Engine will become more choked by turbine as engine speed increases



**HIGH ASPECT RATIO**

- Higher flow capacity
- Engine less choked by turbine
- Lower gas velocities
- Less boost

# Efficiency of the compressor

From 1st Law and assuming isentropic process through compressor:

Isentropic power

$$\dot{W}_{c\ \text{isen}} = \dot{m}_a\left(h_{2s} - h_1\right) = \dot{m}_a c_p\left(T_{2s} - T_1\right)$$

Assume negligible heat transfer, kinetic energy and potential energy terms

Compressor isentropic efficiency

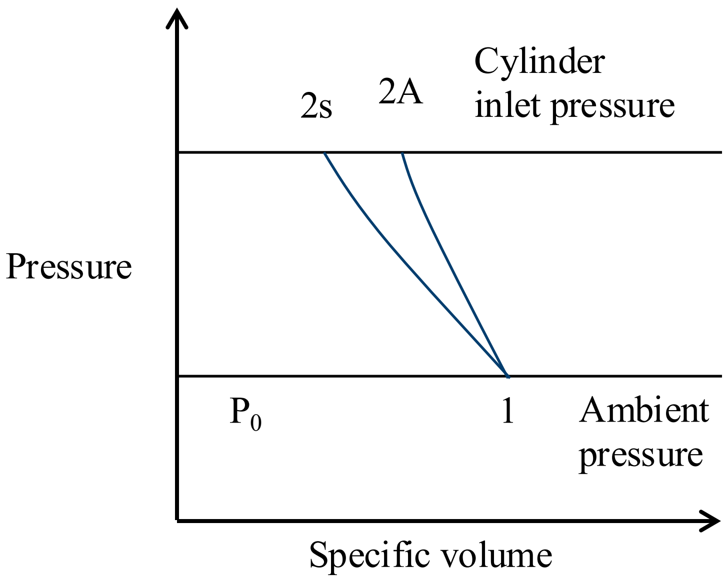
$$\eta_{c_{isen}} = \frac{\dot{W}_{c\ \text{isen}}}{\dot{W}_c} = \frac{\text{isentropic}}{\text{actual}}$$

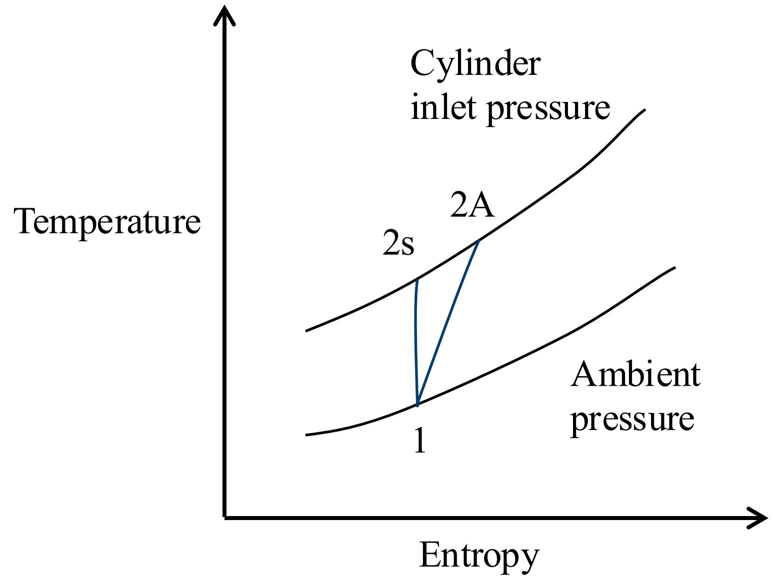
Actual power

$$\dot{W}_c = \frac{\dot{m}_a c_p}{\eta_{c_{isen}}}\left(T_{2s} - T_1\right)$$

# Turbocharger compressor

All compressors have isentropic efficiencies less than 100%, so the actual power needed from the engine to drive the supercharger's compressor is actually greater.



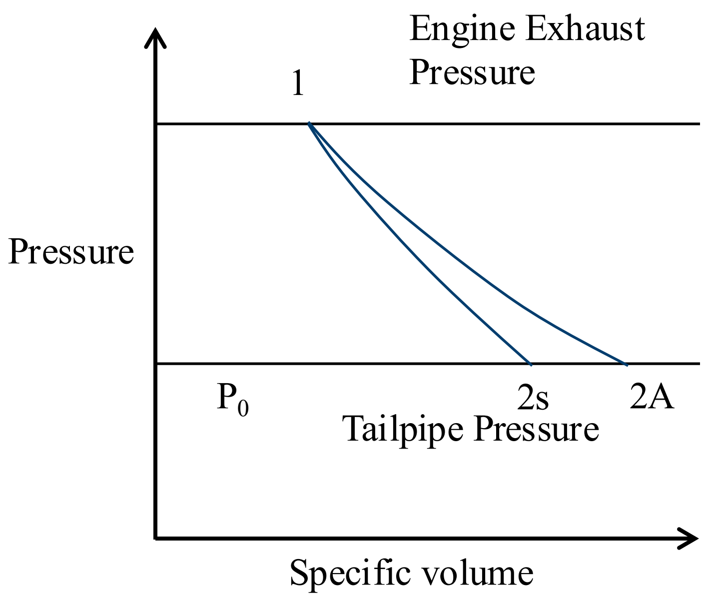

$$\eta_{c_{isen}} = \frac{\dot{W}_{c\ \text{isen}}}{\dot{W}_c} = \frac{\dot{m}_a c_p (T_{2s} - T_1)}{\dot{m}_a c_p (T_{2A} - T_1)} = \frac{(T_{2s} - T_1)}{(T_{2A} - T_1)}$$

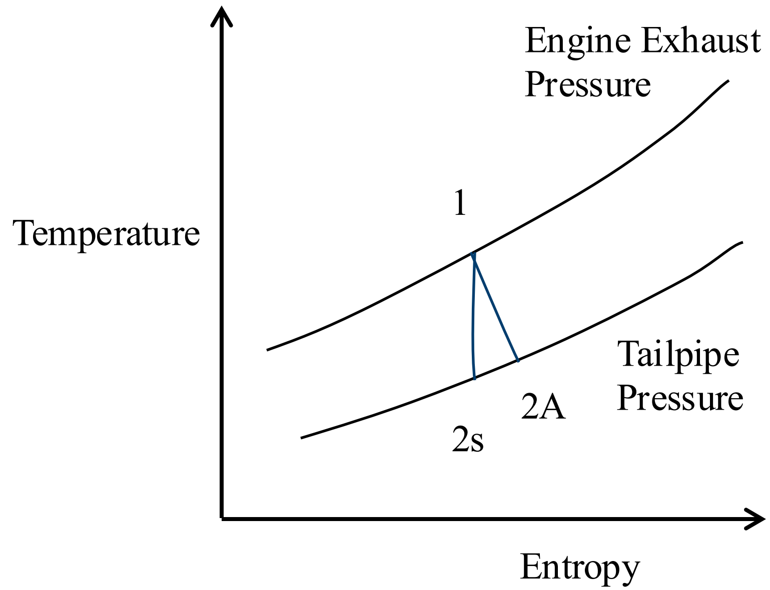
Ideal gas isentropic relationship:

$$T_{2s} = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}}$$

# Turbocharger turbine

From 1st Law and assuming isentropic process through turbocharger's turbine:




$$\eta_{T_{isen}} = \frac{\dot{W}_T}{\dot{W}_{T\ \text{isen}}} = \frac{\dot{m}_{ex} c_p (T_1 - T_{2A})}{\dot{m}_{ex} c_p (T_1 - T_{2S})} = \frac{(T_1 - T_{2A})}{(T_1 - T_{2S})}$$

# Example

A 6-cyliner, 4.8-litre, supercharged engine with volumetric efficiency 158% and compressor isentropic efficiency of 92% is running at 3500 rev/min. Ambient conditions are 23°C and 0.98 bar, and air needs to be delivered into the cylinders at 65°C and 1.8 bar. Intake air density is assumed to be at 1.181 kg/m³ and $c_p$ = 1.005 kJ/(kg.K). What is the rate of heat transfer required in the aftercooler for this setup,

First, find the mass flow rate into the engine.

$$\dot{m}_a = \eta_v \rho_{a,i} (nV_s) N/2 = 1.58 \times 1.181 \times 4.8 \times 10^{-3} \times \frac{3500}{120} = 0.261 \frac{kg}{s}$$

To find the ideal isentropic temperature after compression,

$$T_{2s} = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = 296 \left( \frac{1.8}{0.98} \right)^{\frac{1.4-1}{1.4}} = 352K$$

# Example: heat rejection in intercooler

Use the isentropic relationship to find the actual temperature after compression:

$$\eta_{c_{isen}} = \frac{\dot{W}_{c\ \text{isen}}}{\dot{W}_c} = \frac{\dot{m}_a c_p \left(T_{2s} - T_1\right)}{\dot{m}_a c_p \left(T_{2A} - T_1\right)} = \frac{\left(T_{2s} - T_1\right)}{\left(T_{2A} - T_1\right)}$$

$$T_{2A} = T_1 + \frac{\left(T_{2s} - T_1\right)}{\eta_{c_{isen}}} = \ 296 + \frac{\left(352 - 296\right)}{0.92} = 357K \qquad 84^{o}C \quad > 65$$

Amount of aftercooling needed to reduce the temperature to the required level at 338 K is 19 K, hence the rate of cooling is

$$\dot{Q} = \dot{m}_a c_p \left(T_{2A} - T_2\right) = \ 0.261 \times 1.005 \left(357 - 338\right) = 5kW$$