

Engine Turbo/Super Charging

Super and Turbo-charging

Why super/ turbo-charging?

- Fuel burned per cycle in an IC engine is air limited

$$- (F/A)_{\text{stoich}} = 1/14.6$$

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

$$\text{Power} = \text{Torq} \cdot 2\pi N$$

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

η_f, η_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

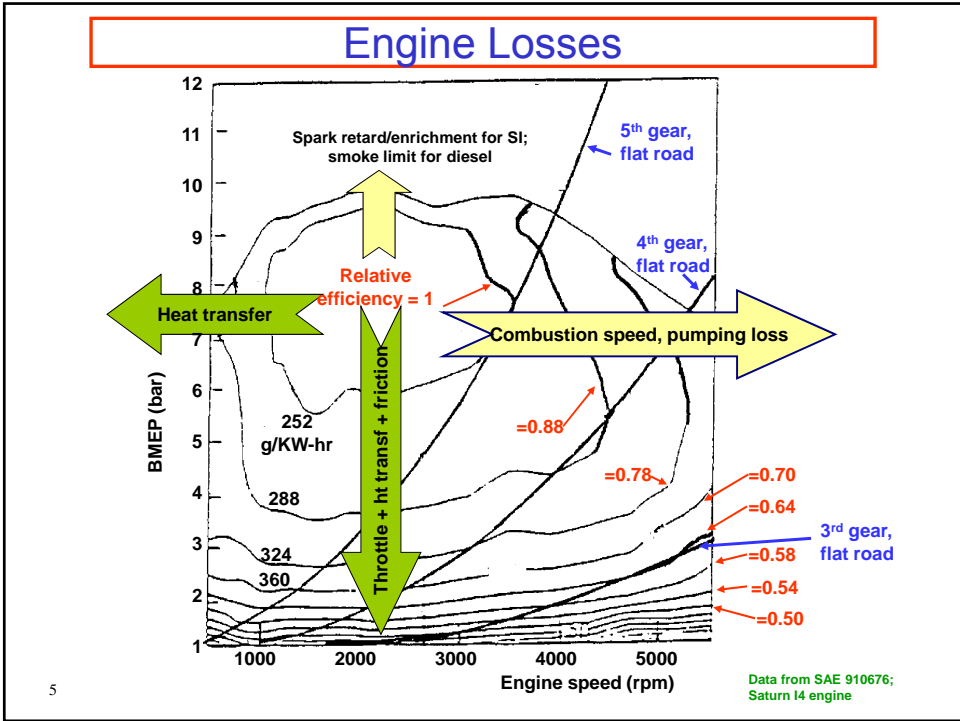
Super- and Turbo- Charging

Purpose: To increase the charge density

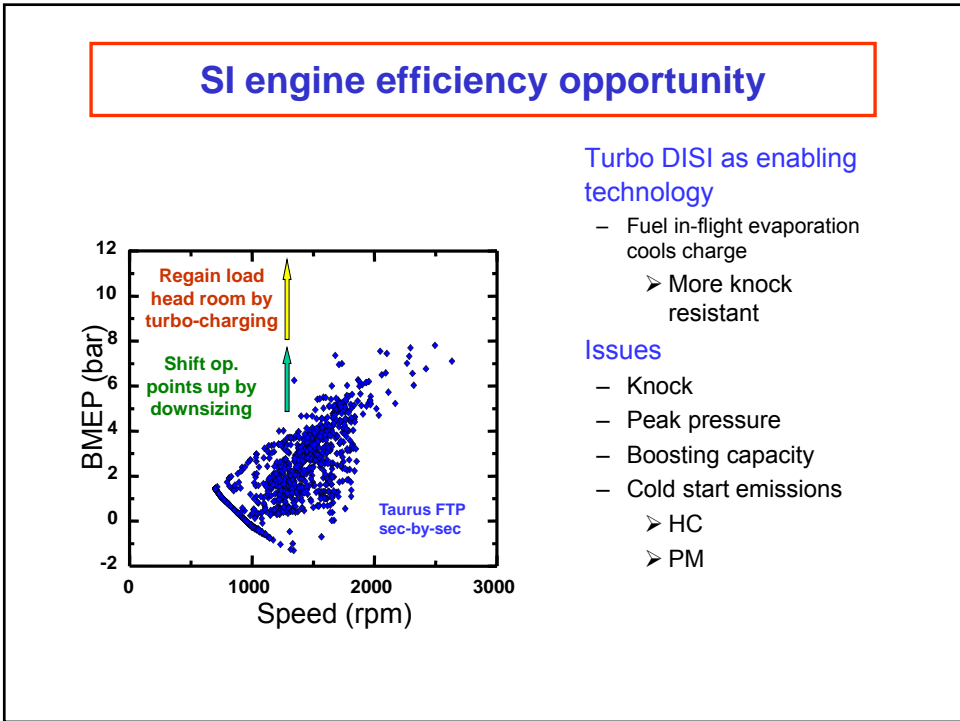
- Supercharge: compressor powered by engine output
 - No turbo-lag
 - Does not impact exhaust treatment
 - Less efficient than turbo-charging
- Turbo-charge: compressor powered by exhaust turbine
 - More directly utilize exhaust energy
 - Turbo- lag problem
 - Affects exhaust treatment
- Intercooler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

Additional benefit of turbo-charging

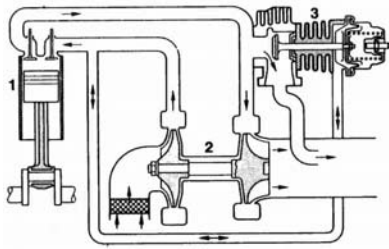
- Can downsize engine while retaining same max power
 - Less throttle loss under part load in SI engine
- Higher BMEP reduces relative friction and heat transfer losses



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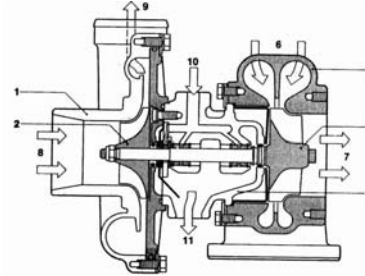


Charge-air pressure regulation with wastegate on exhaust gas end. 1.Engine, 2. Exhaust-gas turbochager, 3. Wastegate



Exhaust-gas turbocharger for trucks

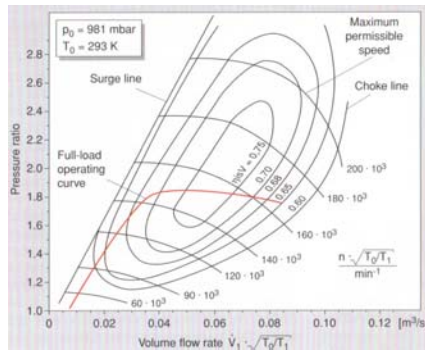
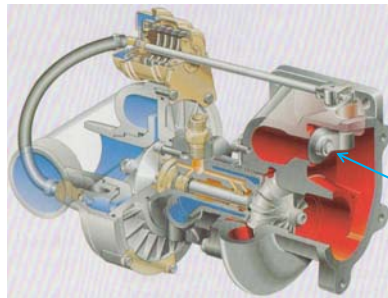
1.Compressor housing, 2. Compressor impeller, 3. Turbine housing, 4. Rotor, 5. Bearing housing, 6. inflowing exhaust gas, 7. Out-flowing exhaust gas, 8. Atmospheric fresh air, 9. Pre-compressed fresh air, 10. Oil inlet, 11. Oil return



From Bosch Automotive Handbook

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Turbo-charger

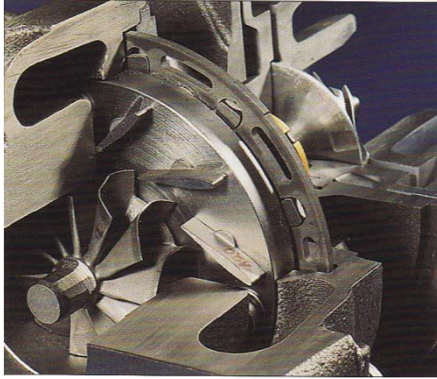


Waste gate

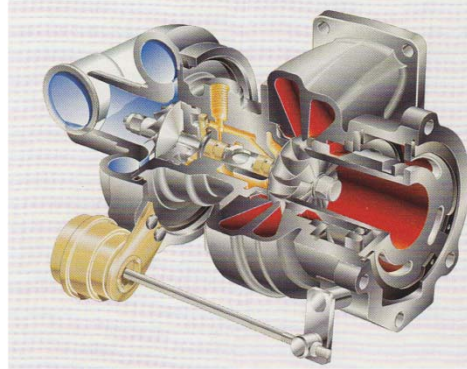
Source: BorgWarner Turbo Systems

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Variable geometry turbo-charger



Variable Guide Vane

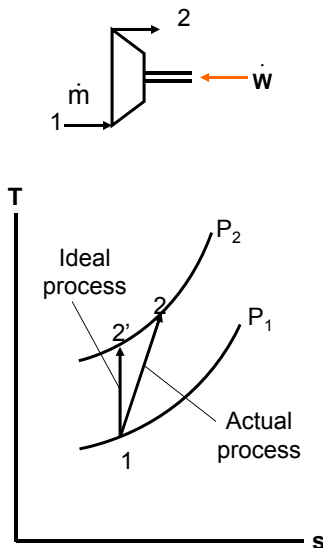


Variable sliding ring

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Compressor: basic thermodynamics



Compressor efficiency η_c

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

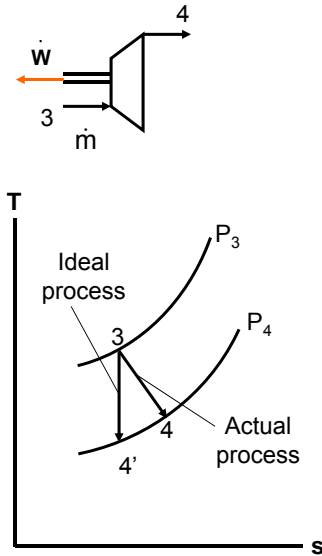
$$\dot{W}_{ideal} = \dot{m} c_p T_1 \left(\frac{T_2'}{T_1} - 1 \right)$$

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

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

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

Turbine: basic thermodynamics



Turbine efficiency η_t

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

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

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

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

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

Properties of Turbochargers

- Power transfer between fluid and shaft $\propto \text{RPM}^3$
 - Typically operate at $\sim 60\text{K}$ to 120K RPM
- RPM limited by centrifugal stress: usually tip velocity is approximately sonic
- RPM also limited by shock waves
- 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

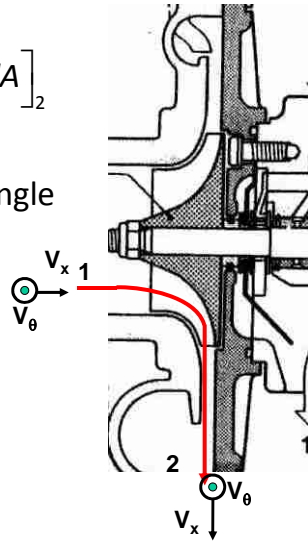
$$\text{Torq} = \left[\int (rV_\theta) \rho V_x dA \right]_1 - \left[\int (rV_\theta) \rho V_x dA \right]_2$$

$$\propto V_x V_\theta$$

both V_x and V_θ are fixed by the blade angle
so that both are $\propto \text{RPM}$, therefore:

$$\text{Torq} \propto (\text{RPM})^2$$

$$\text{Power} \propto (\text{RPM})^3$$



Rotor stress

Force balance over mass element from r to dr

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

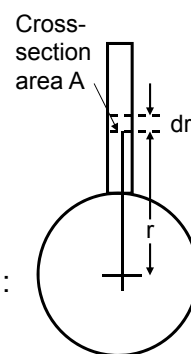
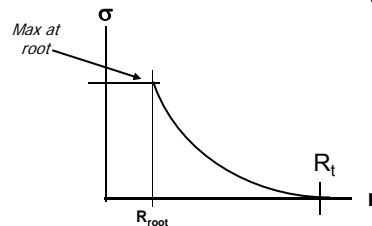
or

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

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

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

- σ Tensile stress
- ρ_m Material density
- ω 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

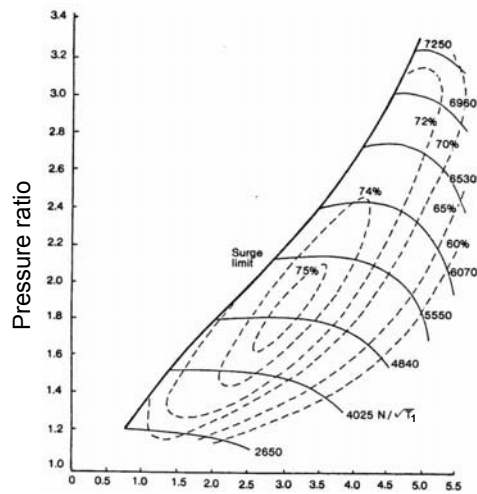
- Delivered pressure P_2
- $P_2 = f(\dot{m}, RT_1, P_1, N, D, \mu, \gamma, \text{geometric ratios})$
- Dimensional analysis:
 - 7 dimensional variables $\rightarrow (7-3) = 4$ dimensionless parameters (plus γ and geometric ratios)

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

High Re number flow \rightarrow 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



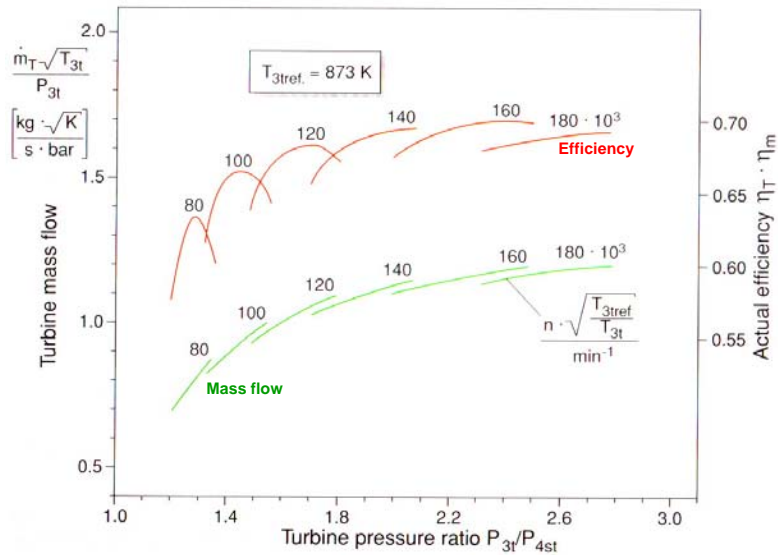
"Corrected" Flow rate $\dot{m} \sqrt{T_1}/P_1$

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_θ/V_x)
 - Stall causes flow blockage
- Surge
 - Flow inertia/resistance, and compression system internal volume comprise a LRC resonance system
 - Oscillatory flow behavior when flow blockage occurs because of compressor stall
 - reverse flow and violent flow rate surges

Turbine Map

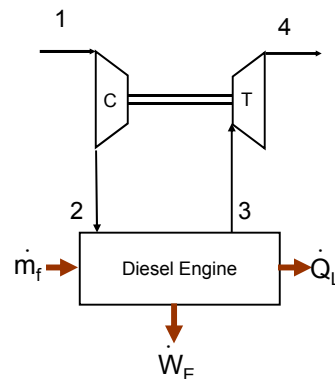


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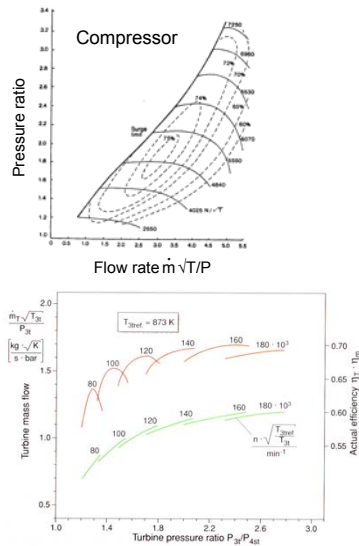
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Compressor Turbine Matching Exercise

- For simplicity, take away intercooler and wastegate
- Given engine brake power output (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 (π_c and π_t) etc.



Compressor/
turbine/engine matching
solution



Procedure:

1. Guess π_c ; can get engine inlet conditions:

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

2. Then engine volumetric efficiency calibration will give the air flow \dot{m}_a that can be 'swallowed'

3. From \dot{m}_a and π_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:

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

5. Engine exhaust temperature T_3 may be obtained from energy balance (with known engine mech. eff. η_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 π_t , then get turbine speed N_t from turbine map and mass flow

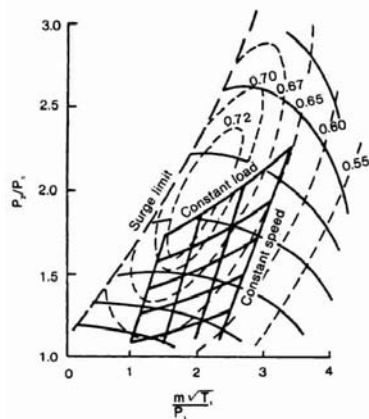
7. Determine turbine power from turbine efficiency on map

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

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

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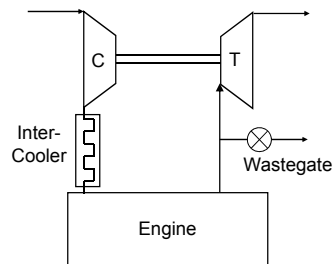
Compressor/ Engine/ Turbine Matching



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

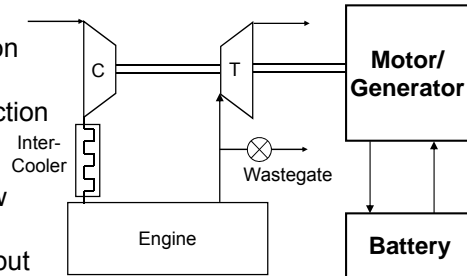


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Advanced turbocharger development

Electric assisted turbo-charging

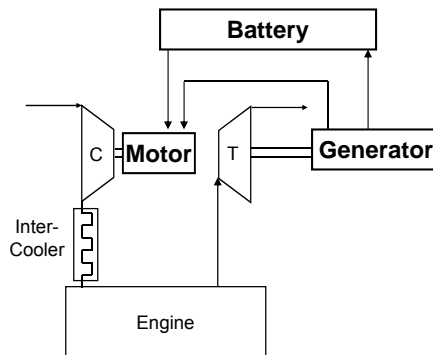
- **Concept**
 - Put motor/ generator on turbo-charger
 - reduce wastegate function
- **Benefit**
 - increase air flow at low engine speed
 - auxiliary electrical output at part load



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

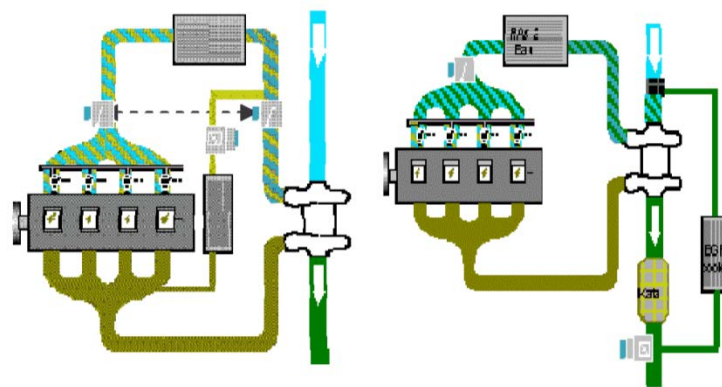


Advanced turbocharger development

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

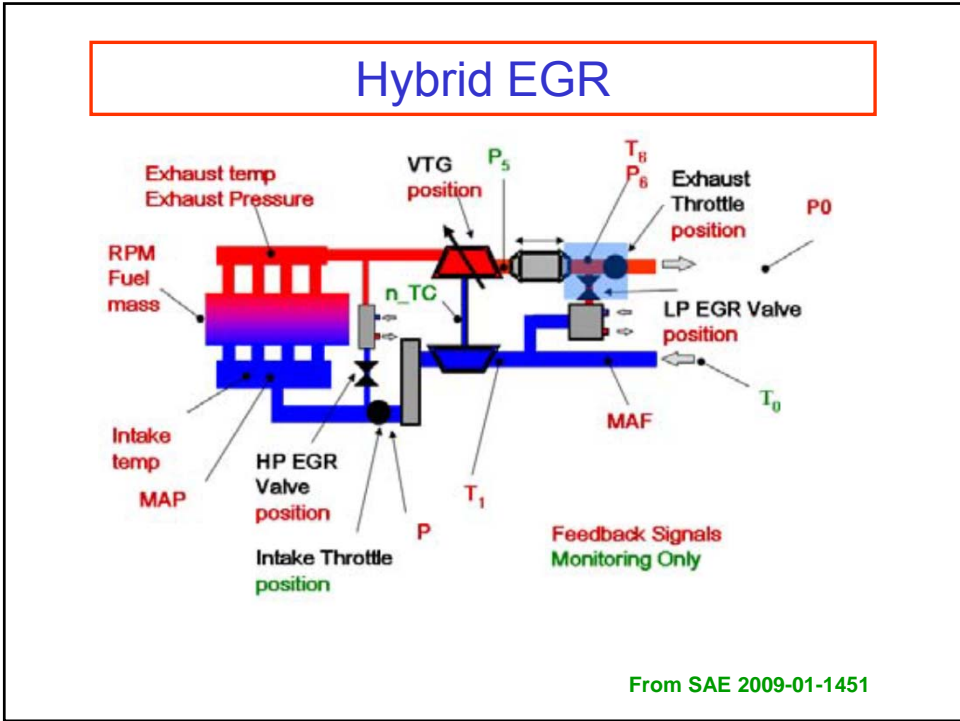
EGR/ turbo Configurations



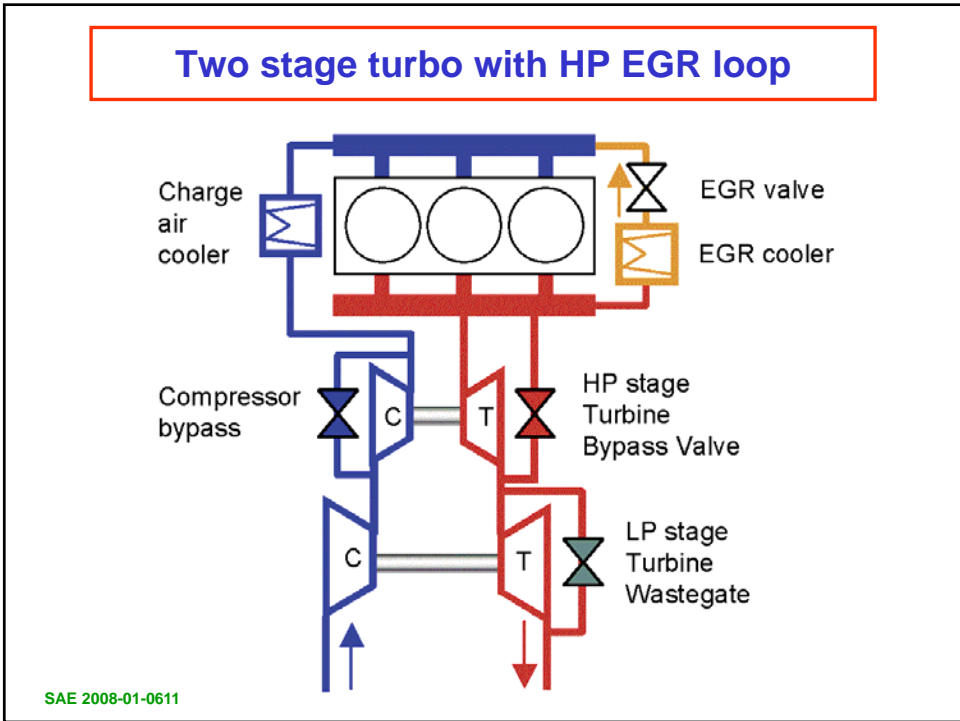
High pressure architecture

Low pressure architecture

From SAE 2007-01-2978



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