Engine Turbo/Super Charging

Super and Turbo-charging

Why super/ turbo-charging?
• Fuel burned per cycle in an IC engine is air limited
  \((F/A)_{\text{stoich}} = 1/14.6\)

\[
\text{Torq} = \frac{\eta_f m_f Q_{HV}}{2\pi \eta_R}
\]

\[
\text{Power} = \text{Torq} \cdot 2\pi N
\]

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

\(\eta_f, \eta_v, \eta_m\) – fuel conversion and volumetric efficiencies

\(m_f\) – fuel mass per cycle

\(Q_{HV}\) – fuel heating value

\(\eta_R\) – 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 NOₓ 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

Spark retard/enrichment for SI; smoke limit for diesel

Relative efficiency = 1

Combustion speed, pumping loss

BMEP (bar)

Heat transfer

Throttle + ht transf + friction

BMEP (bar)

Engine speed (rpm)

Data from SAE 910676
Saturn I4 engine

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

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

From Bosch Automotive Handbook
Variable geometry turbo-charger

Variable Guide Vane

Variable sliding ring

Compressor: basic thermodynamics

Compressor efficiency $\eta_c$

$\eta_c = \frac{W_{\text{ideal}}}{W_{\text{actual}}}$

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

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

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

$T_2 = T_1 + \frac{W_{\text{actual}}}{\dot{m} c_p}$
Turbine: basic thermodynamics

Turbine efficiency $\eta_t$

$\eta_t = \frac{W_{\text{actual}}}{W_{\text{ideal}}}$

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

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

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

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

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
- 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 \( V_\theta \) are fixed by the blade angle so that both are \( \propto \text{RPM} \), therefore:

\[ \text{Torq} \propto (\text{RPM})^2 \]

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

---

**Rotor stress**

Force balance over mass element from \( r \) to \( dr \)

\[ (\sigma A)_r - (\sigma A)_{r+dr} = \rho_m A \frac{d}{r} \left( \frac{\omega r^2}{r} \right) \]

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 \( \approx 2.5 \)
• BMEP up to 24 bar
• Limits:
  – compressor aerodynamics
  – cylinder peak pressure
  – NOx emissions

Compressor/Turbine Characteristics

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

\[
\left( \frac{P_2}{P_1} \right) = f\left( \frac{N}{\sqrt{RT_1/D}}, \frac{m}{RT_1\sqrt{D^2}}, \frac{P_1}{RT_1}, \text{Re, } \gamma, \text{ geometric ratios} \right)
\]

High Re number flow \( \rightarrow \) weak Re dependence
For fixed geometry machinery and gas properties

\[
\left( \frac{P_2}{P_1} \right) = f\left( \frac{N}{\sqrt{T_1}}, \frac{m\sqrt{T_1}}{P_1} \right)
\]
Compressor Map

\[ \text{Corrected} \text{ Flow rate } m \sqrt{\frac{T_1}{P_1}} \]

- \( T_1 = \text{inlet temperature (K)} \)
- \( P_1 = \text{inlet pressure (bar)} \)
- \( N = \text{rev. per min.} \)
- \( m = \text{mass flow rate (kg/s)} \)

(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 \( V_0/V_x \))
  - 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
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 ($m_a$), fuel flow rate ($m_f$), turbo-shaft revolution per second (N), compressor and turbine pressure ratios ($\pi_c$ and $\pi_t$) etc.
Compressor/ Engine/ Turbine Matching

• 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

Compressor characteristics, with airflow requirements of a four-stroke truck engine superimposed. (From “Principles and Performance in Diesel Engineering,” Ed. by Haddad and Watson)
**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

**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

From SAE 2007-01-2978
Hybrid EGR

From SAE 2009-01-1451

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Two stage turbo with HP EGR loop

SAE 2008-01-0611

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