Part 2: Turbochargers, Engine Performance Metrics

Reciprocating Internal Combustion Engines

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Summer School on Combustion

Course Length: 15 hrs

(Mon.- Fri., June 23 – 27, 2014)

Short course outine:

Engine fundamentals and performance metrics, computer modeling supported by in-depth understanding of fundamental engine processes and detailed experiments in engine design optimization.

Day 1 (Engine fundamentals)
- Part 1: IC Engine Review, 0, 1 and 3-D modeling
- Part 2: Turbochargers, Engine Performance Metrics

Day 2 (Combustion Modeling)
- Part 3: Chemical Kinetics, HCCI & SI Combustion
- Part 4: Heat transfer, NOx and Soot Emissions

Day 3 (Spray Modeling)
- Part 5: Atomization, Drop Breakup/Coalescence
- Part 6: Drop Drag/Wall Impinge/Vaporization/Sprays

Day 4 (Engine Optimization)
- Part 7: Diesel combustion and SI knock modeling
- Part 8: Optimization and Low Temperature Combustion

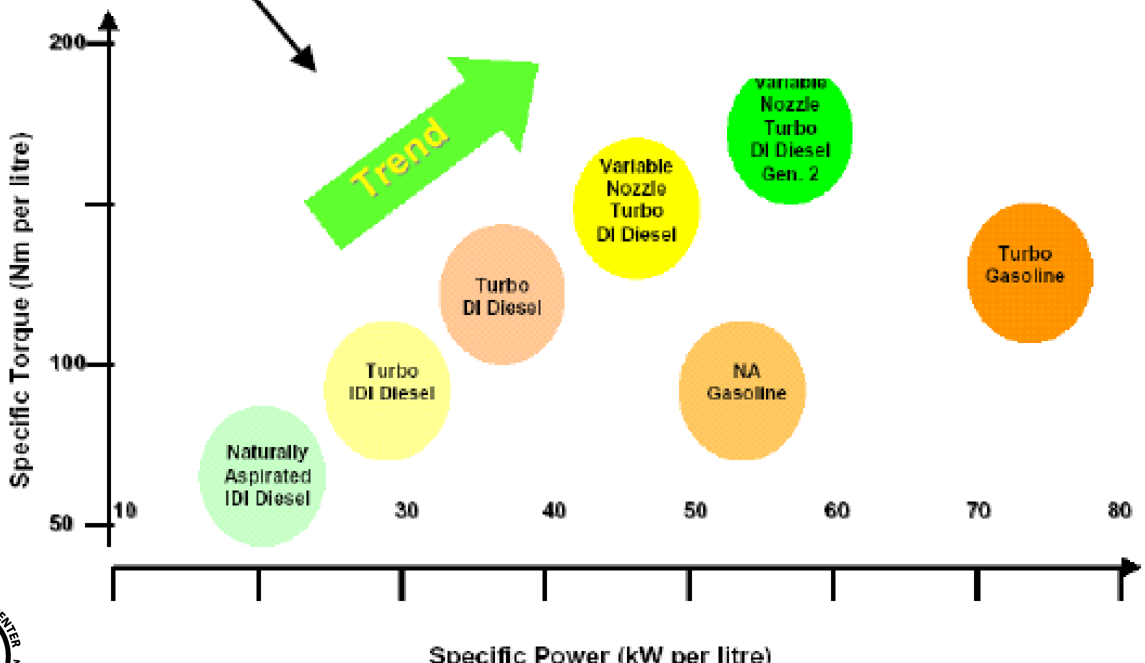
Day 5 (Applications and the Future)
- Part 9: Fuels, After-treatment and Controls
- Part 10: Vehicle Applications, Future of IC Engines

Turbocharging

Pulse-driven turbine was invented and patented in 1925 by Büchi to increase the amount of air inducted into the engine.

- Increased engine power more than offsets losses due to increased back pressure
- Need to deal with turbocharger lag

Power & Torque Trends for Diesel & Gasoline



Improved

- Fuel economy
- Torque
- Power density

Turbocharging

Purpose of turbocharging or supercharging is to increase inlet air density,
- increase amount of air in the cylinder.

Mechanical supercharging
- driven directly by power from engine.

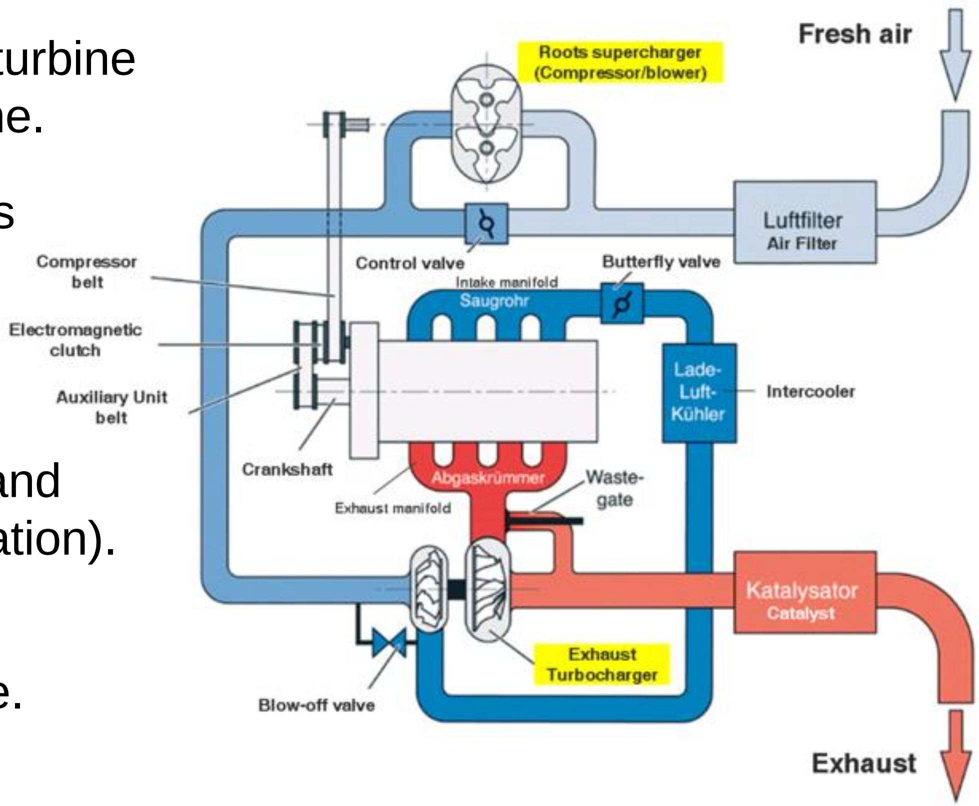
Turbocharger - connected compressor/turbine
- energy in exhaust used to drive turbine.

Supercharging necessary in two-strokes for effective scavenging:
- intake P > exhaust P
- crankcase used as a pump

Some engines combine engine-driven and mechanical (e.g., in two-stage configuration).

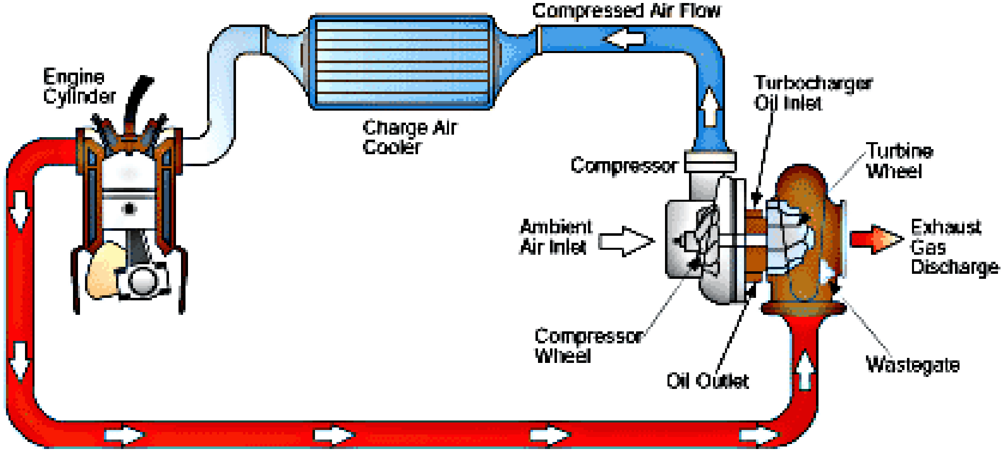
Intercooler after compressor
- controls combustion air temperature.

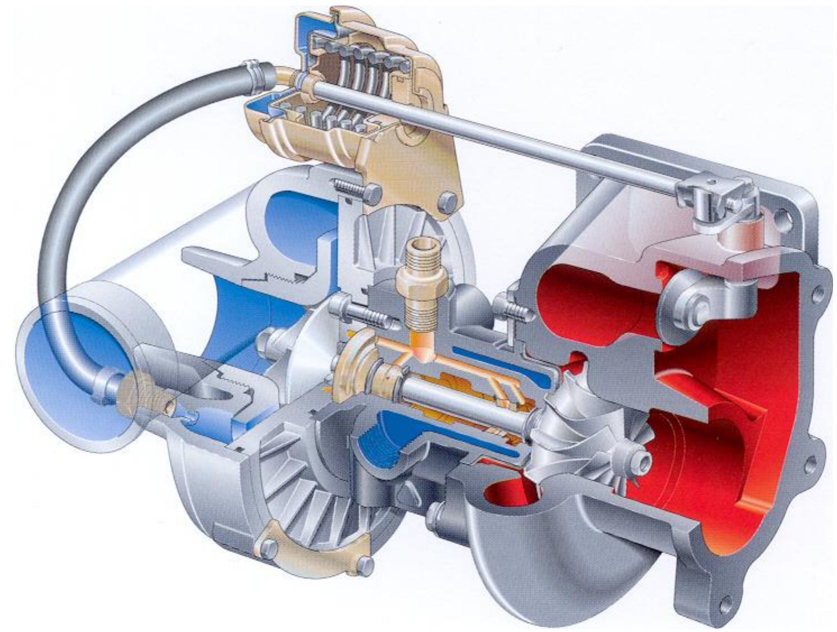
Air Flow in the VW Twincharged TSI

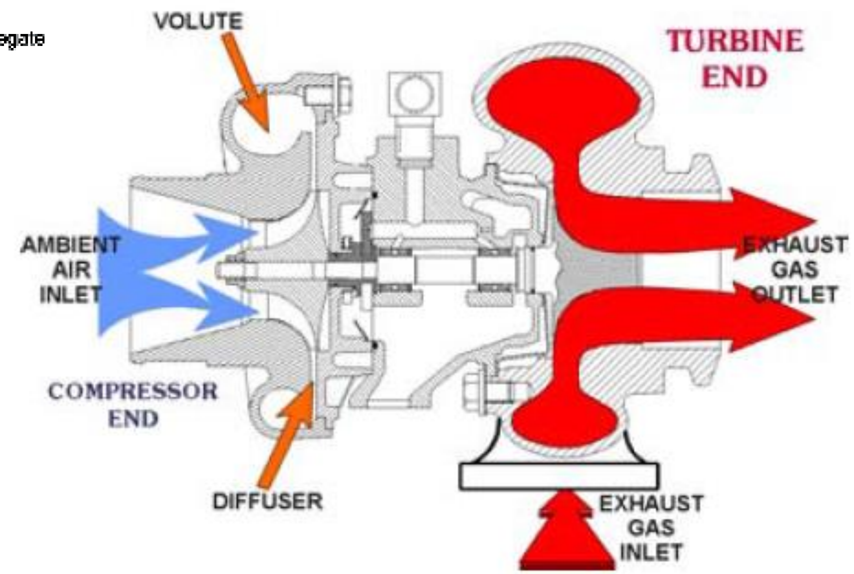


Turbocharging

Energy in exhaust is used to drive turbine which drives compressor





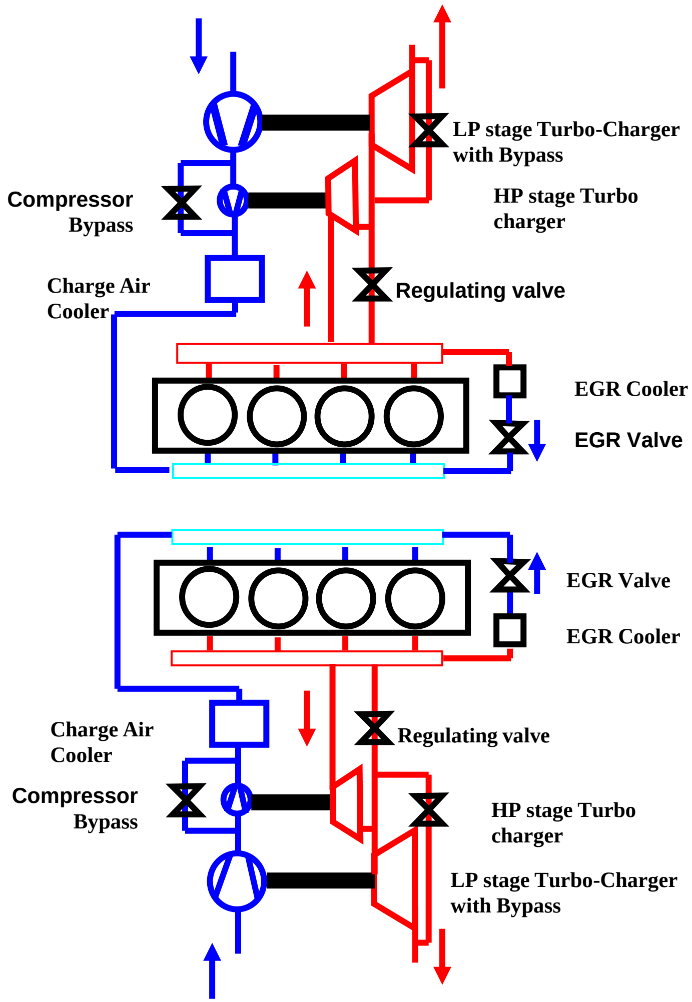


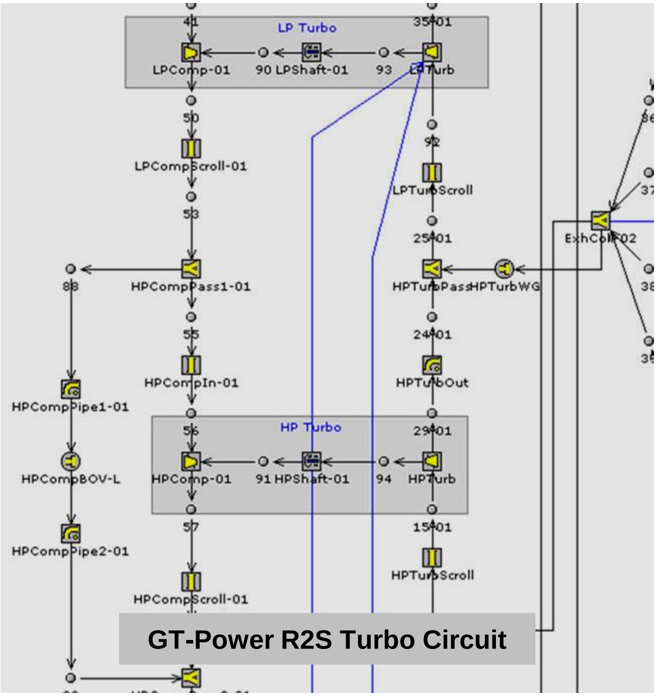
Wastegate used to by-pass turbine

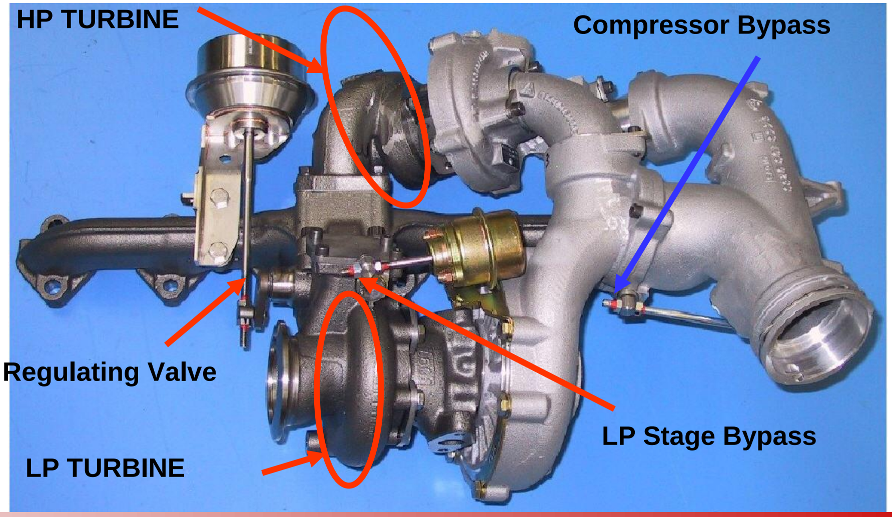
Charge air cooling after compressor further increases air density
- more air for combustion

Regulated two-stage turbocharger

Duplicated Configuration per Cylinder Bank

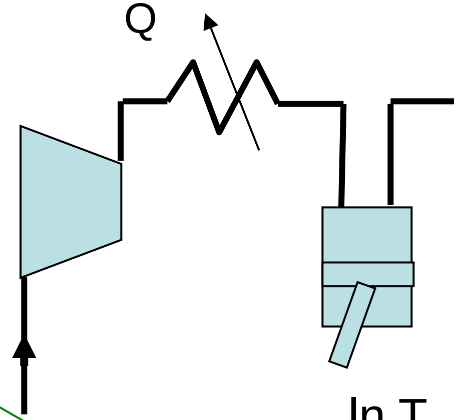


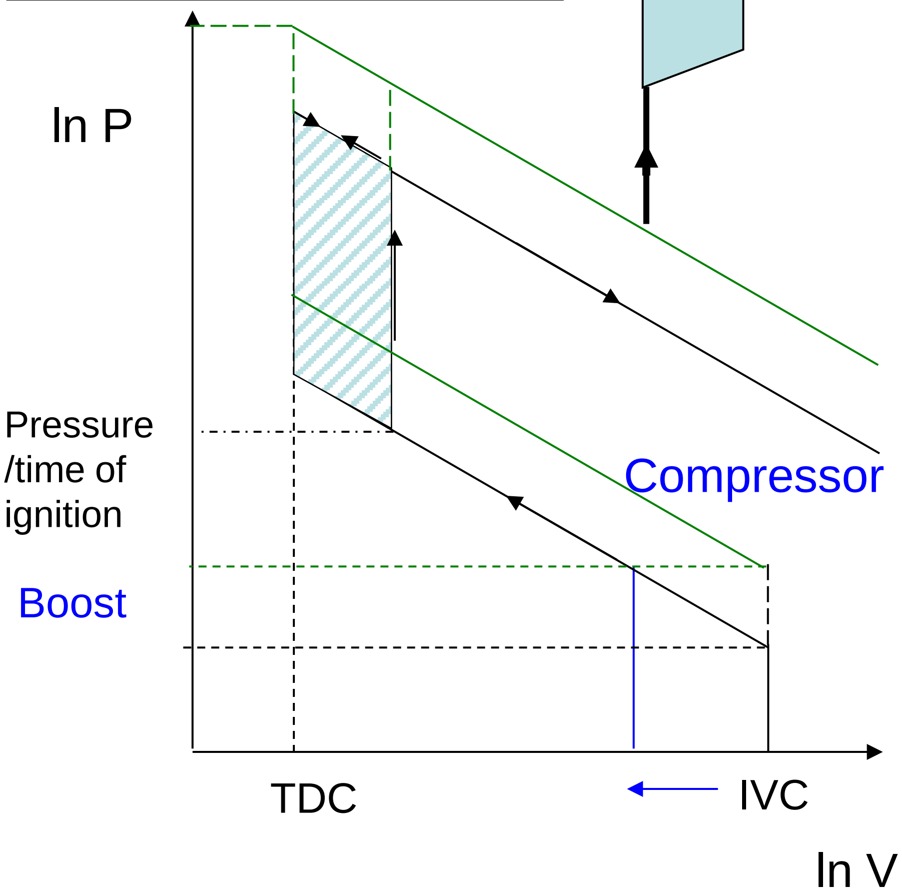




Intercooler for IVC temperature control

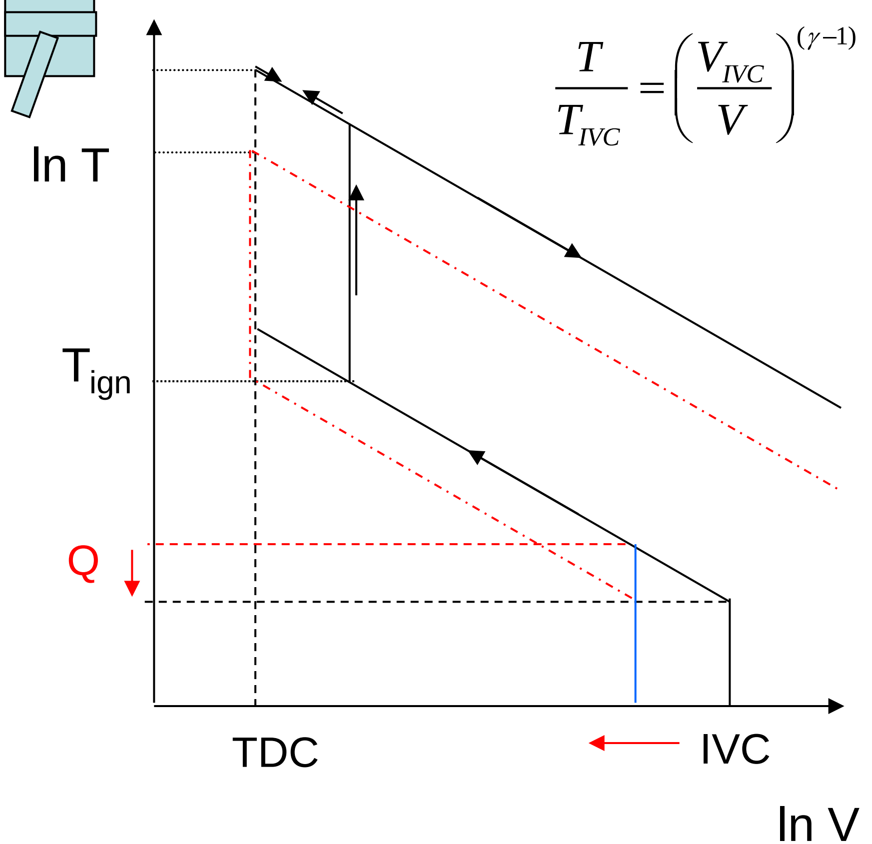
Isentropic $\frac{P}{P_{IVC}} = \left(\frac{V_{IVC}}{V}\right)^{\gamma}$





Reduced Peak Temp (NOx)
Improved phasing

$$\frac{T}{T_{IVC}} = \left(\frac{V_{IVC}}{V}\right)^{(\gamma-1)}$$



Boost explains 20% of the improved fuel efficiency of diesel vs. SI

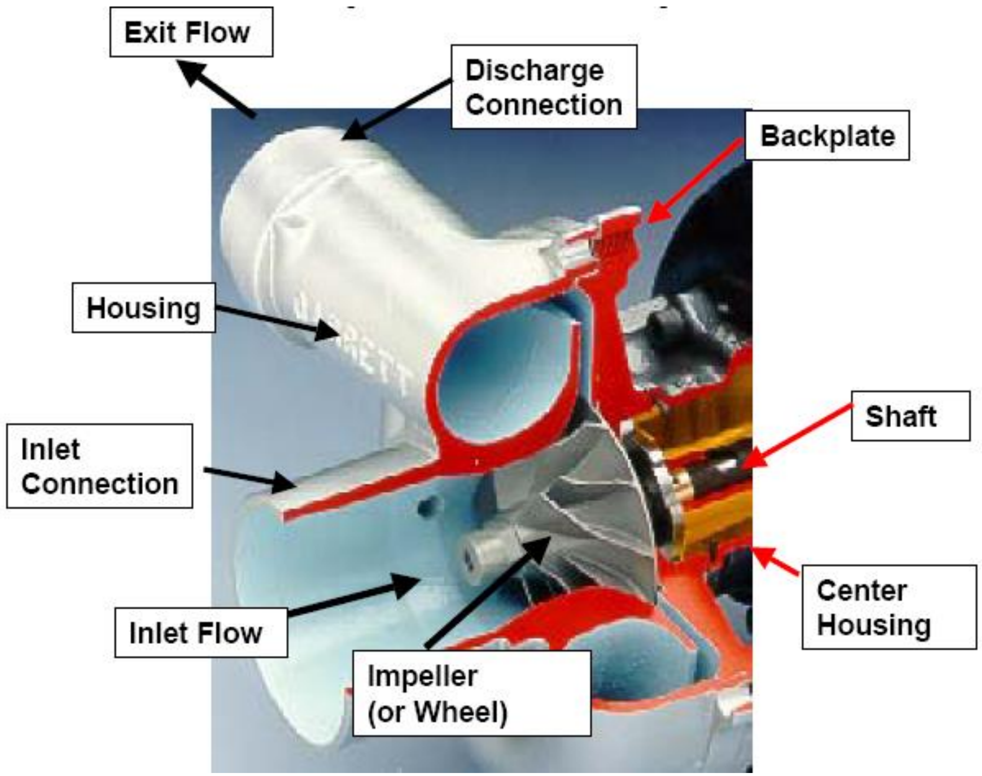
Automotive compressor

Centrifugal compressor typically used in automotive applications

Provides high mass flow rate at relatively low pressure ratio ~ 3.5

Rotates at high angular speeds
- direct coupled with exhaust-driven turbine
- less suited for mechanical supercharging

Consists of:
  stationary inlet casing,
  rotating bladed impeller,
  stationary diffuser (w or w/o vanes)
  collector - connects to intake system



Anderson, 1990

<u>Compressible flow</u> – A review

Gibbs ➔ $\cancel{Tds} = dh - dp/\rho$

Energy ➔ $dh = -VdV$

Euler ➔ $dP = -\rho VdV$

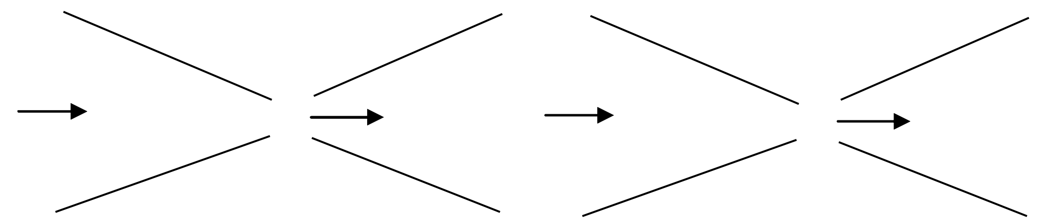
$\rho AV = Const$ ➔ $\dfrac{d\rho}{\rho} + \dfrac{dA}{A} + \dfrac{dV}{V} = 0$

$$\boxed{\frac{dA}{A} = (M^2 - 1)\frac{dV}{V}}$$

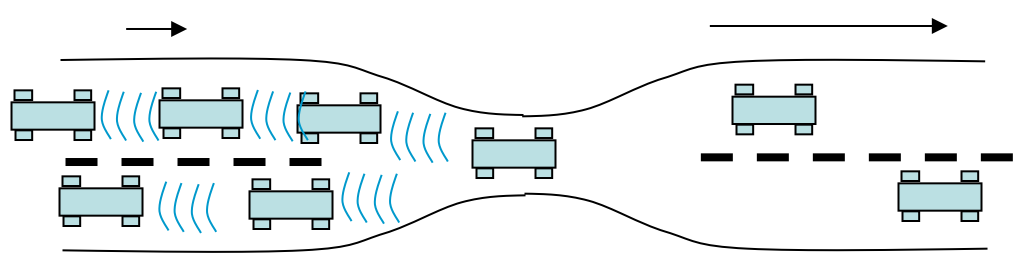
$$\boxed{\frac{dA}{A} = \frac{(1 - M^2)}{\rho V^2} dP}$$

Area-velocity relations

for *M<1* for *M>1*



	<u>Subsonic nozzle</u>	<u>Subsonic diffuser</u>	<u>Supersonic diffuser</u>	<u>Supersonic nozzle</u>
	$dA<0$	$dA>0$	$dA<0$	$dA>0$
from ρAV →	$dV>0$	$dV<0$	$dV<0$	$dV>0$
from Euler →	$dP<0$	$dP>0$	$dP>0$	$dP<0$
	kinetic energy	pressure recovery		kinetic energy



Traffic flow behaves like a supersonic flow!

Anderson, 1990

Model passages as compressible flow in converging-diverging nozzles

$$\dot{m} = \rho A V = \frac{P}{RT} A \frac{V}{c} \sqrt{\gamma R T}$$

$$= P_0 \sqrt{\frac{\gamma}{R T_0}} A M (P/P_0)/(T/T_0)^{-1/2}$$

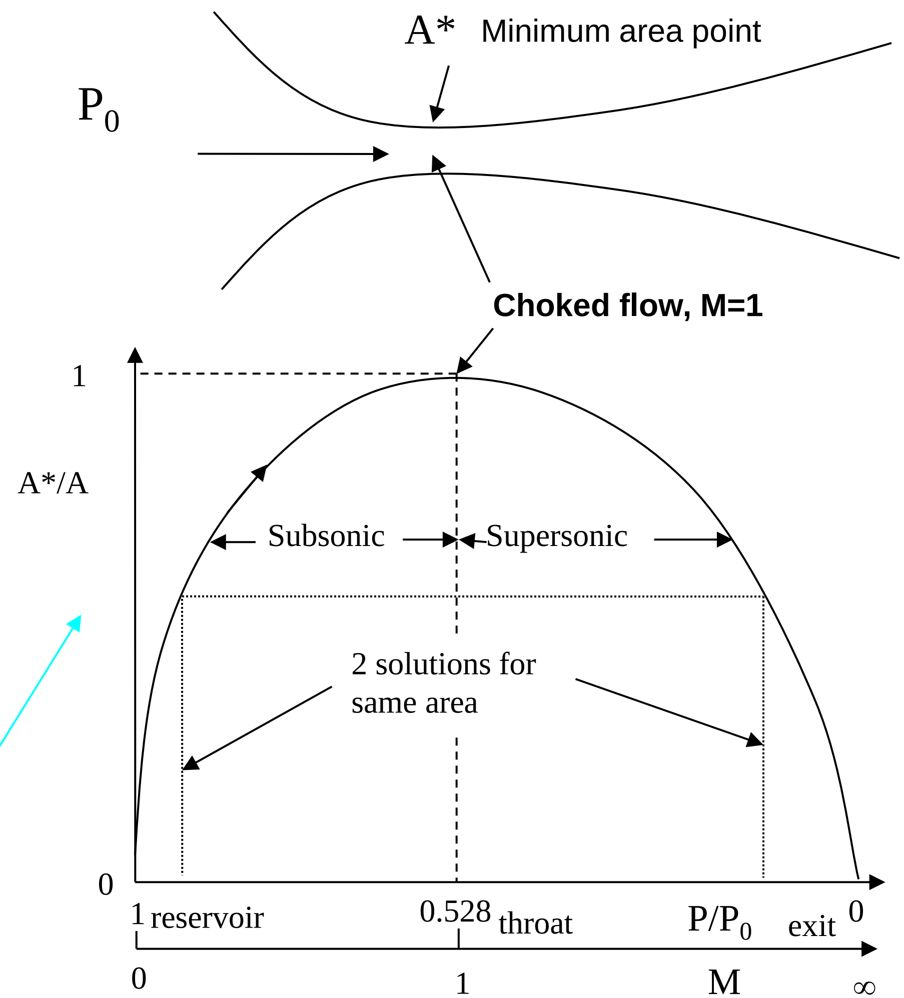
With *M=1*: Fliegner's formula

$$\boxed{\dot{m}_{M=1} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \sqrt{\frac{\gamma}{R T_0}} P_0 A^*}$$

Area Mach number relations

$$\frac{A}{A^*} = \frac{1}{M}\left\{\frac{2}{\gamma+1}\left(1+\frac{(\gamma-1)}{2}M^2\right)\right\}^{\frac{\gamma+1}{2(\gamma-1)}}$$

$$\frac{A}{A^*} = \left(\frac{P}{P_0}\right)^{\frac{1}{\gamma}}\left(\frac{2}{\gamma-1}\left[1-\left(\frac{P}{P_0}\right)^{\frac{\gamma-1}{\gamma}}\right]\left(\frac{\gamma+1}{2}\right)^{\frac{\gamma+1}{\gamma-1}}\right)^{1/2}$$

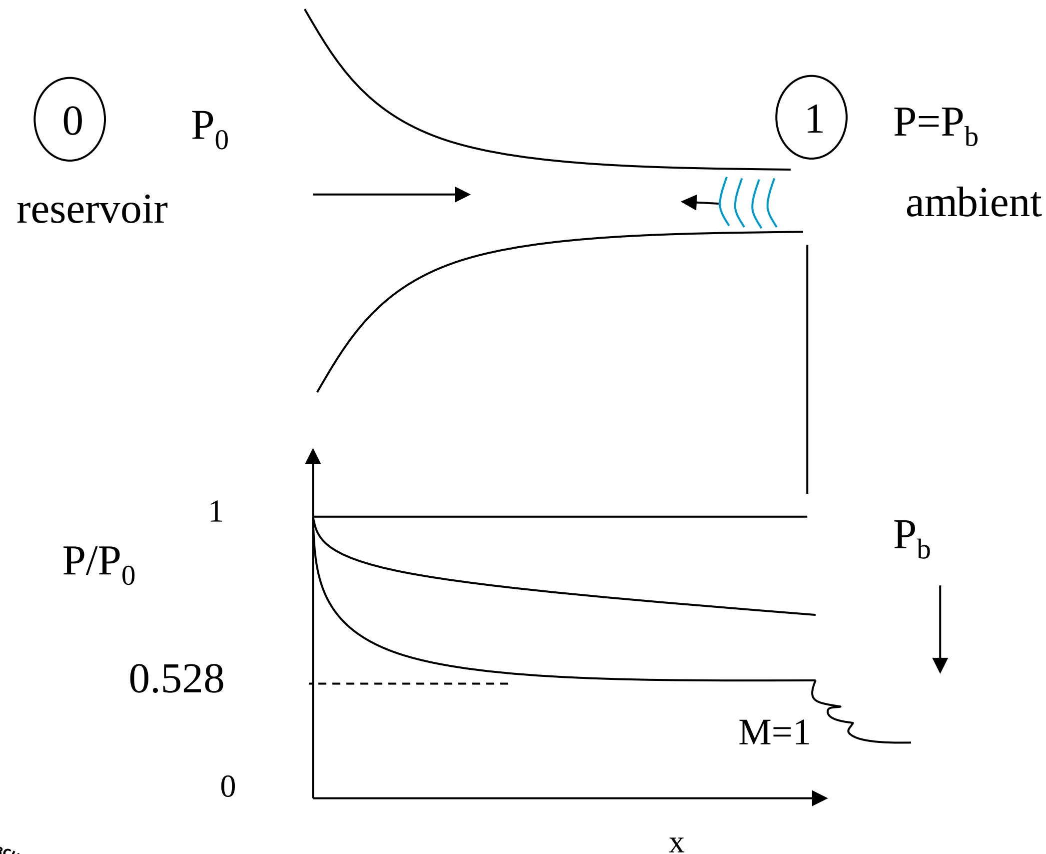


Anderson, 1990

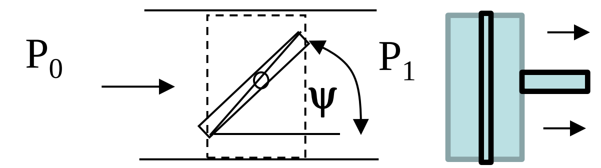
Isentropic nozzle flows

$$\frac{T_0}{T_1} = 1 + \frac{\gamma - 1}{2} M_1^{\ 2}$$

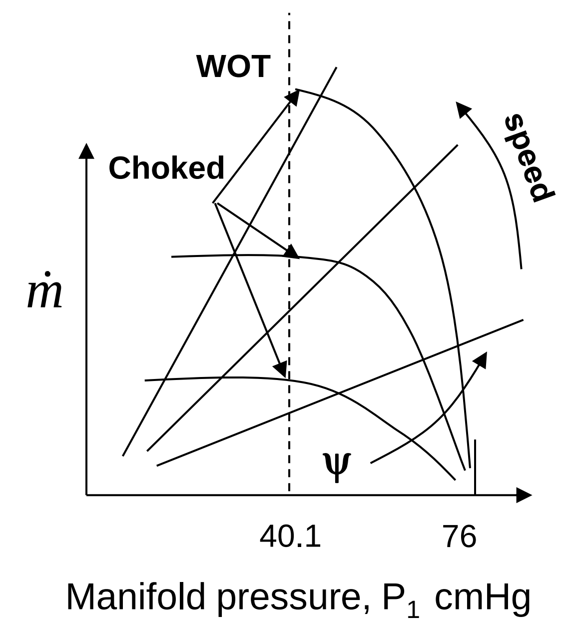
$$\frac{P_0}{P_1} = \left(1 + \frac{\gamma - 1}{2} M_1^{\ 2}\right)^{\frac{\gamma}{\gamma - 1}}$$



Ex. Flow past throttle plate



Choked flow for $P_2 < 53.5$ kPa = 40.1cmHg



Anderson, 1990

Application to turbomachinery

Fliegner's Formula:

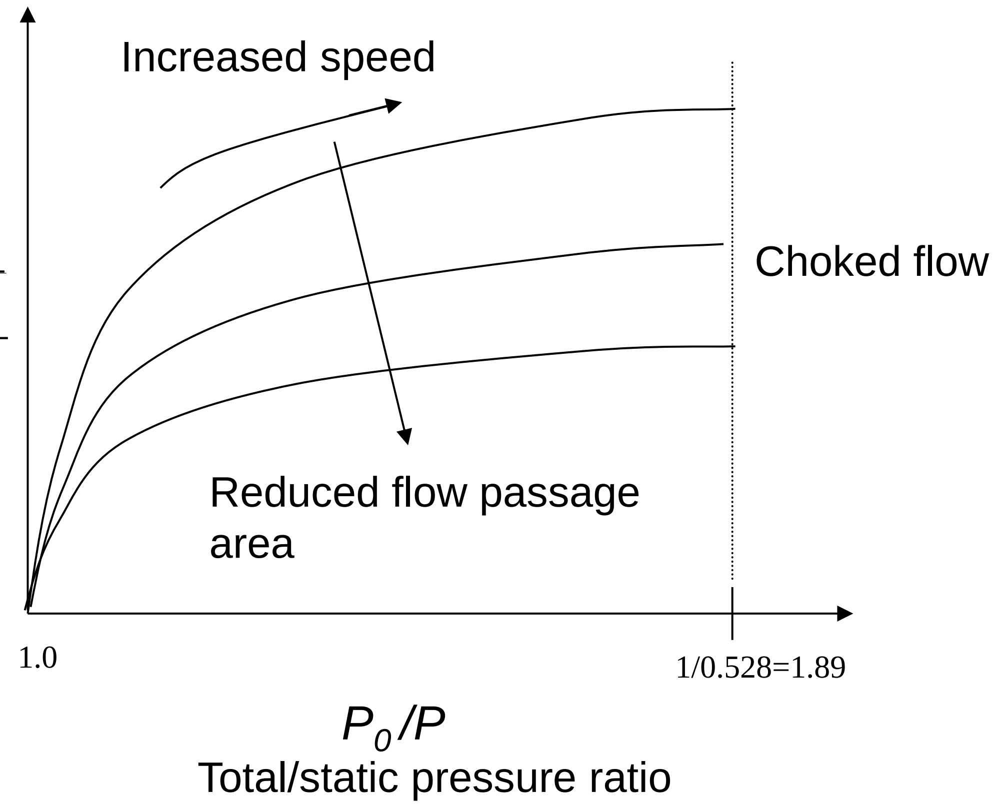
$$\dot{m}_{M=1} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \sqrt{\frac{\gamma}{RT_0}} P_0 A^*$$

Variable Geometry Compressor/ turbine performance map

"Corrected mass flow rate"

$$\frac{\dot{m}\sqrt{T_{ref}/T_0}}{P_0/P_{ref}}$$

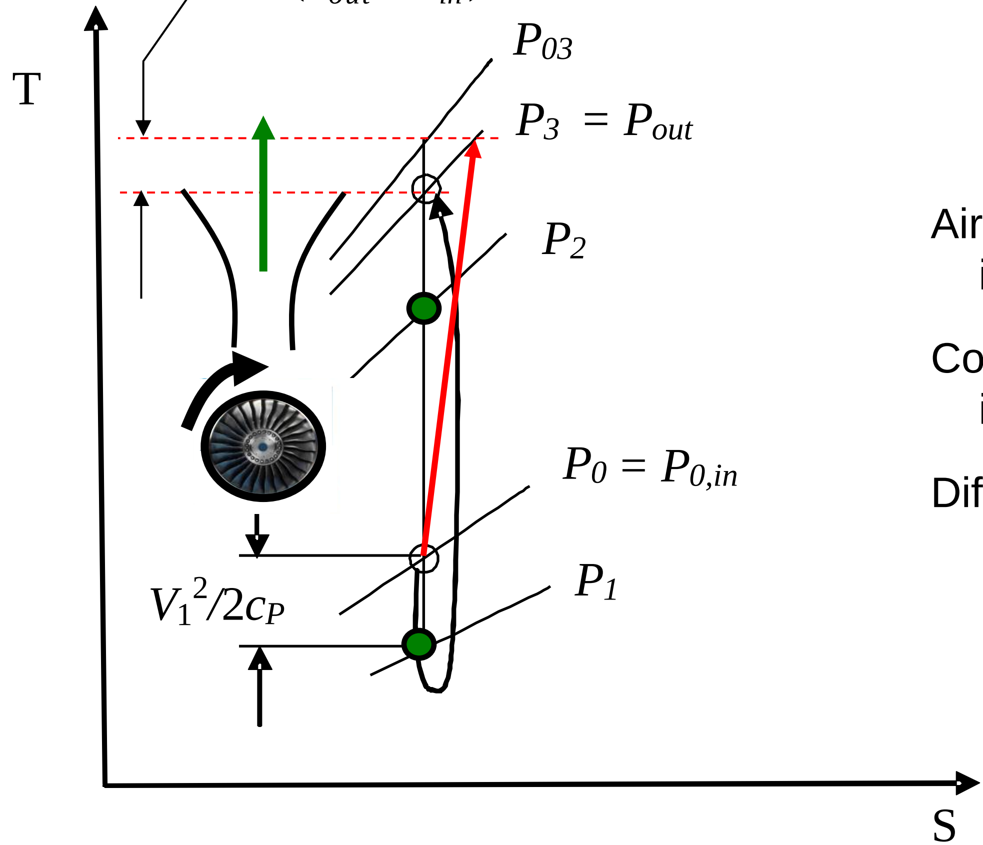
A measure of effective flow area

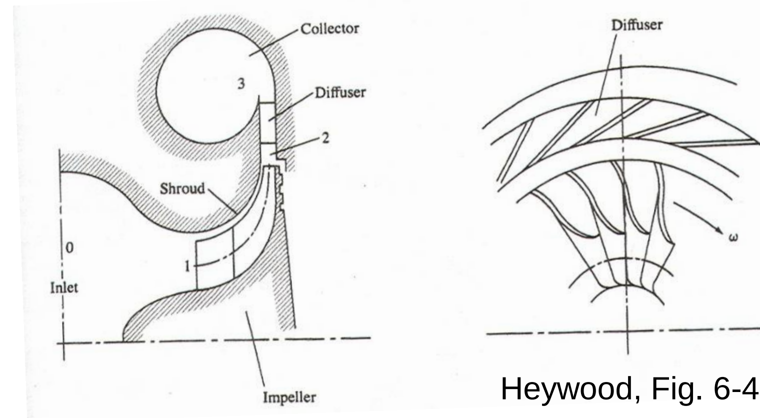


Heywood, 1988

Compressor

$$\eta_c = \frac{(T_{out-isen} - T_{in})}{(T_{out} - T_{in})}$$





Heywood, Fig. 6-43

Air at stagnation state 0,in accelerates to inlet pressure, P_1, and velocity V_1.

Compression in impeller passages increases pressure to P_2, and velocity V_2.

Diffuser between states 2 and out, recovers air kinetic energy at exit of impeller producing pressure rise to, P_{out} and low velocity V_{out}

$$\dot{W}_c = \dot{m}_a \left(h_{out} - h_{in} \right)$$

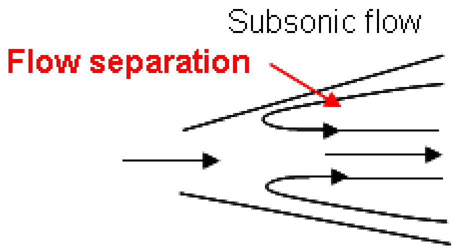
$$\dot{W}_c = \frac{\dot{m}_a \cdot c_{P_a} \cdot T_{in}}{\eta_c} \left(\left(\frac{p_{out}}{p_{0,in}} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right)$$

<u>Note:</u> use exit static pressure and inlet total pressure, because kinetic energy of gas leaving compressor is usually not recovered

Heywood, 1988

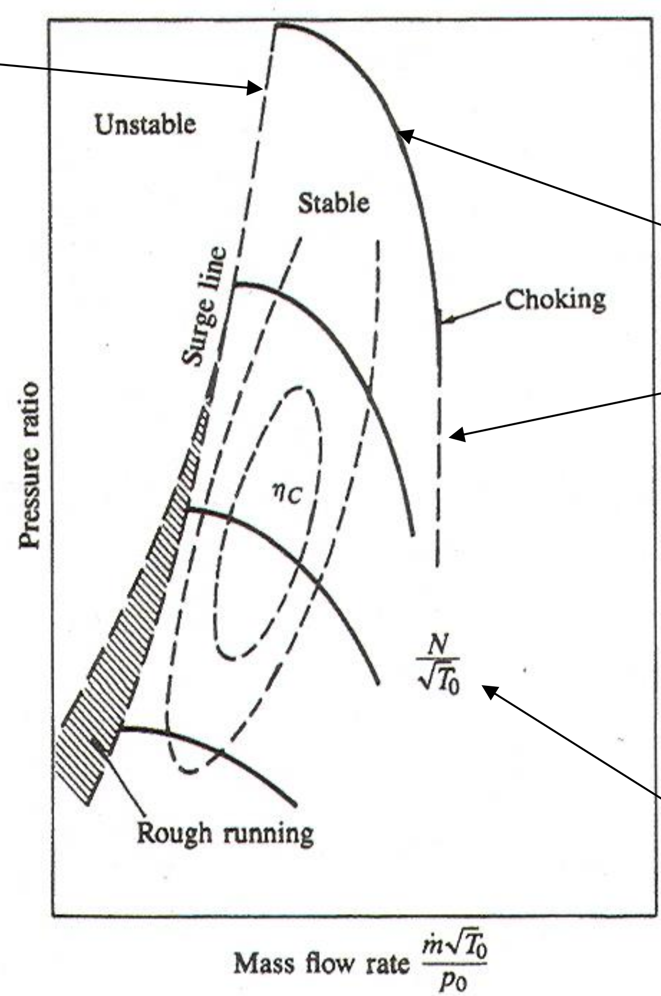
Compressor maps

Work transfer to gas occurs in impeller via change in gas angular momentum in rotating blade passage



Surge limit line
– reduced mass flow due to periodic flow reversal/reattachment in passage boundary layers. Unstable flow can lead to damage

Pressure ratio evaluated using total-to-static pressures since exit flow kinetic energy is not recovered

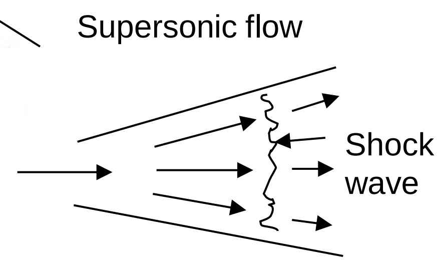


Heywood, Fig. 6-46

Speed/pressure limit line

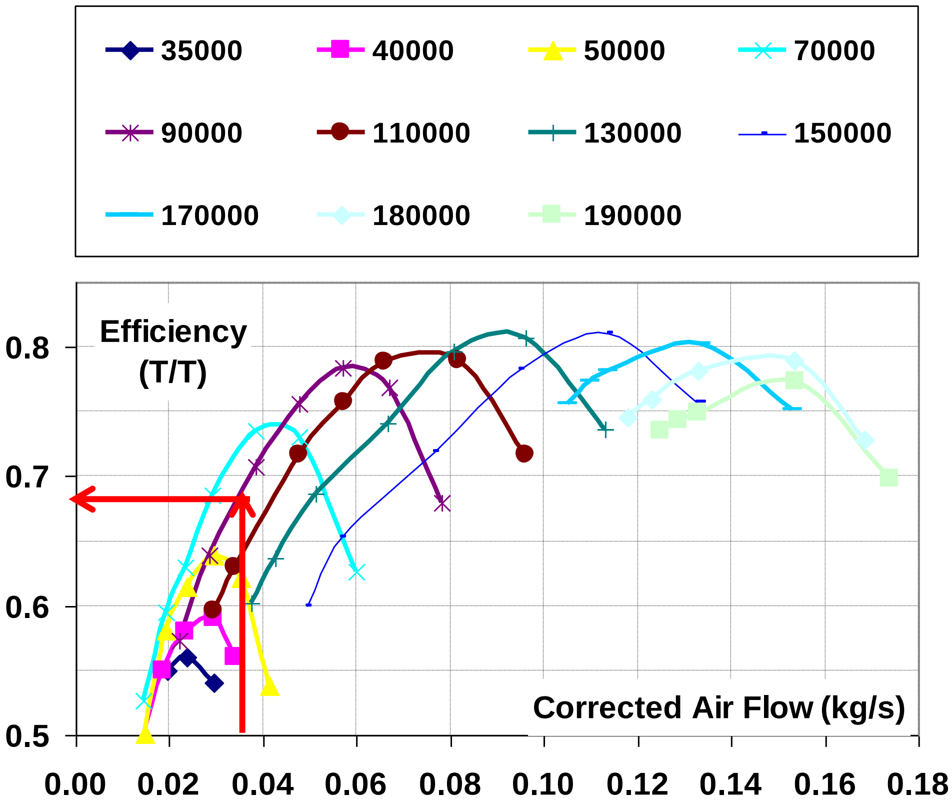
Non-dimensionalize blade tip speed (~ND) by speed of sound

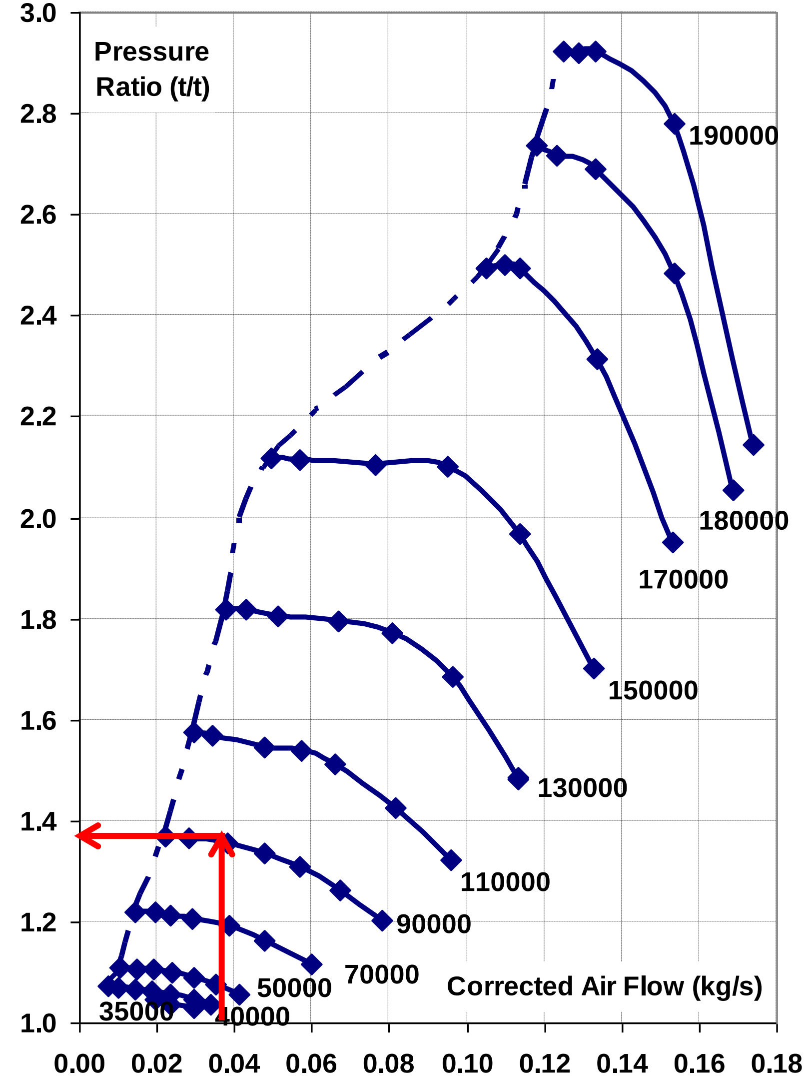
At high air flow rate, operation is limited by choking at the minimum area point within compressor



Compressor maps

GM 1.9L diesel engine





Automotive turbines

Naturally aspirated:

$P_{intake}=P_{exhst}=P_{atm}$ (5-7-8-9-1)

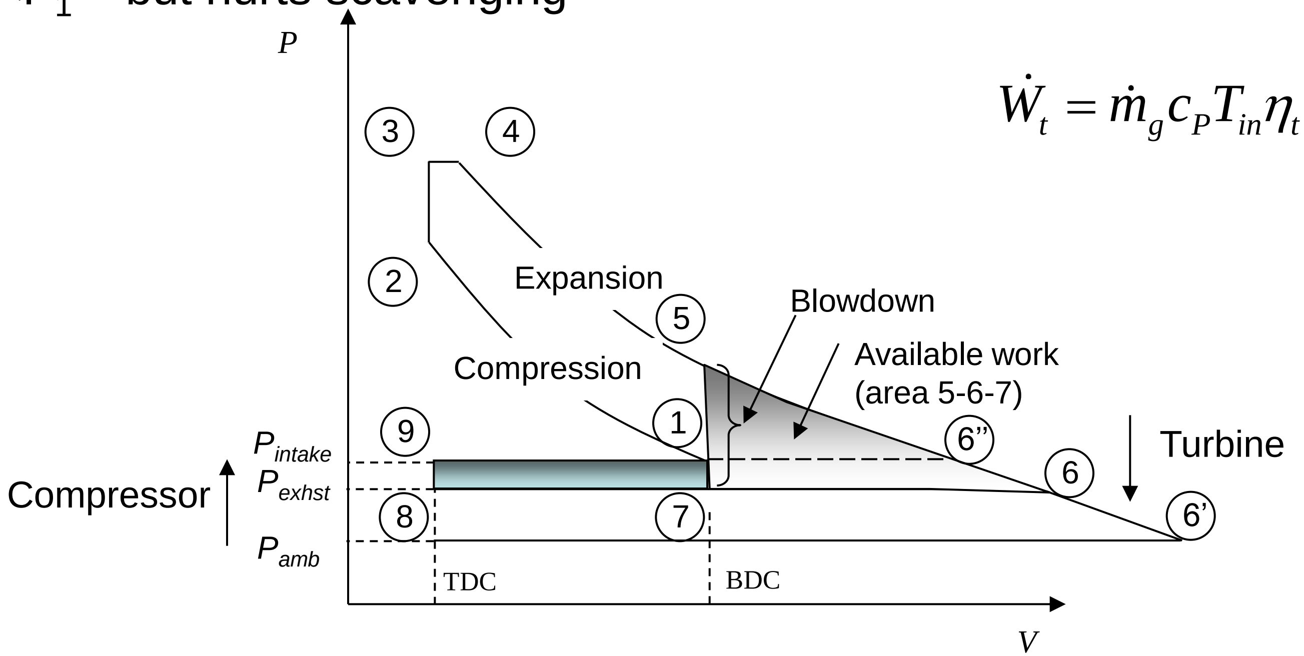
Boosted operation:

Negative pumping work:

$P_7<P_1$ – but hurts scavenging

$$\dot{W}_t = \dot{m}_g (h_{in} - h_{0,out})$$

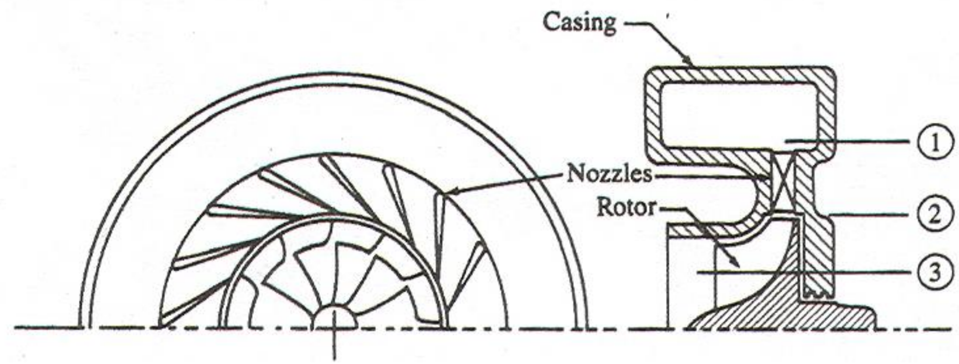
$$\dot{W}_t = \dot{m}_g c_P T_{in} \eta_t \left\{ 1 - \left[\frac{P_{0,out}}{P_{in}} \right]^{\frac{\gamma_g - 1}{\gamma_g}} \right\}$$

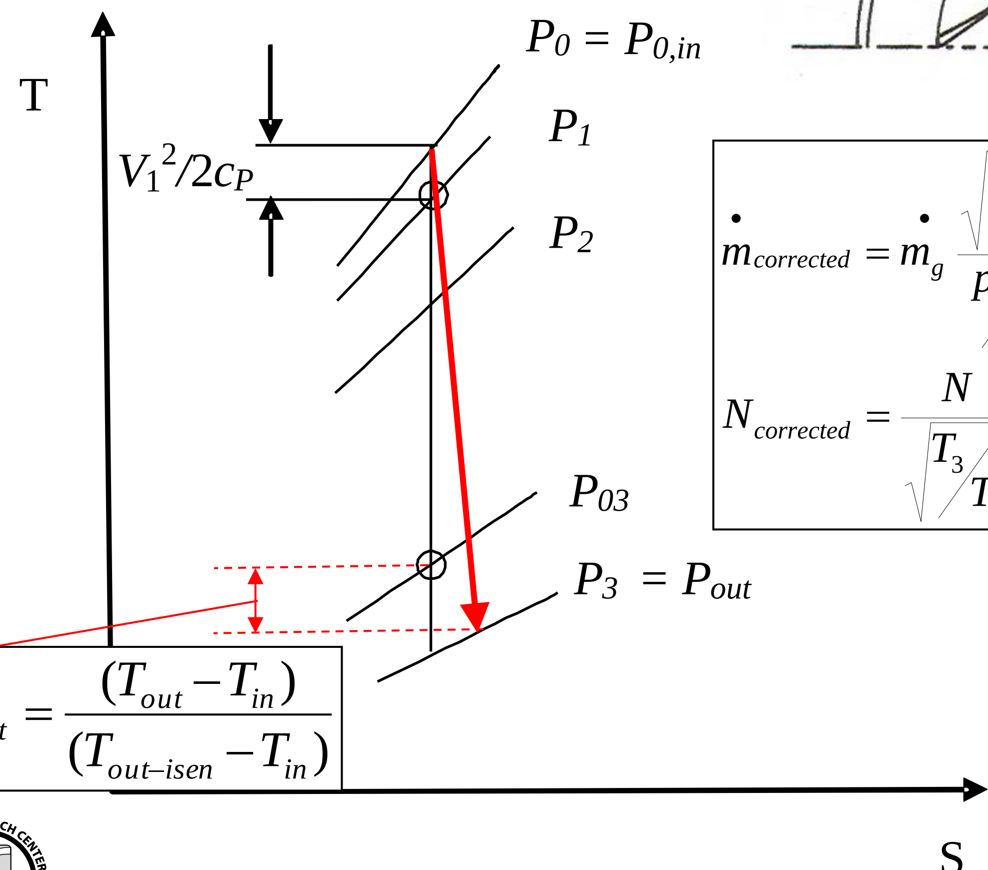


P-V diagram showing available exhaust energy
- turbocharging, turbocompounding, bottoming cycles and thermoelectric generators further utilize this available energy

Turbochargers

Radial flow – automotive;
axial flow – locomotive, marine

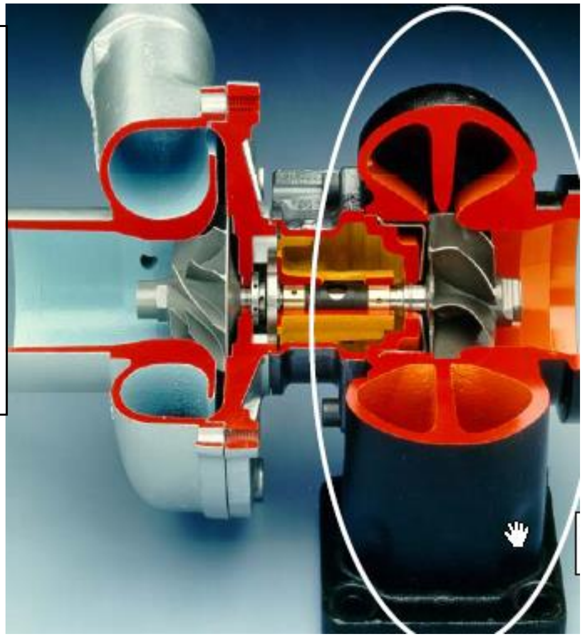




$$\eta_t = \frac{(T_{out} - T_{in})}{(T_{out-isen} - T_{in})}$$

$$\dot{m}_{corrected} = \dot{m}_g \frac{\sqrt{T_3 / T_0}}{p_3 / p_0}$$

$$N_{corrected} = \frac{N}{\sqrt{T_3 / T_0}}$$



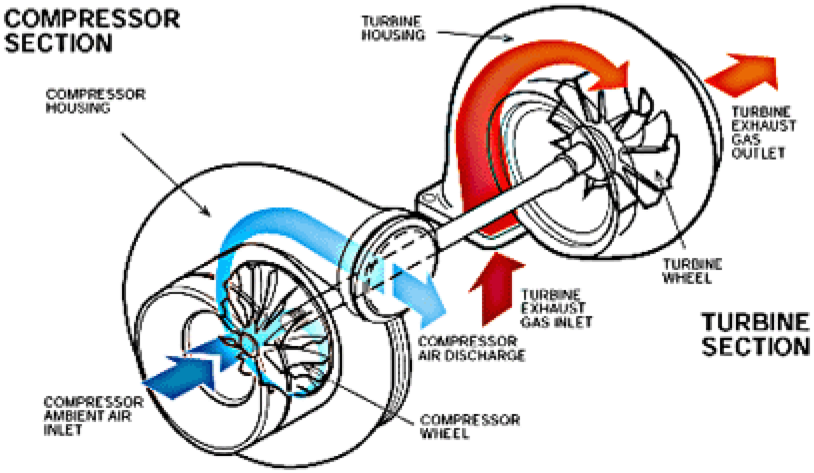
out

Compressor selection

To select compressor, first determine engine breathing lines.

The mass flow rate of air through engine for a given pressure ratio is:

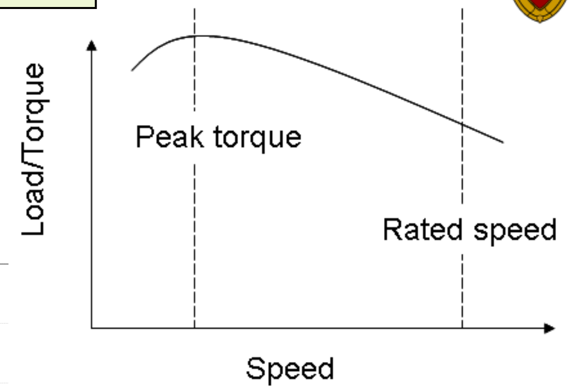
$$\dot{m}_{intake} = \left[\frac{\eta_{vol} \times D * N \times P_{ref}}{2 \times R \times T_{ref}} \right]$$

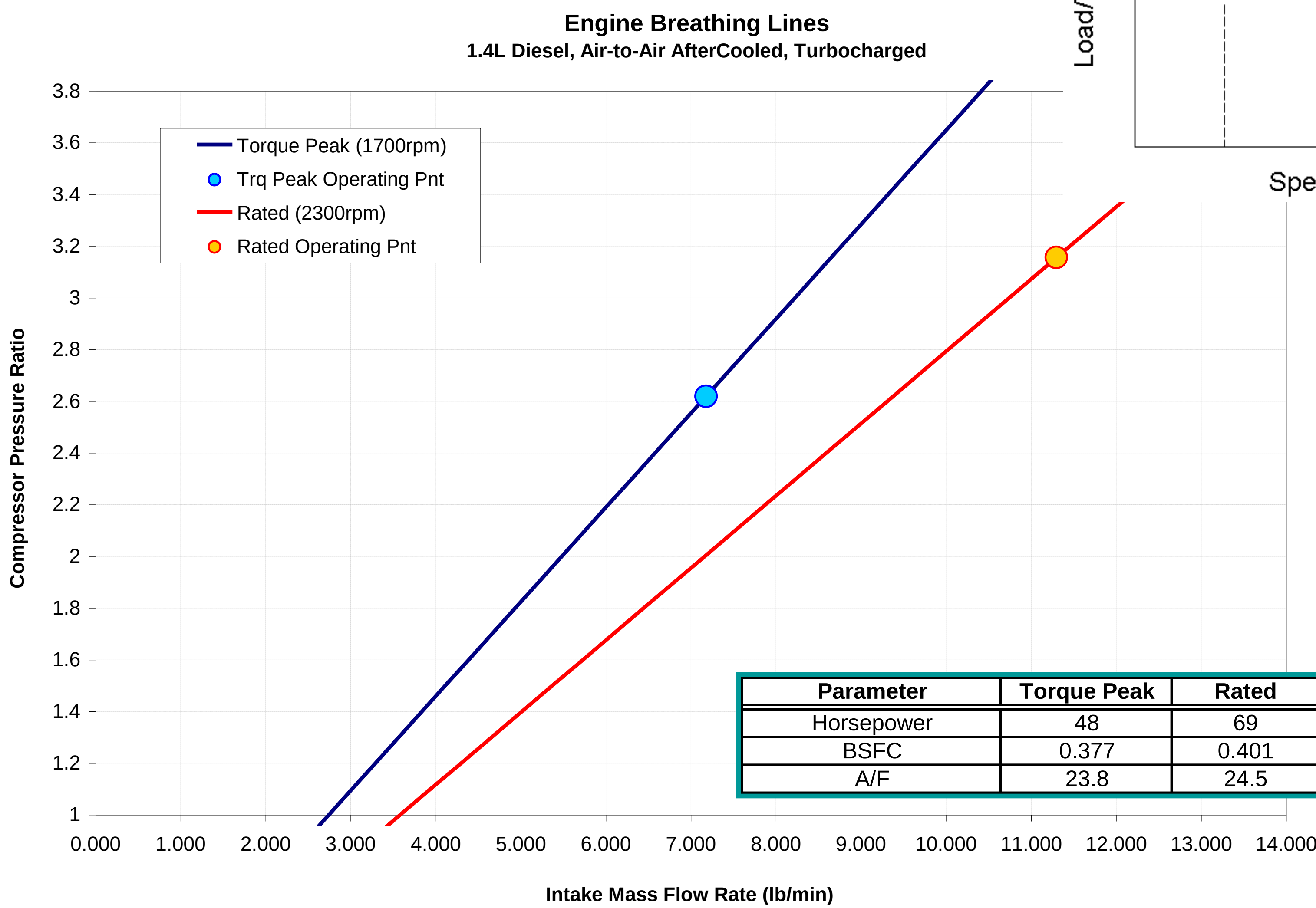


Where:

$\dot{m}_{intake}$ = Physical mass flow of air through engine (mass/time)
η_{vol} = Volumetric efficiency (unitless)
D = Displacement of engine per cycle ($length^3$/cycle)
N = Engine speed (rev/time)
P_{ref} = Reference pressure (psi) **= IMP = PR * atmospheric pressure (no losses)**
R = Gas constant for air (length*force / mass*temperature)
T_{ref} = Reference temperature (Rankin) **= IMT = Roughly constant for given Speed**

Engine breathing lines





Parameter	Torque Peak	Rated	Units
Horsepower	48	69	hp
BSFC	0.377	0.401	lb/hp-hr
A/F	23.8	24.5	none

Heywood, 1988

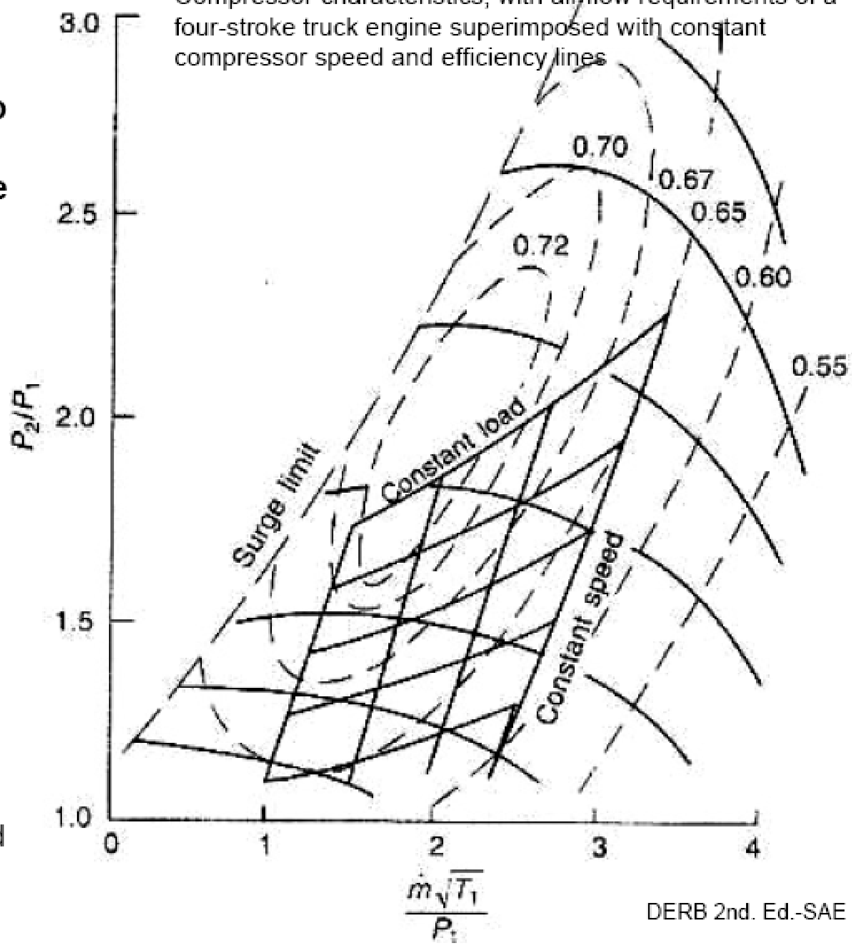
Matching

Centering the Engine Map on the Compressor Map for Optimum Performance

The flow characteristics of rotary turbomachines and reciprocating engines are not ideally suited to operate in tandem.

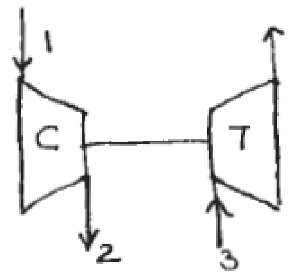
- Automotive engines
  - wide speed, load and flow range
  - positive displacement
  - discontinuous flow
- Turbochargers
  - high mass flow, with high design point efficiency.
  - narrow range
  - continuous flow no defined displacement

Compressor characteristics, with air flow requirements of a four-stroke truck engine superimposed with constant compressor speed and efficiency lines



DERB 2nd. Ed.-SAE

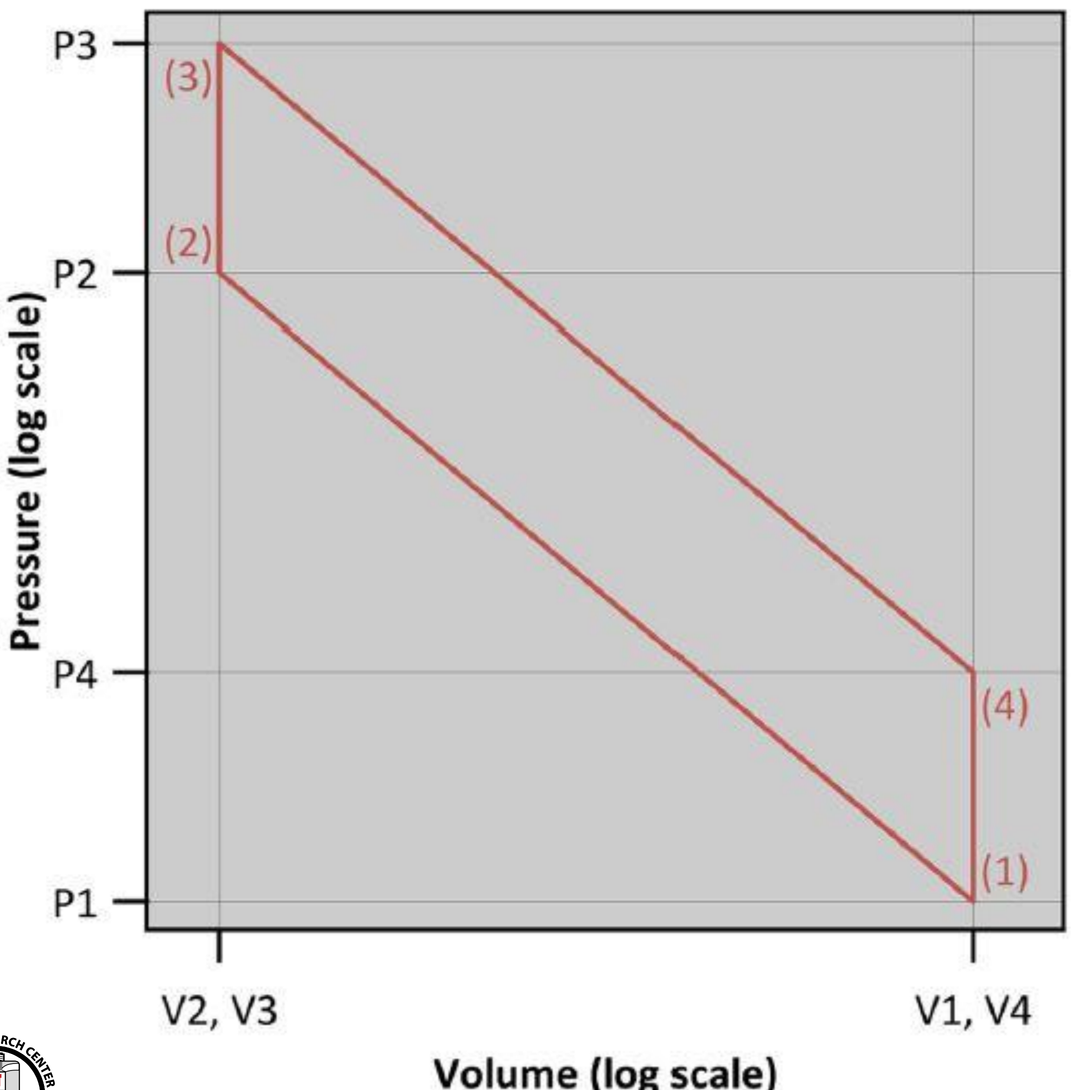
$$\dot{W}_t = \dot{W}_c$$



$$\left(\frac{p_2}{p_1}\right) = \left[1 + \frac{Cp_g \cdot T_3}{Cp_a \cdot T_1}\left(1 + \frac{\dot{m}_{fuel}}{\dot{m}_{air}}\right)\left(\eta_t \cdot \eta_c \cdot \eta_{mech}\right)\left(1 - \left(\frac{p_4}{p_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}}\right)\right]^{\frac{\gamma_a}{\gamma_a - 1}}$$

Heywood, 1988

Ideal engine efficiency – Otto cycle



Maximum possible closed-cycle efficiency ("ideal efficiency")

State (1) to (2) isentropic
(i.e., adiabatic and reversible)
compression from max (V1) to min cylinder volume (V2)
Compression ratio rc = V1/V2.

State (2) to (3) adiabatic
and isochoric (constant volume) combustion,
State (3) to (4) isentropic expansion.

State (4) to (1) exhaust process
- available energy is rejected
- can be converted to mechanical or electrical work:

Ideal engine efficiency – Otto cycle

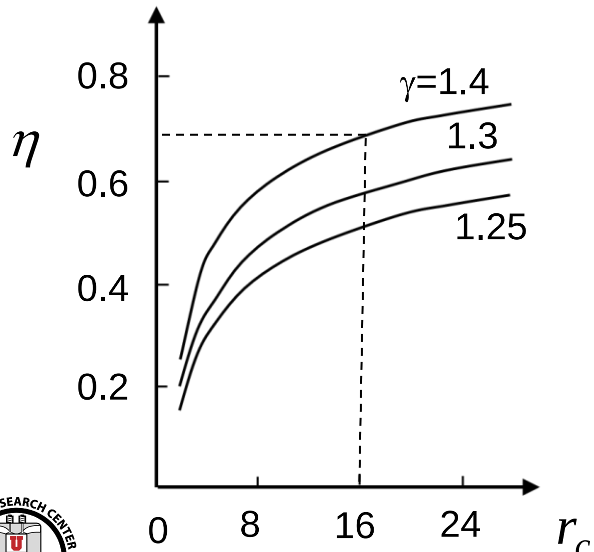
Efficiency = net work / energy supplied

$$\eta = [(T_3 - T_4) - (T_2 - T_1)]/(T_3 - T_2)$$

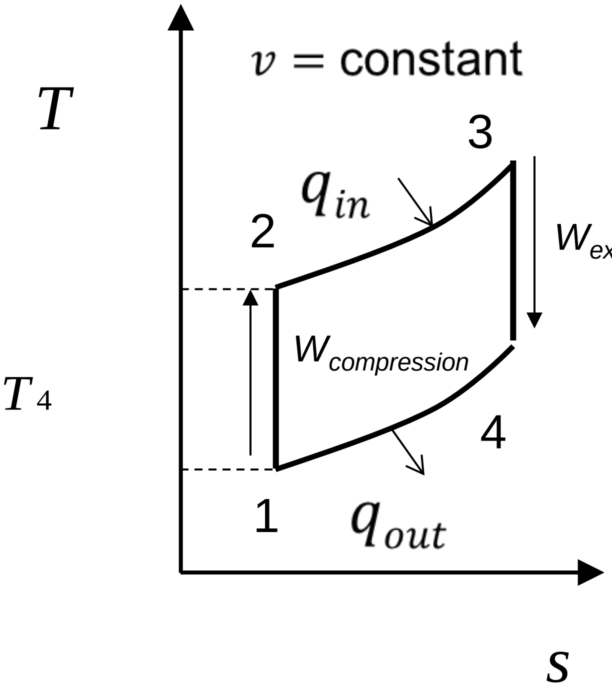
$$= 1 - (T_4 - T_1)/(T_3 - T_2)$$

However,

$$T_2/T_1 = (V_1/V_2)^{\gamma-1} = r_c^{\gamma-1} = (V_4/V_3)^{\gamma-1} = T_3/T_4$$



Otto



$$\boxed{\eta = 1 - 1/r_c^{\gamma-1}}$$

$$\eta_{ideal} = 1 - \frac{1}{r_c^{\gamma-1}}$$

η_{ideal} Function of only two variables, compression ratio (r_c) and ratio of specific heats (γ)

Increasing r_c increases operating volume for compression and expansion
Increasing γ increases pressure rise during combustion and increases work extraction during expansion stroke.

Both effects result in an increase in net system work for a given energy release and thereby increase engine efficiency.

<u>Actual closed-cycle efficiencies to deviate from ideal:</u>

1.) Assumption of isochoric (constant volume) combustion:
Finite duration combustion in realistic engines.

Kinetically controlled combustion has shorter combustion duration than diesel or SI
- duration limited by mechanical constraints, high pressure rise rates with audible engine noise and high mechanical stresses

2.) Assumption of calorically perfect fluid:

Specific heats decrease with increasing gas temperature; species conversion during combustion causes γ to decrease

3.) Adiabatic assumption:

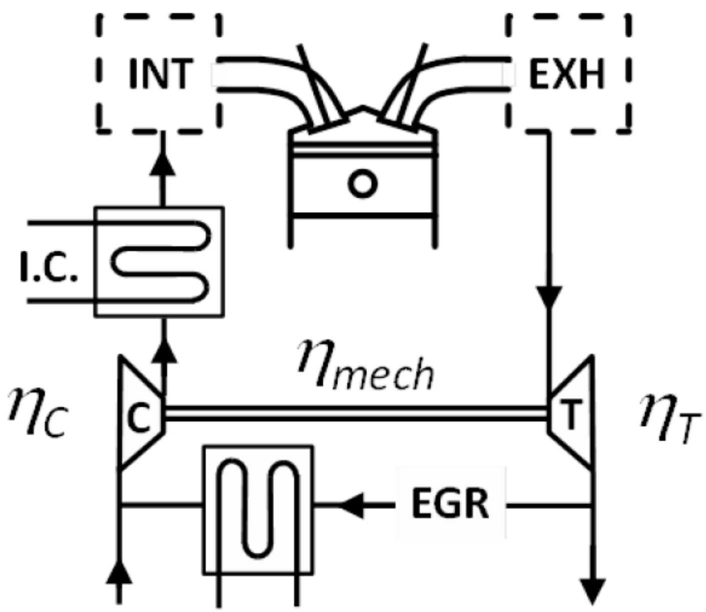
Large temperature gradient near walls results in energy being lost to heat transfer rather than being converted to crank work

Other assumptions:

In engine system models, compressors, supercharger, turbines modeled with constant isentropic efficiency instead of using performance map.

- typically, compressors, superchargers, and fixed geometry turbines have isentropic efficiencies of 0.7. VGT has isentropic efficiency of 0.65.

Charge coolers - intercooler, aftercooler, and EGR cooler modeled with zero pressure drop, a fixed effectiveness of 0.9, constant coolant temperature of 350 K.



Zero-dimensional closed-cycle analysis:

Combustion represented as energy addition to closed system

Fuel injection mass addition from user-specified start of injection crank angle (θ_{SOI}) and injection duration ($\Delta\theta_{\mathrm{inj}}$).

Pressure and mass integrated over the closed portion of cycle with specified initial conditions at IVC of pressure (p_0), temperature (T_0), and composition ($x_{n,0}$ for all species considered - N_2, O_2, Ar, CO_2, and H_2O) and initial trapped mass (m_0), including trapped residual mass

Post-combustion composition determined assuming complete combustion of delivered fuel mass.

Minor species resulting from dissociation during combustion not considered

Herold, 2011

First law energy balance: $de = dq - Pdv$

$$\left.\frac{dp}{d\theta}\right|_i = \left(\left.\frac{dQ_C}{d\theta}\right|_i - \left.\frac{dQ_{HT}}{d\theta}\right|_i - \frac{\gamma_i}{\gamma_i - 1} p_i \left.\frac{dV}{d\theta}\right|_i \right) \frac{\gamma_i - 1}{V_i}$$

Combustion:

$$\left.\frac{dQ_C}{d\theta}\right|_i = \frac{x_{b,i+1} - x_{b,i-1}}{\theta_{i+1} - \theta_{i-1}} \left(m_f LHV_f \right)$$

Wall heat transfer:

$$\left.\frac{dQ_{HT}}{d\theta}\right|_i = h_{c,i} \left[A_{IP,i}\left(T_i - T_{m,IP}\right) + A_{EP,i}\left(T_i - T_{m,EP}\right) + A_{l,i}\left(T_i - T_{m,l}\right) \right]$$

Combustion model - Wiebe function

$$x_{b,i} = 1 - \exp\left\{ -\left[\left(2.302^{\frac{1}{m_c+1}} - 0.105^{\frac{1}{m_c+1}} \right) \left(\frac{\theta_i - \theta_{SOC}}{\Delta\theta_{10-90}} \right) \right]^{m_c+1} \right\}$$

Heat transfer model - Woschni

$$h_{c,i} = 5 b^{m_{ht}-1} p_i^{m_{ht}} w_i^{m_{ht}} T_i^{0.75 - 1.62 m_{ht}}$$

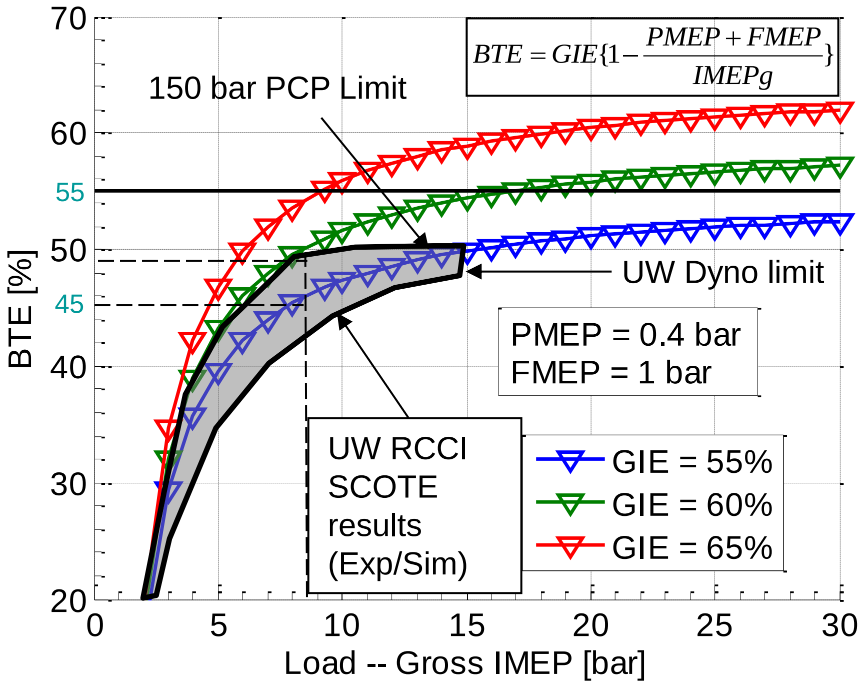
$$w_i = 2.28 v_p + (3.25 \times 10^{-3}) \frac{V_d T_0}{p_0 V_{tr}} \left(p_i - p_{mot,i} \right)$$

Chen-Flynn, 1965

Engine brake thermal efficiency BTE

BTE*LHV=IMEPg-PMEP-FMEP

DOE goal BTE=55%



Friction model

Chen-Flynn model (SAE 650733).

$$FMEP = C + (PF*P_{max}) + (MPSF*Speed_{mp}) + (MPSSF*Speed_{mp}^2)$$

where: C = constant part of FMEP (0.25 bar)

PF = Peak Cylinder Pressure Factor (0.005)

P_{max} = Maximum Cylinder Pressure

MPSF = Mean Piston Speed Factor (0.1)

MPSSF = Mean Piston Speed Squared Factor (0)

$Speed_{mp}$ = Mean Piston Speed

Lavoie, 2012

1-D modeling for engine performance analysis

Table 1. Engine Specifications

Bore/Stroke	90 mm/100 mm
CR	12
Intake valves (2)	32.4 mm Diam/ 10.7 mm Lift
IVO (at 0 lift) IVC (at 0 lift)	-12°ATC gas exch. 60 to 224°ATC gas exch.
Exhaust valves (2)	26.1 mm Diam/ 10.7 mm Lift
EVO (at 0 lift) EVC (at 0 lift)	135°ATC firing 371°ATC firing

Mid load

Table 3. Operating conditions and parameters

RPM	2400 (U_P = 8 m/s)
Φ	0.2 – 1.2
EGR	0 – 80%
P_{EX}	1-3 (bar)
T_{IN}	333 K (60°C)
T_{ATM}	298 K (25°C)
T_{WALL} *(K)*	460 (head), 510 (pist), 390 (cyl)
T/C Eff (η_{OTC})	40, 50, 60%
Burn 10-90	25° CAD
CA50	10 ° ATC (~max eff.)

Lavoie, 2012

Table 2. Submodel specifications	
Heat Transfer	Standard Woshni [27, 30]
Heat Release	Standard Wiebe [31]
Friction	Chen-Flynn [27, 32]
NOx model	2-zone Zeldovich [27, 31]

Woshni, 1967

Turbocharger equation

$$\left[1-\left(\frac{P_{ATM}}{P_{EX}}\right)^{\frac{\gamma_C-1}{\gamma_C}}\right]=\frac{\dot{m}_C C_{PC} T_{ATM}}{\dot{m}_T C_{PT} T_{EX}}\frac{1}{\eta_{OTC}}\left[\left(\frac{P_{IN}}{P_{ATM}}\right)^{\frac{\gamma_T-1}{\gamma_T}}-1\right]$$

$$\eta_{OTC}=\eta_T\eta_{MECH}\eta_C$$

Burn duration

$$x_b=1-exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{w+1}\right]$$

Heat transfer

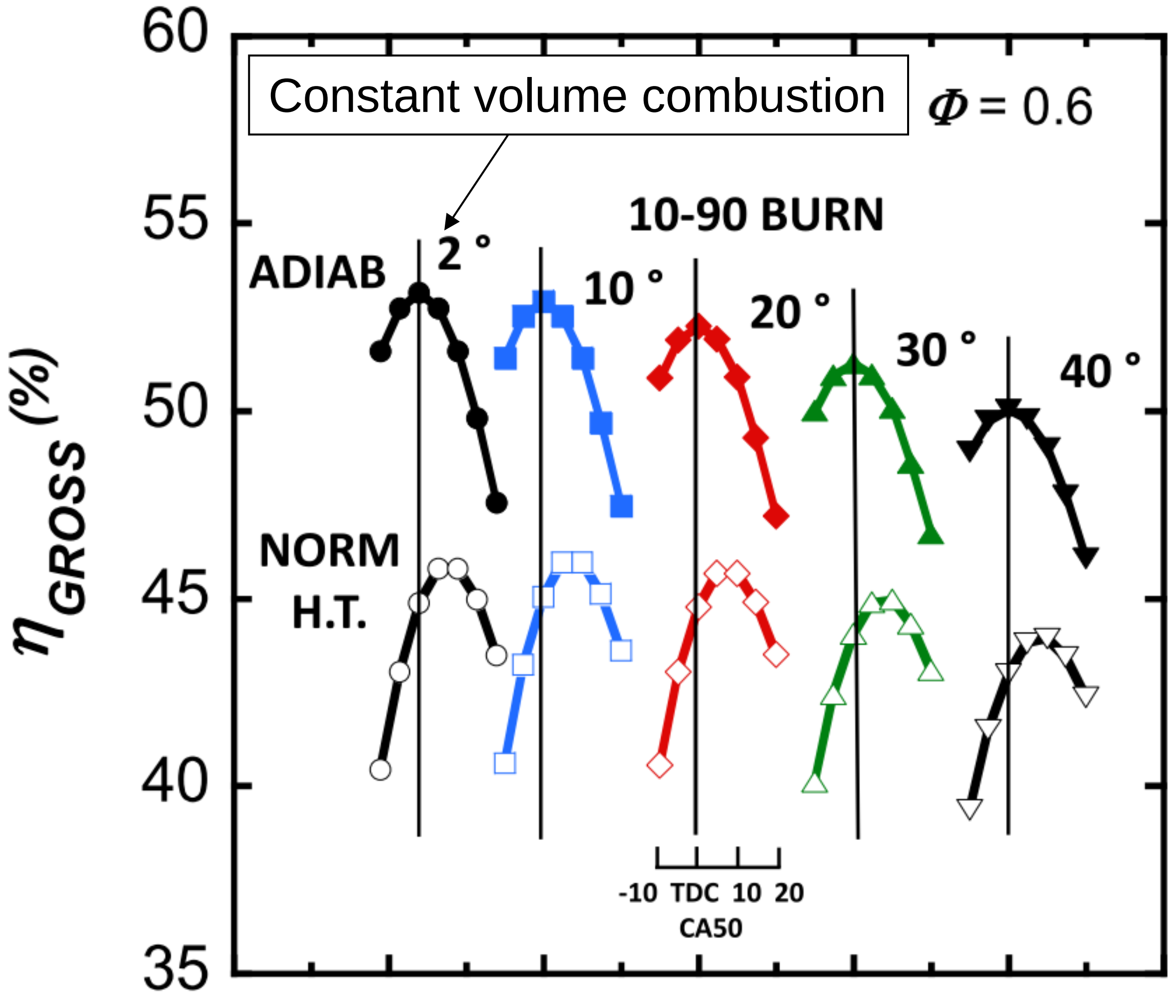
$$Nu\equiv\frac{hB}{k}\propto Re^m$$

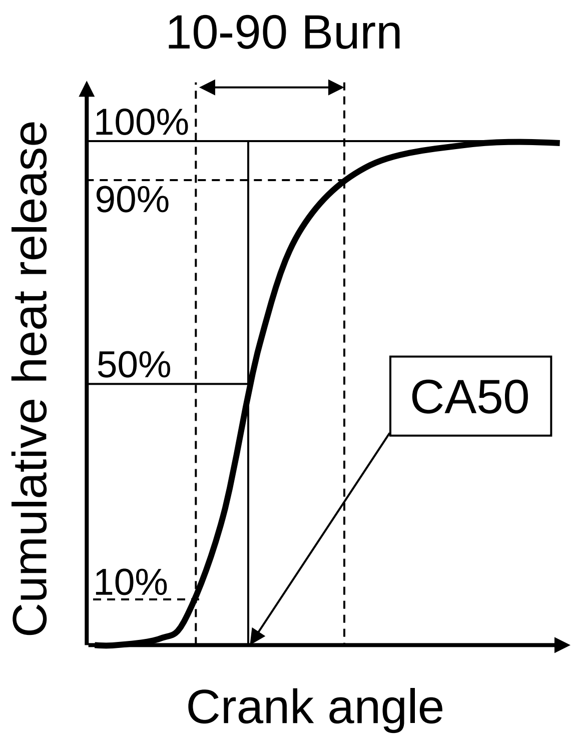
m~0.8, Re increases with Bore and ρ (boost)

Friction

$$FMEP\ (bar)=\ 0.4+0.005P_{MAX}+0.09U_P+\ 0.0009U_P^2$$

Effect of combustion phasing on efficiency

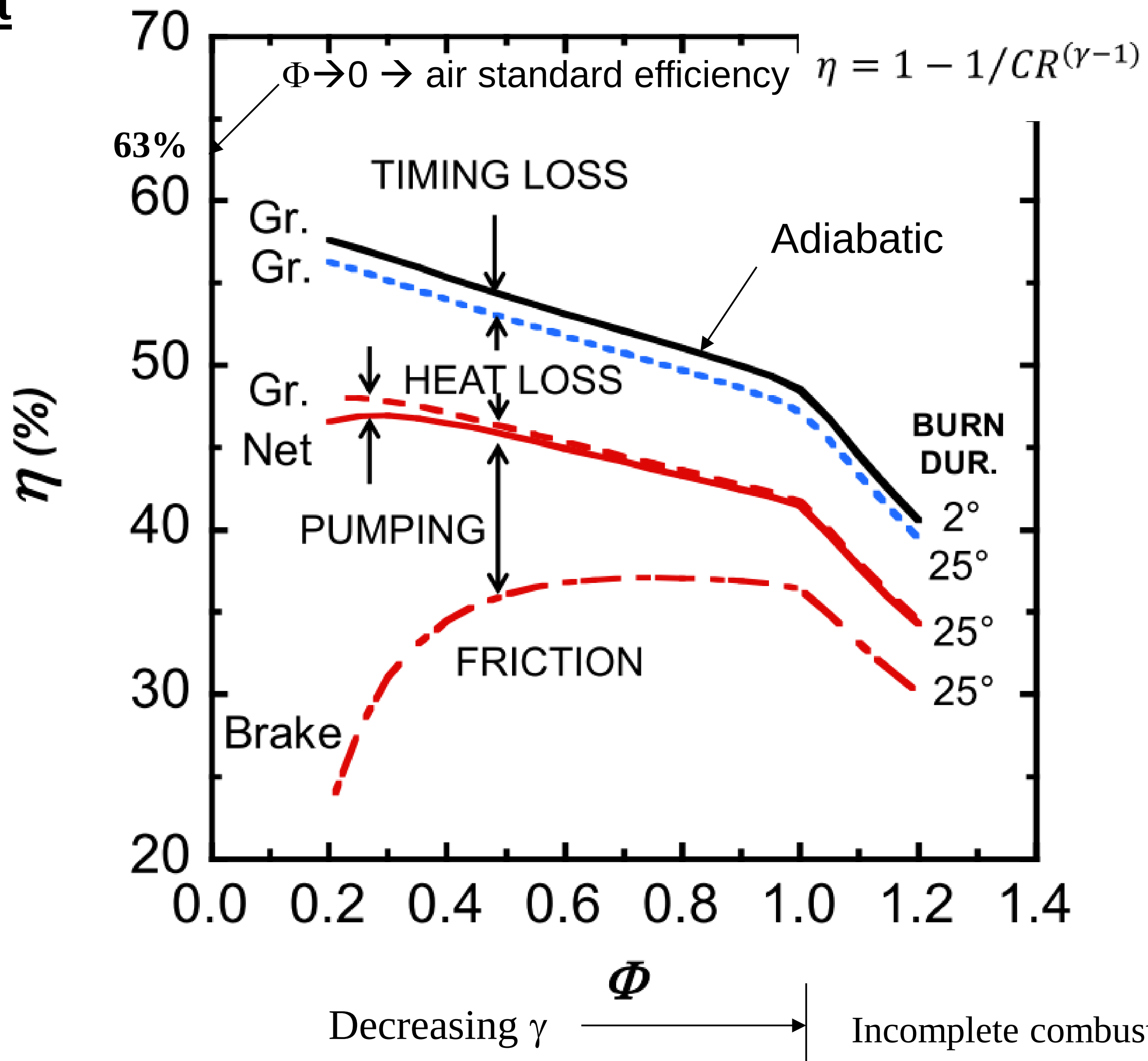




Without HT: Best efficiency CA50~TDC

With HT: best efficiency with CA50~10 deg – tradeoff between heat loss/late expansion

Energy budget

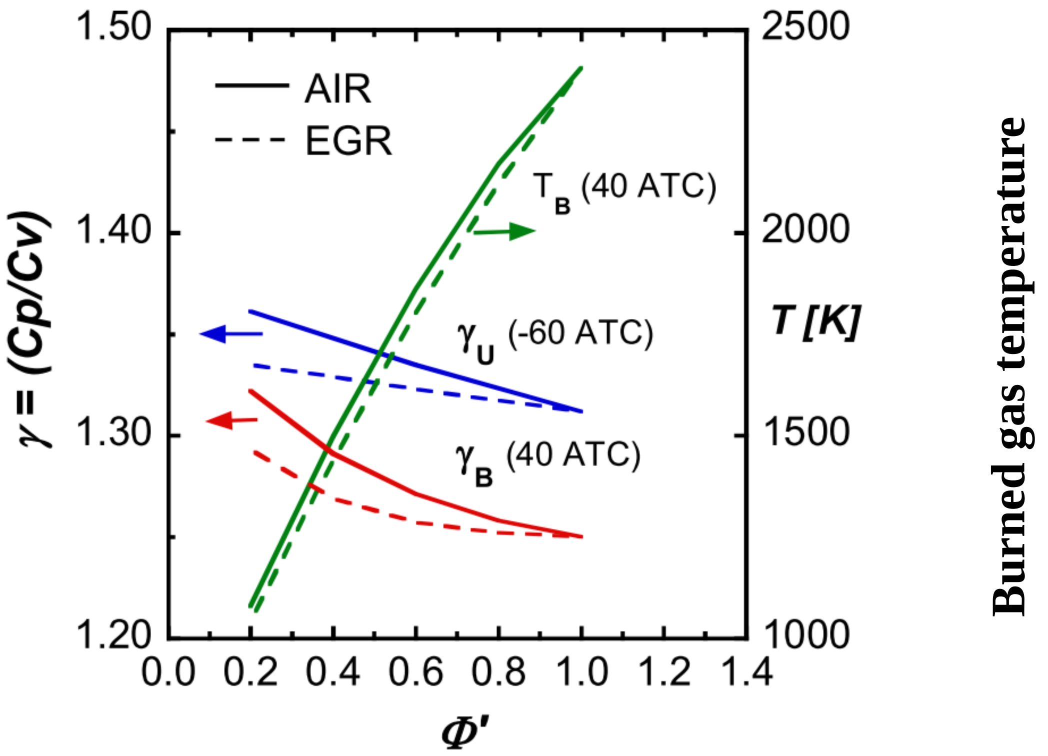


Effect of dilution

Fuel-to-charge equivalence ratio, ϕ'

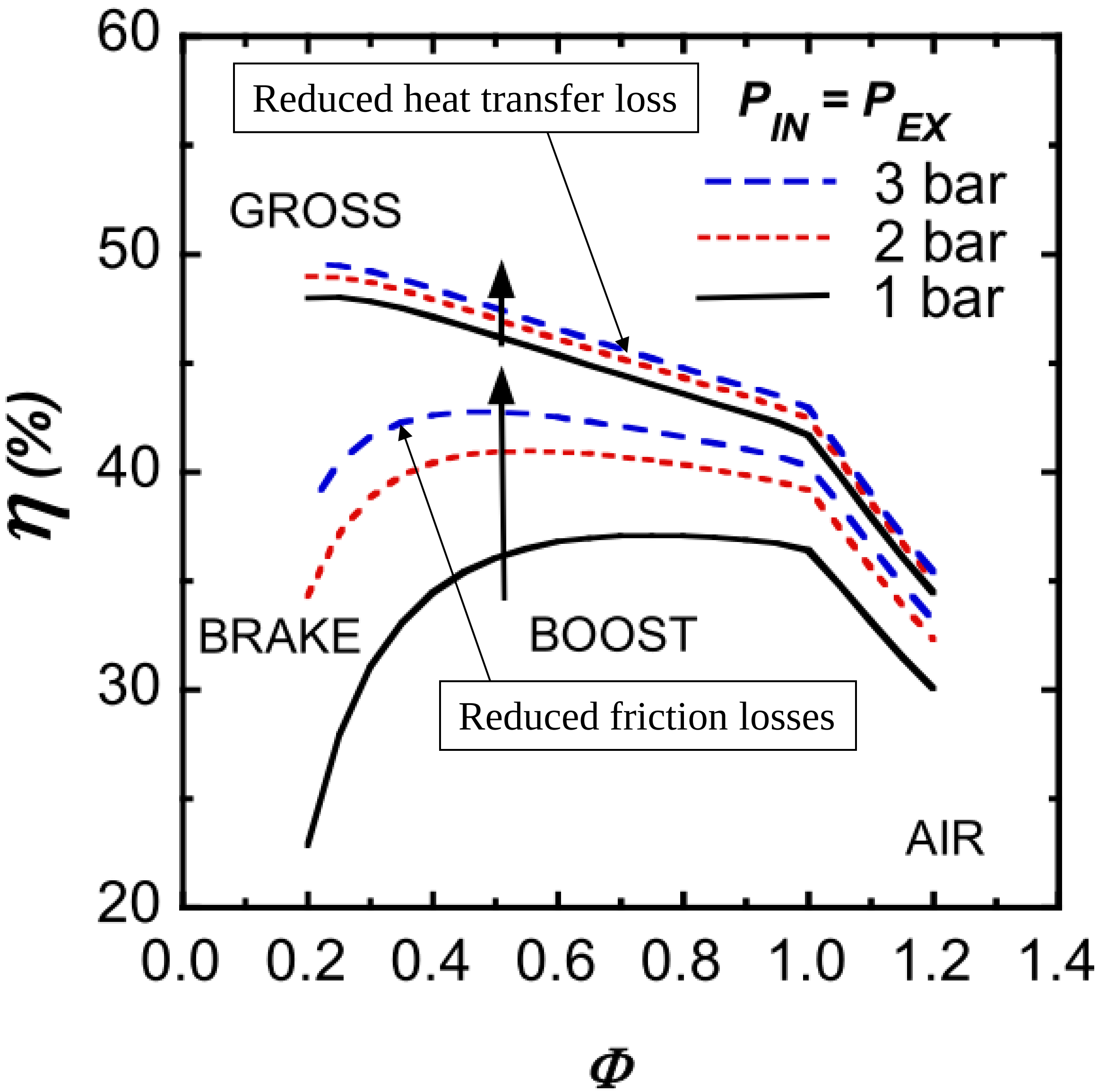
$$\Phi' \equiv \frac{F/(A+R)}{(F/A)_{ST}} = \frac{\Phi(1-RGF)}{[1+\Phi\cdot RGF\cdot(F/A)_{ST}]} \cong \Phi(1-RGF)$$

where F, A, and R denote mass of fuel, air, and residual gas, RGF is the total residual gas fraction

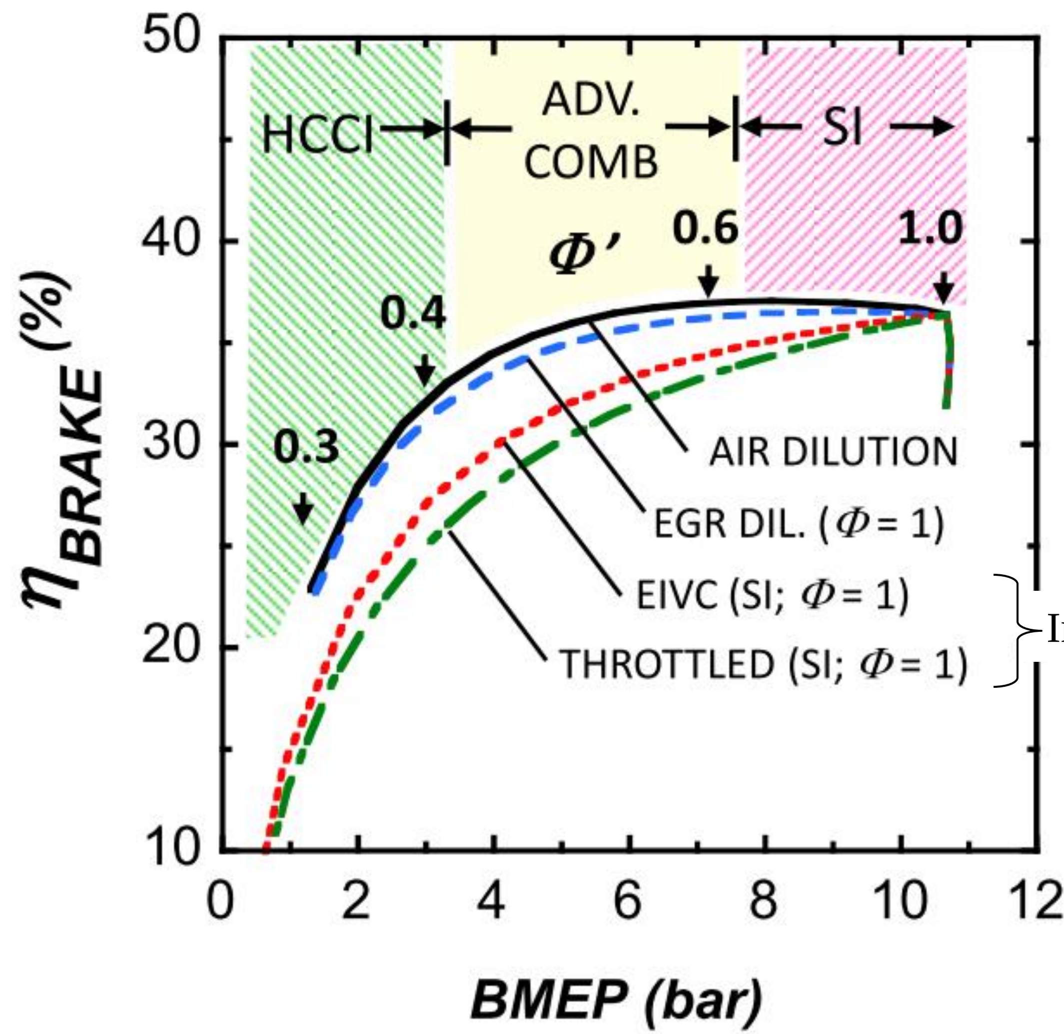


ϕ ranges from 0.2 to 1 with air, EGR ranges from 0 to 80% with ϕ=1

Effect of boost on efficiency



Potential brake efficiencies of naturally aspirated engines



Increased pumping losses

Summary

Turbocharging can increase engine efficiency by using available energy in exhaust and by reducing pumping work

Air standard "ideal cycle" analysis provides a bound on engine efficiency estimates.

0-D engine system models provide estimates of engine system efficiencies, if combustion details (e.g., timing and duration) and heat transfer losses are assumed

The goal of multi-dimensional models (to be discussed next) is to predict the combustion details