## Engine Turbo/Super Charging

## Super and Turbo-charging

**Why super/ turbo-charging?**

- Fuel burned per cycle in an IC engine is air limited
  - $(F/A)_{stoich} = 1/14.6$

$$Torq = \frac{\eta_f m_f Q_{HV}}{2\pi n_R}$$

$$Power = Torq \cdot 2\pi N$$

$$m_f = \left(F/A\right)\eta_V \rho_{a,0} V_D$$

$\eta_f, \eta_v$ – fuel conversion and volumetric efficiencies
$m_f$ – fuel mass per cycle
$Q_{HV}$ – fuel heating value
$n_R$ – 1 for 2-stroke, 2 for 4-stroke engine
$N$ – revolution per second
$V_D$ – engine displacement
$\rho_{a,0}$ – air density

Super/turbo-charging: increase air density

## Super- and Turbo- Charging

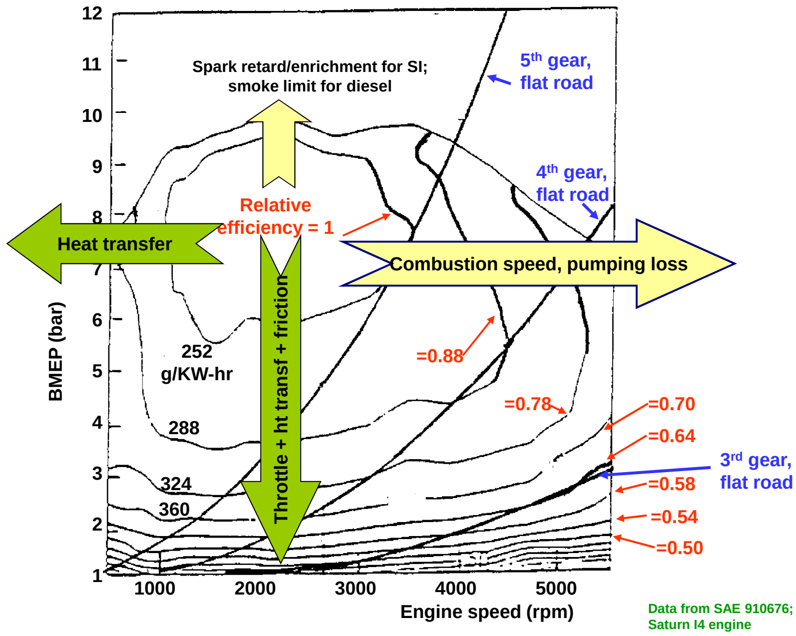
Purpose: To increase the charge density

- Supercharge: compressor powered by engine output
  - No turbo-lag
  - Does not impact exhaust treatment
  - Less efficient than turbo-charging
- Turbo-charge: compressor powered by exhaust turbine
  - More directly utilize exhaust energy
  - Turbo- lag problem
  - Affects exhaust treatment
- Intercooler
  - Increase charge density (hence output power) by cooling the charge
  - Lowers $NO_x$ emissions
  - Suppresses knock

## Additional benefit of turbo-charging

- Can downsize engine while retaining same max power
  - Less throttle loss under part load in SI engine
- Higher BMEP reduces relative friction and heat transfer losses

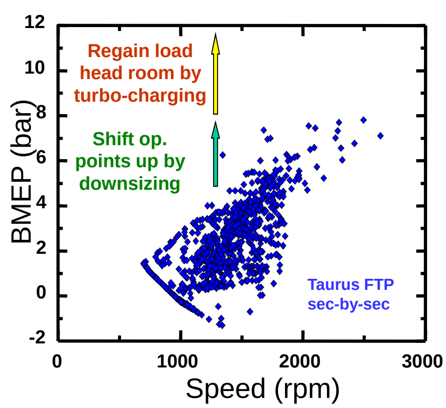
# Engine Losses




Data from SAE 910676;
Saturn I4 engine

5

# SI engine efficiency opportunity




Turbo DISI as enabling technology

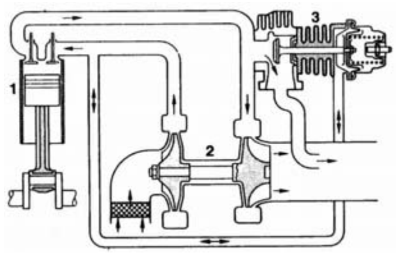
- Fuel in-flight evaporation cools charge
  - More knock resistant

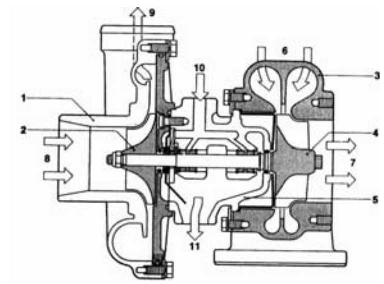
Issues

- Knock
- Peak pressure
- Boosting capacity
- Cold start emissions
  - HC
  - PM

**Exhaust-gas turbocharger for trucks** 1.Compressor housing, 2. Compressor impeller, 3. Turbine housing, 4. Rotor, 5. Bearing housing, 6. inflowing exhaust gas, 7. Out-flowing exhaust gas, 8. Atmospheric fresh air, 9. Pre-compressed fresh air, 10. Oil inlet, 11. Oil return

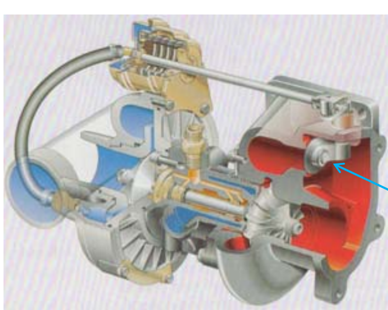
**Charge-air pressure regulation with wastegate on exhaust gas end.** 1.Engine, 2. Exhaust-gas turbochager, 3. Wastegate

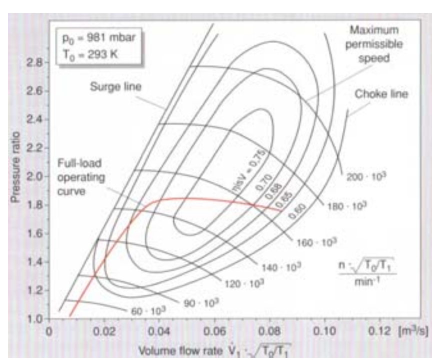



From Bosch Automotive Handbook

# Turbo-charger

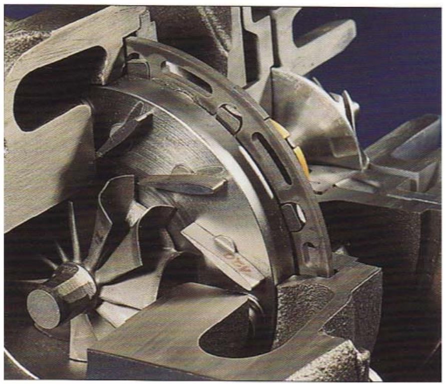




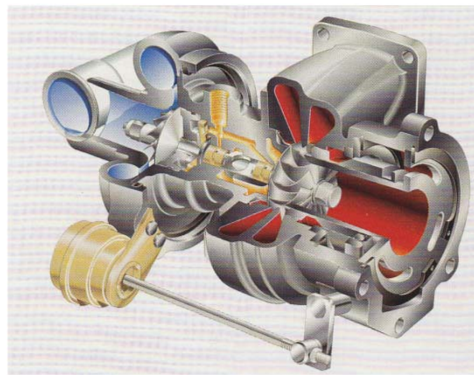

Waste gate

Source: BorgWarner Turbo Systems

## Variable geometry turbo-charger



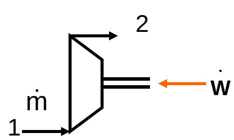
Variable Guide Vane

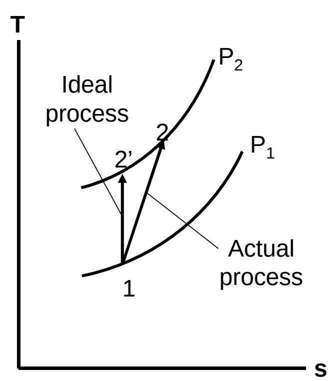


Variable sliding ring

Source: BorgWarner Turbo Systems

## Compressor: basic thermodynamics




**Compressor efficiency $\eta_c$**

$$\eta_c = \frac{\dot{W}_{ideal}}{\dot{W}_{actual}}$$

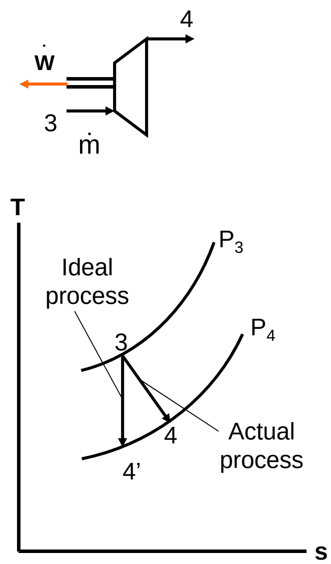
$$\dot{W}_{ideal} = \dot{m} c_p T_1 \left( \frac{T_2{}'}{T_1} - 1 \right)$$

$$\frac{T_2{}'}{T_1} = \left( \frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

$$\dot{W}_{actual} = \frac{1}{\eta_c} \dot{m} c_p T_1 \left( \left( \frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)$$

$$T_2 = T_1 + \frac{\dot{W}_{actual}}{\dot{m} c_p}$$

## Turbine: basic thermodynamics




**Turbine efficiency $\eta_t$**

$$\eta_t = \frac{\dot{W}_{actual}}{\dot{W}_{ideal}}$$

$$\dot{W}_{ideal} = \dot{m} c_p T_3 \left(1 - \frac{T_4'}{T_3}\right)$$

$$\frac{T_4'}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\dot{W}_{actual} = \eta_t \dot{m} c_p T_3 \left(1 - \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}\right)$$

$$T_4 = T_3 - \frac{\dot{W}_{actual}}{\dot{m} c_p}$$

## Properties of Turbochargers

- Power transfer between fluid and shaft $\propto$ RPM$^3$
  - Typically operate at ~ 60K to 120K RPM
- RPM limited by centrifugal stress: usually tip velocity is approximately sonic
- Flow devices, sensitive to boundary layer (BL) behavior
  - Compressor: BL under unfavorable gradient
  - Turbine: BL under favorable gradient

## Torque characteristics of flow machinery

Angular momentum theorem

$$Torq = \left[\int (rV_\theta)\rho V_x \, dA\right]_1 - \left[\int (rV_\theta)\rho V_x \, dA\right]_2$$

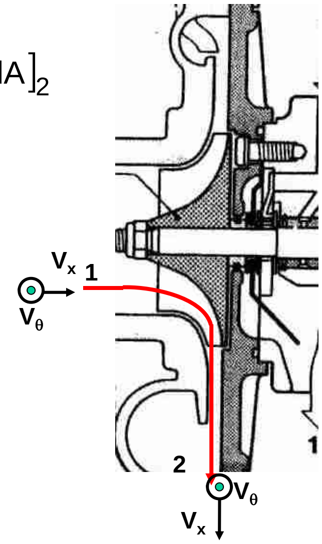
$$\propto V_x V_\theta = V_\theta^2 \left(\frac{V_x}{V_\theta}\right)$$

$\left(\frac{V_x}{V_\theta}\right)$ is fixed by the blade angle

and $V_\theta \propto$ RPM, therefore :

$$Torq \propto (RPM)^2$$

$$Power \propto (RPM)^3$$




## Rotor stress

Force balance over mass element from r to dr

$$(\sigma A)_r - (\sigma A)_{r+dr} = \rho_m A dr \frac{(\omega r)^2}{r}$$

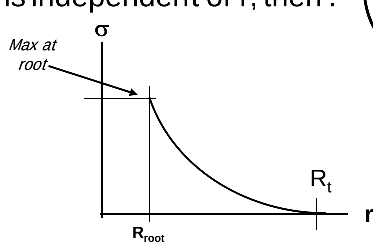
or

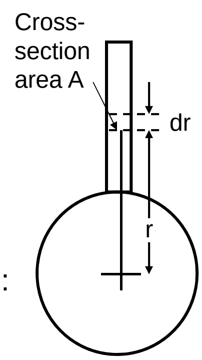
$$\frac{d(\sigma A)}{dr} = -\rho_m A \omega^2 r$$

To illustrate effect, say A is independent of r, then :

$$\sigma(r) = \frac{\rho_m \omega^2}{2}\left(R_t^2 - r^2\right)$$

| | |
|---|---|
| $\sigma$ | Tensile stress |
| $\rho_m$ | Material density |
| $\omega$ | Angular velocity = $2\pi N$ |
| $R_t$ | Tip radius |



## Typical super/turbo-charged engine parameters

- Peak compressor pressure ratio ≈ 2.5
- BMEP up to 24 bar
- Limits:
  - compressor aerodynamics
  - cylinder peak pressure
  - NOx emissions

## Compressor/Turbine Characteristics

- Delivered pressure $P_2$
- $P_2 = f(\dot{m}, RT_1, P_1, N, D, \mu, \gamma,$ geometric ratios$)$
- Dimensional analysis:
  - 7 dimensional variables → (7-3) = 4 dimensionless parameters (plus $\gamma$ and geometric ratios)

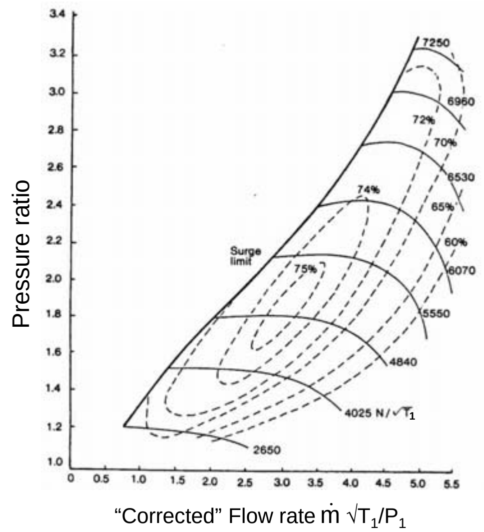
$$\left(\frac{P_2}{P_1}\right) = f\Bigg(\frac{N}{\underbrace{\sqrt{\gamma RT_1}}_{\text{Velocity}}/D}, \frac{\dot{m}}{\underbrace{\left(\frac{P_1}{RT_1}\right)}_{\text{Density}}\underbrace{\sqrt{RT_1}}_{\text{Velocity}}D^2}, Re, \gamma, \text{ geometric ratios}\Bigg)$$

High Re number flow → weak Re dependence

For fixed geometry machinery and gas properties

$$\left(\frac{P_2}{P_1}\right) = f\left(\frac{N}{\sqrt{T_1}}, \frac{\dot{m}\sqrt{T_1}}{P_1}\right)$$

## Compressor Map

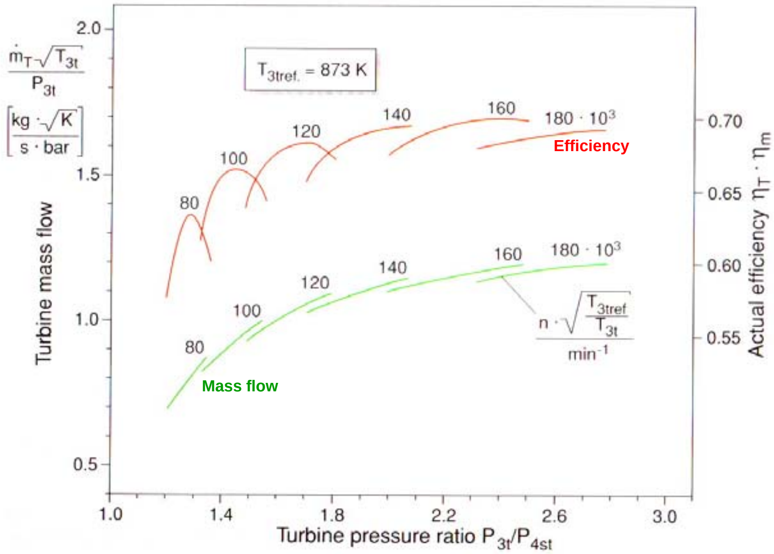



$T_1$= inlet temperature (K); $P_1$= inlet pressure (bar); N = rev. per min.; $\dot{m}$ = mass flow rate (kg/s)
(From "**Principles and Performance in Diesel Engineering**," Ed. by Haddad and Watson)

## Compressor stall and surge

- Stall
  - Happens when incident flow angle is too large (large $V_\theta/V_x$)
  - Stall causes flow blockage
- Surge
  - Flow inertia/resistance, and compression system internal volume comprise a LRC resonance system
  - Oscillatory flow behave when flow blockage occurs because of compressor stall
    - ➢ reverse flow and violent flow rate surges

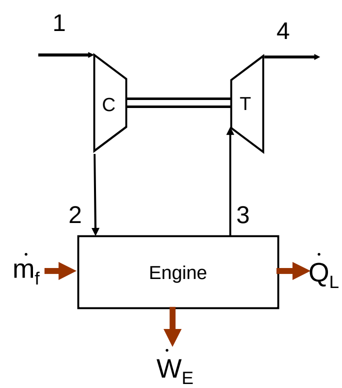
## Turbine Map



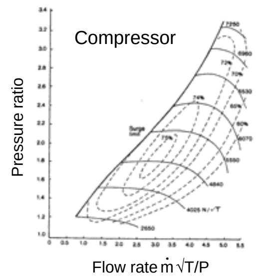

Source: BorgWarner Turbo Systems

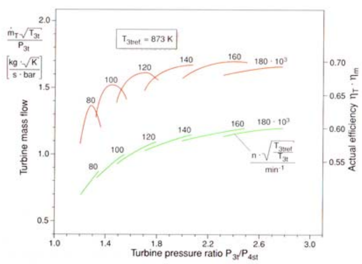
## Compressor Turbine Matching Exercise

- For simplicity, take away intercooler and wastegate
- Given engine brake power output ($\dot{W}_E$ ) and RPM, compressor map, turbine map, and engine map
- Find operating point, i.e. air flow ($\dot{m}_a$), fuel flow rate ( $\dot{m}_f$ ) turbo-shaft revolution per second (N), compressor and turbine pressure ratios ($\pi_c$ and $\pi_t$) etc.



Compressor/ turbine/engine matching solution




Procedure:

1. Guess $\pi_c$; can get engine inlet conditions:

$$P_2 = \pi_c P_1 \qquad T_2 = \frac{T_1}{\eta_c}\left[(\pi_c)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_1$$

2. Then engine volumetric efficiency calibration will give the air flow $\dot{m}_a$ that can be 'swallowed'
3. From $\dot{m}_a$ and $\pi_c$, the compressor speed N can be obtained from the compressor map
4. The fuel flow rate $\dot{m}_f$ may be obtained from the engine map:

$$\dot{W}_E = \dot{m}_f \text{ LHV } \eta_f(\text{RPM}, \dot{W}_E, \text{A/F})$$

5. Engine exhaust temperature $T_3$ may be obtained from energy balance (with known engine mech. eff. $\eta_M$)

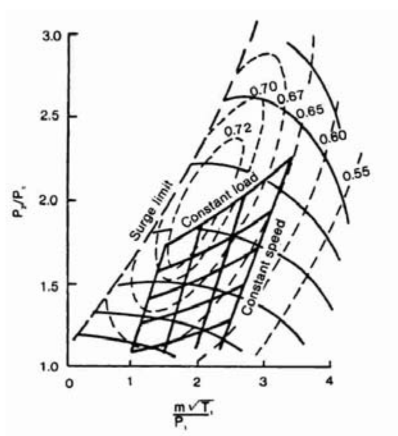
$$(\dot{m}_a + \dot{m}_f)c_p T_3 = \dot{m}_a c_p T_2 + \dot{m}_f \text{LHV} - \frac{\dot{W}_E}{\eta_M} - \dot{Q}_L$$

6. Guess $\pi_t$, then get turbine speed $N_t$ from turbine map and mass flow
7. Determine turbine power from turbine efficiency on map

$$\dot{W}_t = \eta_t\left[1 - \left(\frac{1}{\pi_t}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

8. Iterate on the values of $\pi_c$ and $\pi_t$ until $\dot{W}_t = \dot{W}_c$ and $N_t = N_c$

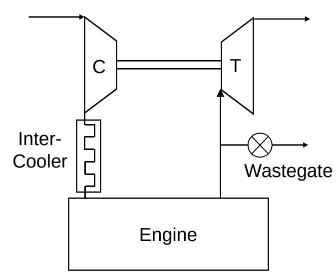
# Compressor/ Engine/ Turbine Matching




Compressor characteristics, with airflow requirements of a four-stroke truck engine superimposed.

(From **"Principles and Performance in Diesel Engineering,"** Ed. by Haddad and Watson)

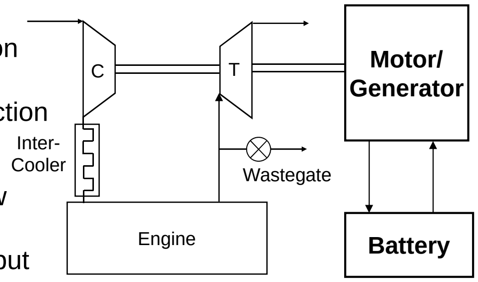
- Mass flows through compressor, engine, turbine and wastegate have to be consistent
- Turbine inlet temperature consistent with fuel flow and engine power output
- Turbine supplies compressor work
- Turbine and compressor at same speed



## Advanced turbocharger development

### Electric assisted turbo-charging

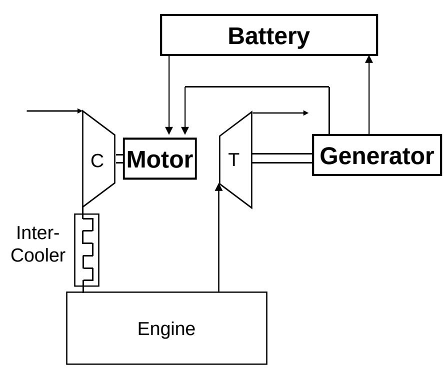
- Concept
  - Put motor/ generator on turbo-charger
  - reduce wastegate function
- Benefit
  - increase air flow at low engine speed
  - auxiliary electrical output at part load




## Advanced turbocharger development

### Electrical turbo-charger

- Concept
  - turbine drives generator; compressor driven by motor
- Benefit
  - decoupling of turbine and compressor map, hence much more freedom in performance optimization
  - Auxiliary power output
  - do not need wastegate; no turbo-lag

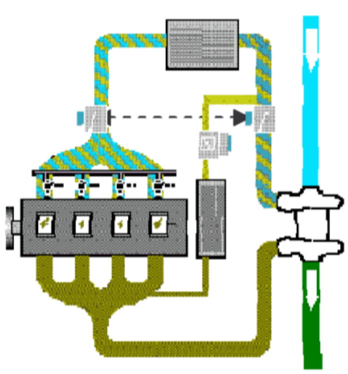


## Advanced turbocharger development

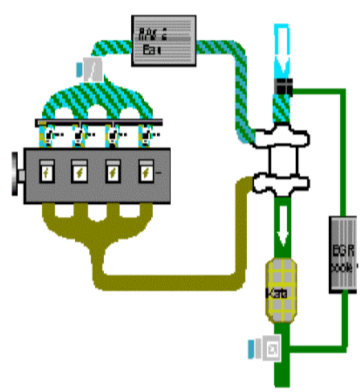
### Challenges

- Interaction of turbo-charging system with exhaust treatment and emissions
  - Especially severe in light-duty diesel market because of low exhaust temperature
  - Low pressure and high pressure EGR circuits
    - ➢Transient response
- Cost

## EGR/ turbo Configurations



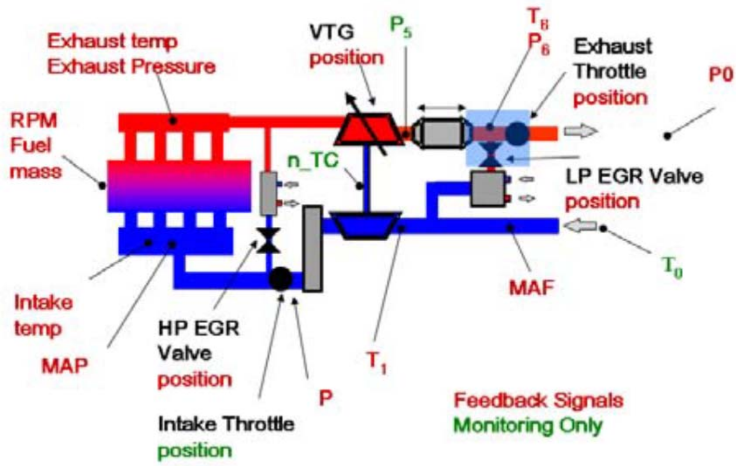
High pressure architecture



Low pressure architecture

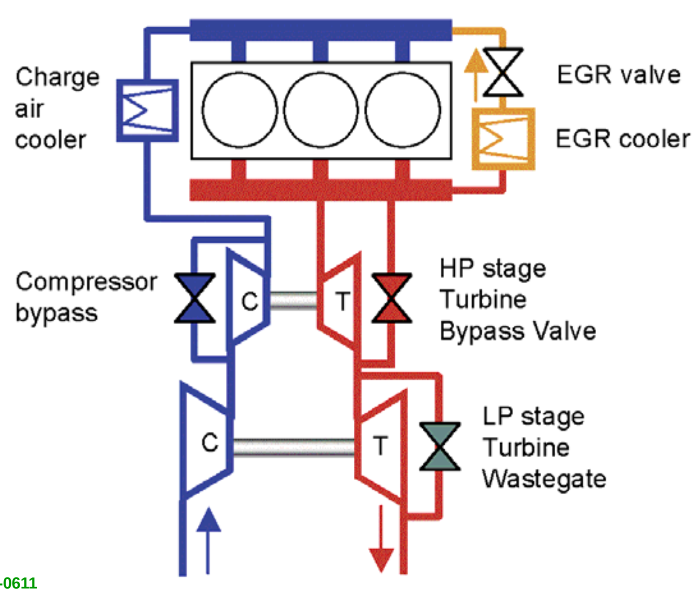
From SAE 2007-01-2978

## Hybrid EGR




From SAE 2009-01-1451

## Two stage turbo with HP EGR loop




SAE 2008-01-0611