Internal Combustion Engines
I: Fundamentals and Performance Metrics

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Summer School on Combustion
Course Length: 9 hrs

(Mon.- Wed., June 25-27)

Short course outline:

Internal Combustion (IC) 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)

- Hour 1: IC Engine Review, Thermodynamics and 0-D modeling
- Hour 2: 1-D modeling, Charge Preparation
- Hour 3: Engine Performance Metrics, 3-D flow modeling

Day 2 (Computer modeling/engine processes)

- Hour 4: Engine combustion physics and chemistry
- Hour 5: Premixed Charge Spark-ignited engines
- Hour 6: Spray modeling

Day 3 (Engine Applications and Optimization)

- Hour 7: Heat transfer and Spray Combustion Research
- Hour 8: Diesel Combustion modeling
- Hour 9: Optimization and Low Temperature Combustion

Anderson, 1990

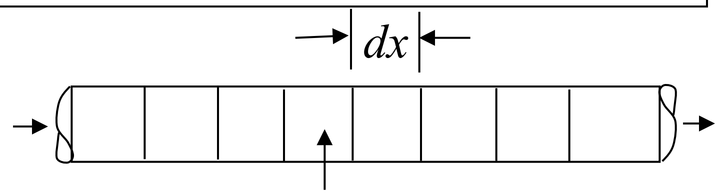
1-D compressible flow

Reynolds Transport Equation

$$\left.\frac{dMg}{dt}\right)_{system} = \frac{d}{dt}\int_{system} \rho g d\forall = \frac{d}{dt}\int_{cv} \rho g d\forall + \int_{cs} \rho g \mathbf{V}_{rel}\cdot \mathbf{n}\, dA$$

Mass conservation:

$$g = 1 \quad dMg/dt)_{System} = 0$$



$$d\forall = Adx$$

cv fixed

Divergence theorem

$$0 = \int_{cv} \left\{ \frac{\partial(\rho A)}{\partial t} + \nabla \cdot (\rho A \mathbf{V}) \right\} dx$$

1. $$\frac{\partial(\rho A)}{\partial t} + \frac{\partial(\rho A V)}{\partial x} = 0$$

Momentum conservation:

2. $$\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial x} + 2fV^2/D = 0$$

Energy conservation:

3. $$\frac{\partial e}{\partial t} + V\frac{\partial e}{\partial x} = \dot{q} + 2fV^3/D - \frac{P}{\rho A}\frac{\partial(VA)}{\partial x}$$

Supplementary:

4. $P = \rho RT$

5. $e = c_v T$

State

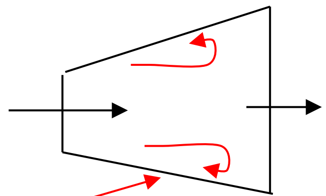
$$f = \tau_w / \rho V^2 / 2$$

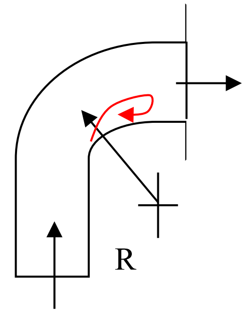
$$\dot{Q} = \dot{q}\rho A dx$$

5 unknowns U: ρ, V, e, P, and T

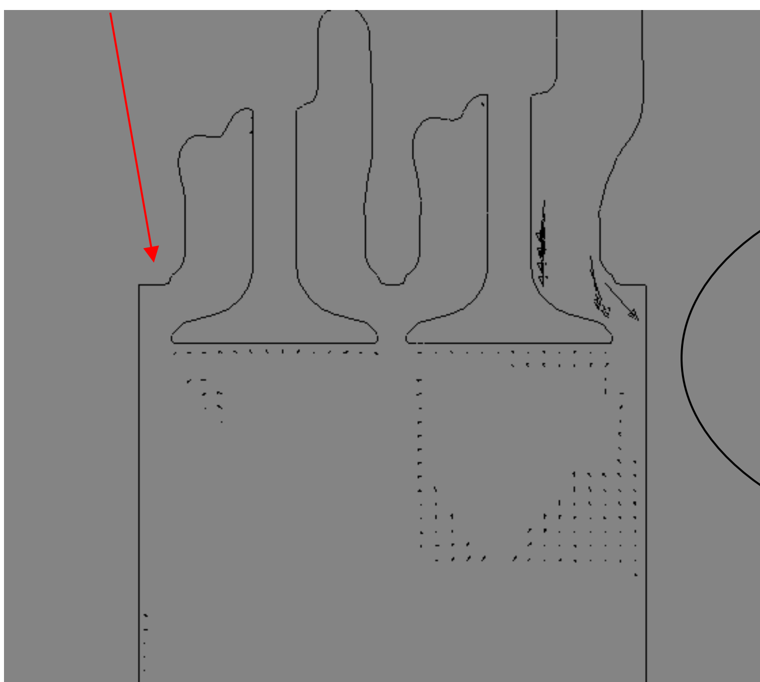
5 equations for variation of flow variables in space and time

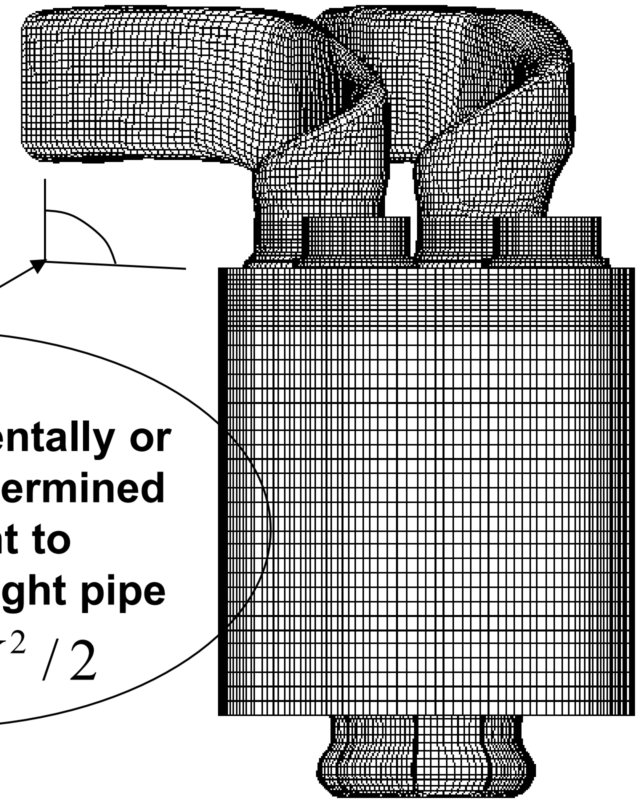
In 1-D models friction factors are used to account for losses at area change or bends by applying a friction factor to an "equivalent" length of straight pipe





Flow losses





Apply experimentally or numerically determined Loss Coefficient to equivalent straight pipe

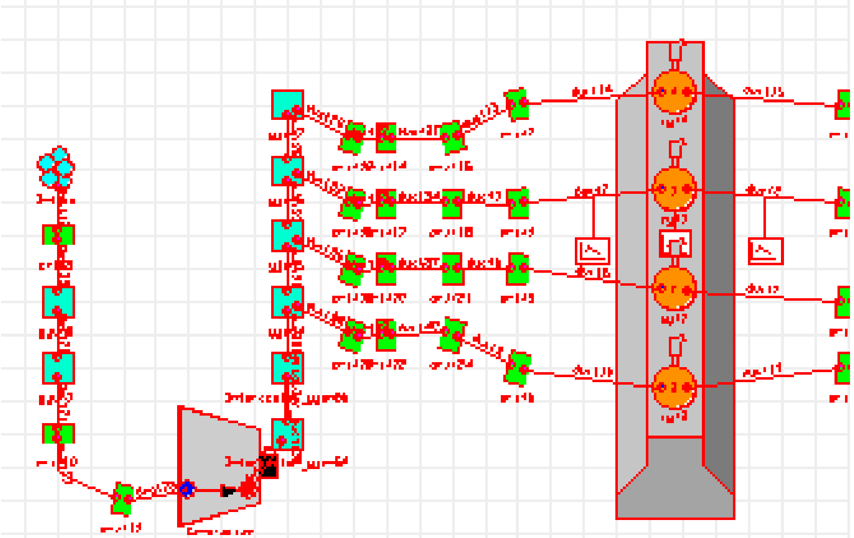
$$\Delta P = C_P \rho V^2 / 2$$

1-D Modeling Codes

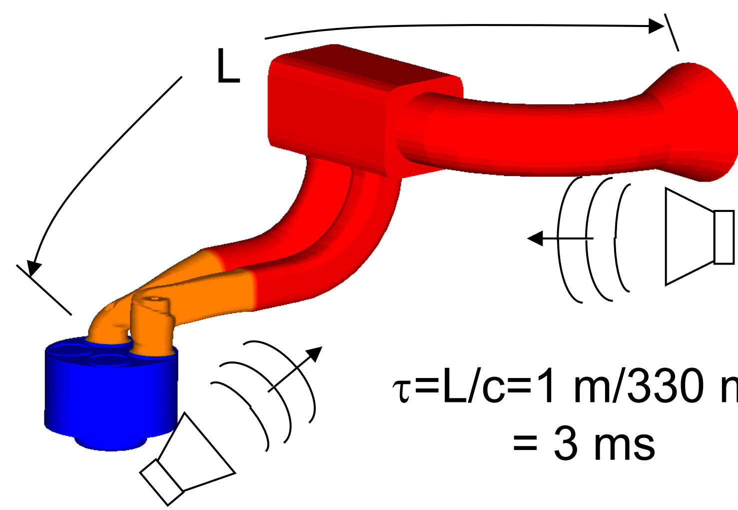
1-D codes (e.g., GT-Power, AVL-Boost, Ricardo WAVE) predict wave action in manifolds
At high engine speed valve overlap can improve engine breathing
→ inertia of flowing gases can cause inflow even during compression stroke.

Variable Valve Actuation (VVA) technologies, control valve timing to change effective compression ratio (early or late intake valve closure), or exhaust gas re-induction (re-breathing) to control in-cylinder temperatures.

Residual gas left from the previous cycle affects engine combustion processes through its influence on charge mass, temperature and dilution.



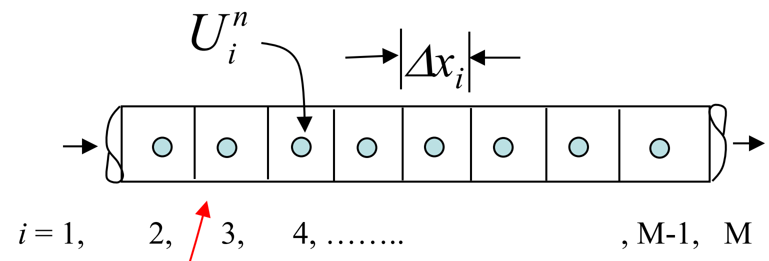
AVL Boost, Ricardo WAVE, GT-Power



$\tau = L/c = 1\text{ m}/330\text{ m/s} = 3\text{ ms}$

1 ca deg = 0.1 ms @ 1800 rev/min

Numerical solution



To integrate the partial differential equations:
Discretize domain with step size, Δx
Time marches in increments of Δt from initial state U_i^0 : $\rho_i^n, V_i^n, e_i^n, P_i^n$, and T_i^n

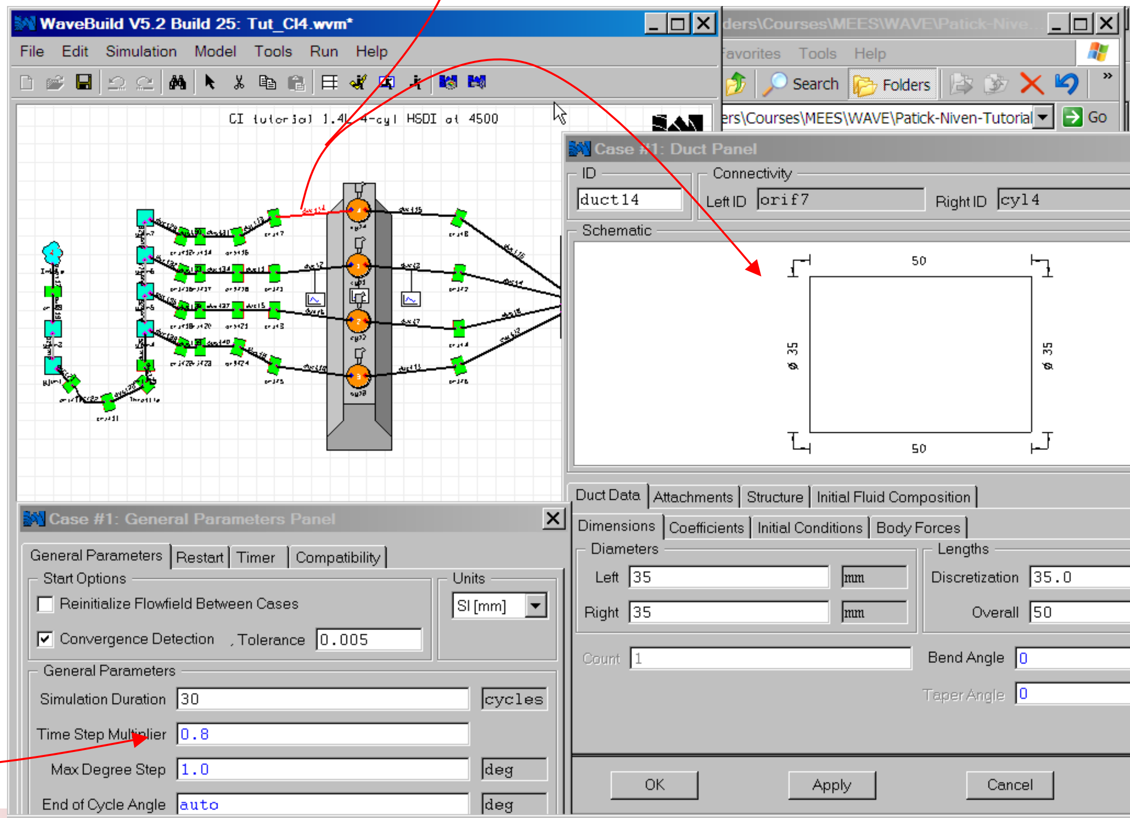
$$t = n\Delta t \quad n = 0, 1, 2, 3, \ldots.$$

$$\frac{\partial U(x,t)}{\partial x} = \frac{\Delta U(x_i, n \cdot dt)}{\Delta x_i} = \frac{U_{i+1}^n - U_i^n}{\Delta x_i}$$

$$\frac{\partial U(x,t)}{\partial t} = \frac{\Delta U(x_i, n \cdot dt)}{\Delta t} = \frac{U_i^{n+1} - U_i^n}{\Delta t}$$

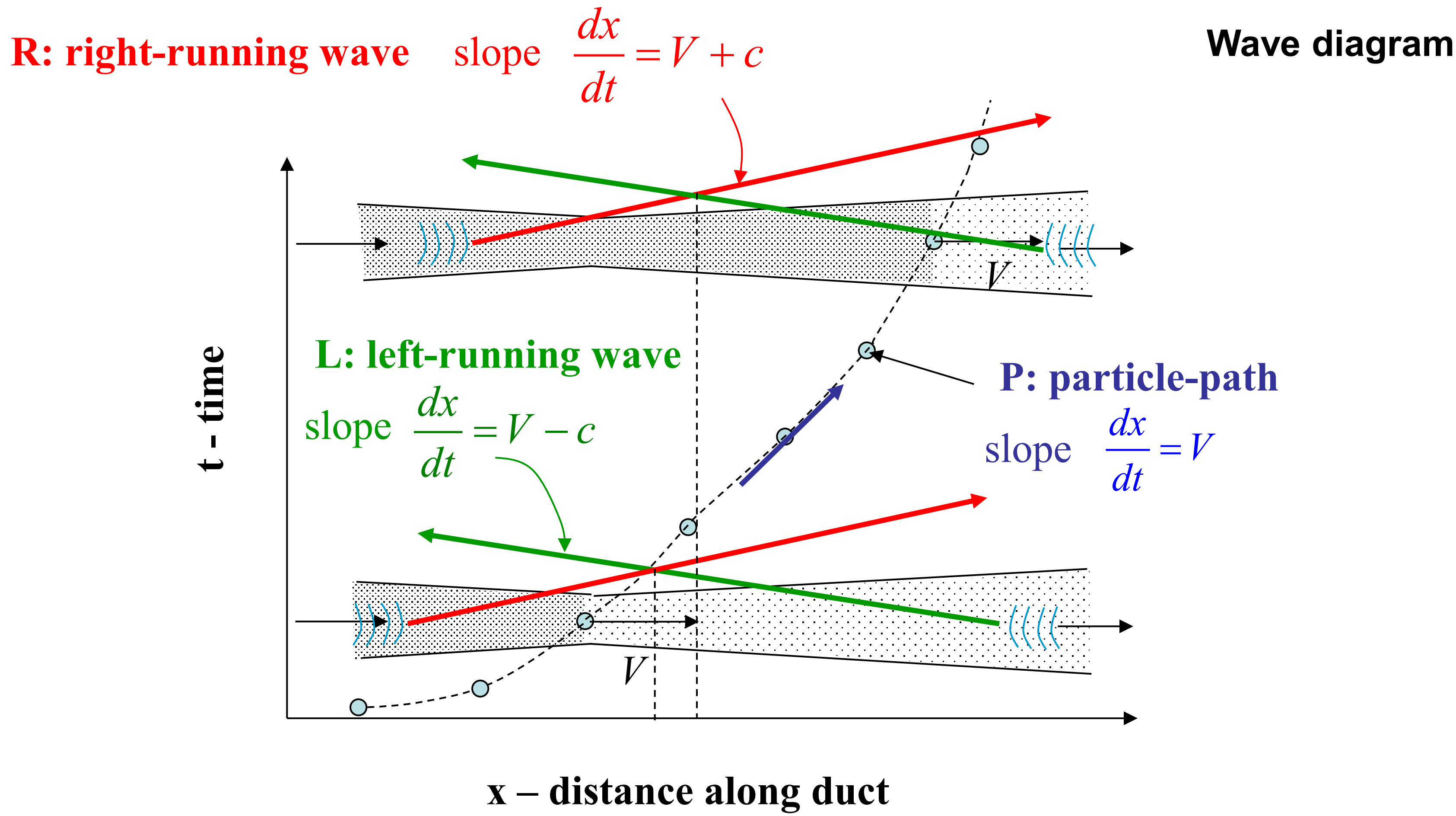
Considerations of stability require the Courant-Friedrichs-Levy (CFL) condition

$$\Delta t \le \min(\Delta x_i / (|V_i^n| + c_i^n))$$



Anderson, 1990

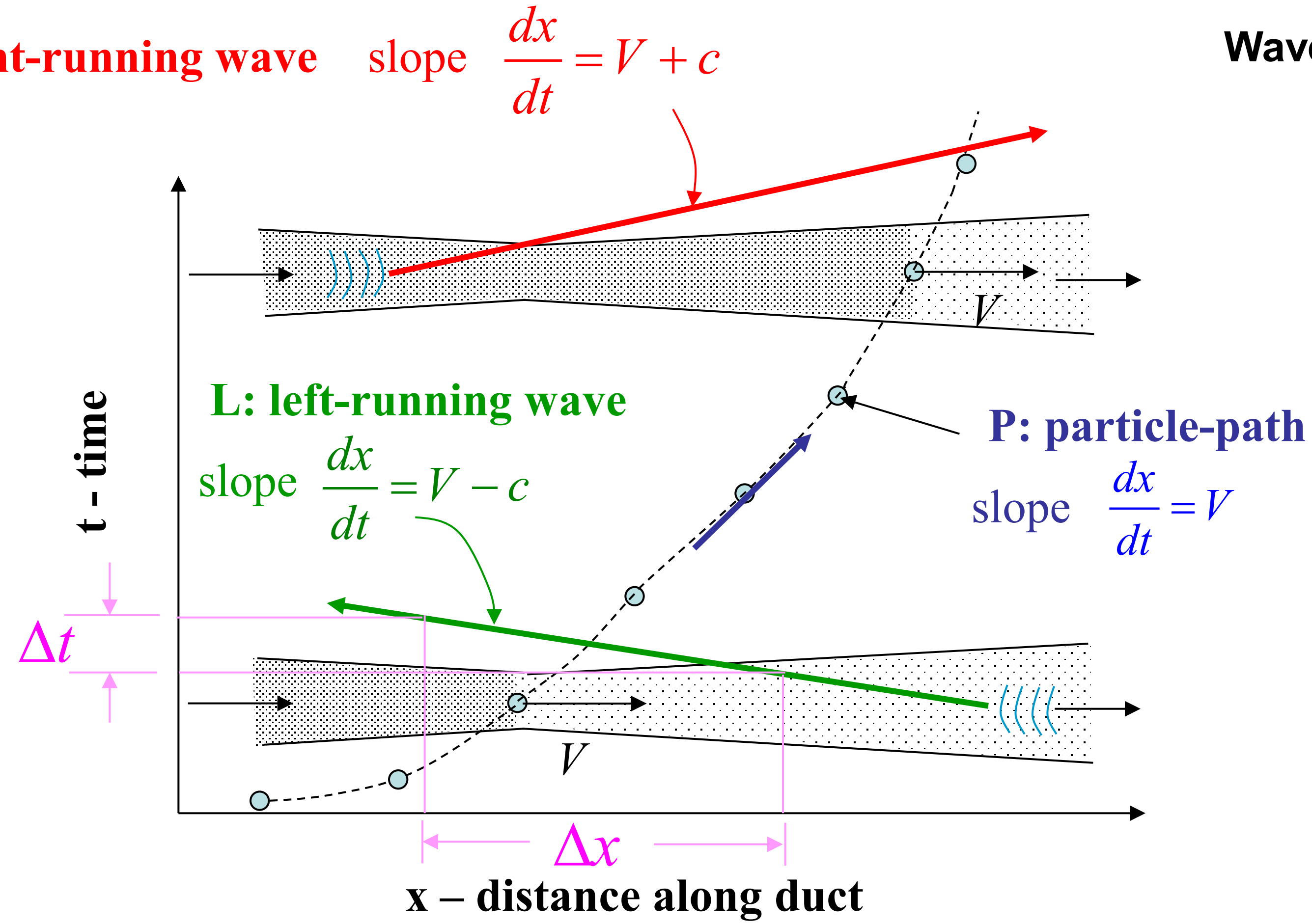
Analytical solutions – Method of Characteristics



All points continuously receive information about both upstream and downstream flow conditions from both left and right-running waves. These waves originate from all points in the flow.

Anderson, 1990

Analytical solutions – Method of Characteristics



R:, L:, P:, are called Characteristic Lines in the flow

$$\Delta t \le \min(\Delta x_i / (|V_i^n| + c_i^n))$$

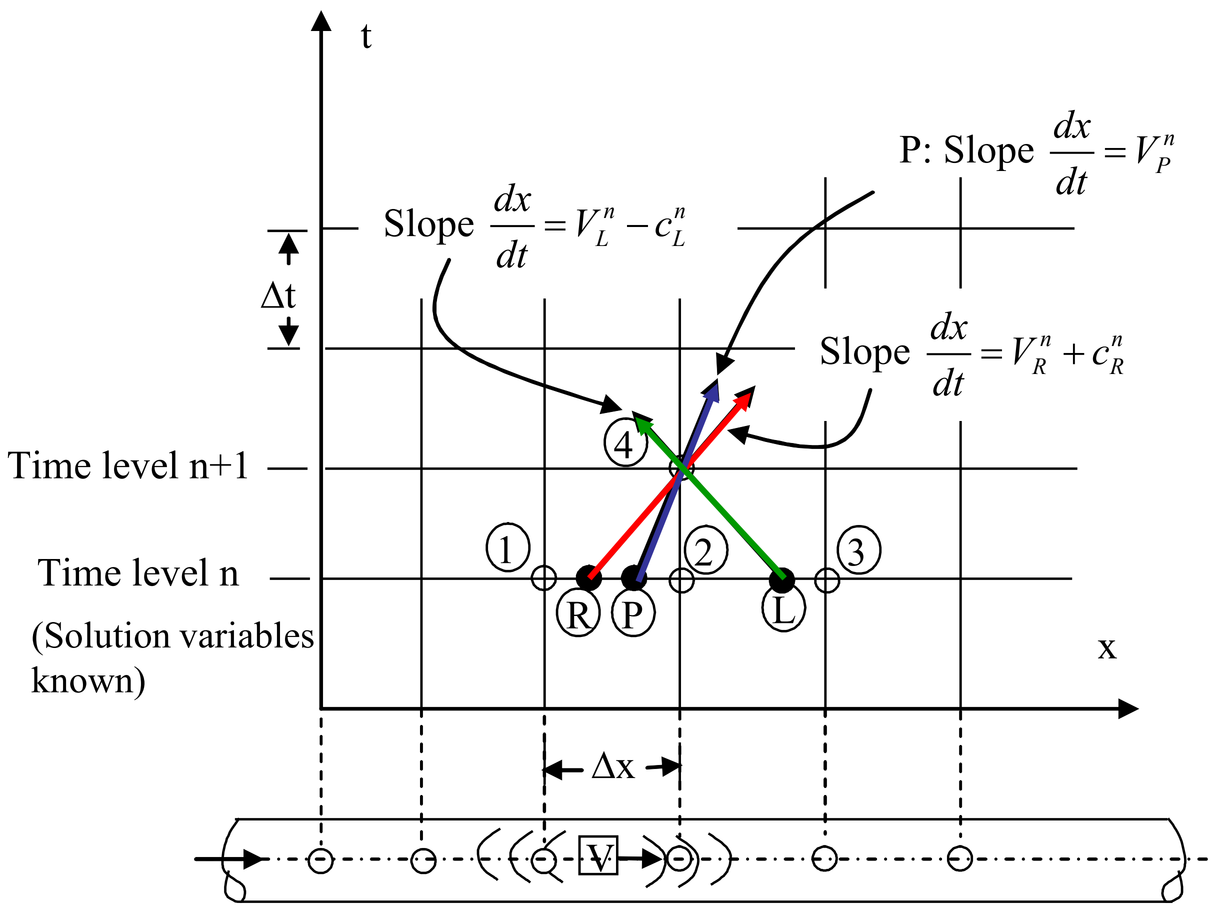
Moody, 1989

Along R: $dP + \rho c dV = F dt$

Along L: $dP - \rho c dV = G dt$

Along P: $d\rho - dp/c^2 = H dt$

$$F, G, H = Functions\ of\ \left(\dot{q}\ f, \ln A/dx\right)$$



The discrete versions are:

$$(P_4 - P_R) + (\rho c)_R (V_4 - V_R) = F_R \Delta t$$

$$(P_4 - P_L) - (\rho c)_L (V_4 - V_L) = G_L \Delta t$$

$$(\rho_4 - \rho_P) - \left(\frac{1}{c^2}\right)_P (P_4 - P_P) = H_P \Delta t$$

3 equations to solve for

ρ_4, V_4 and P_4

Note: from Gibbs' equation

$$dS = \frac{c_P}{\rho}\left(\frac{dP}{c^2} - d\rho\right) = \frac{c_P}{\rho} H dt$$

Thompson, 1972

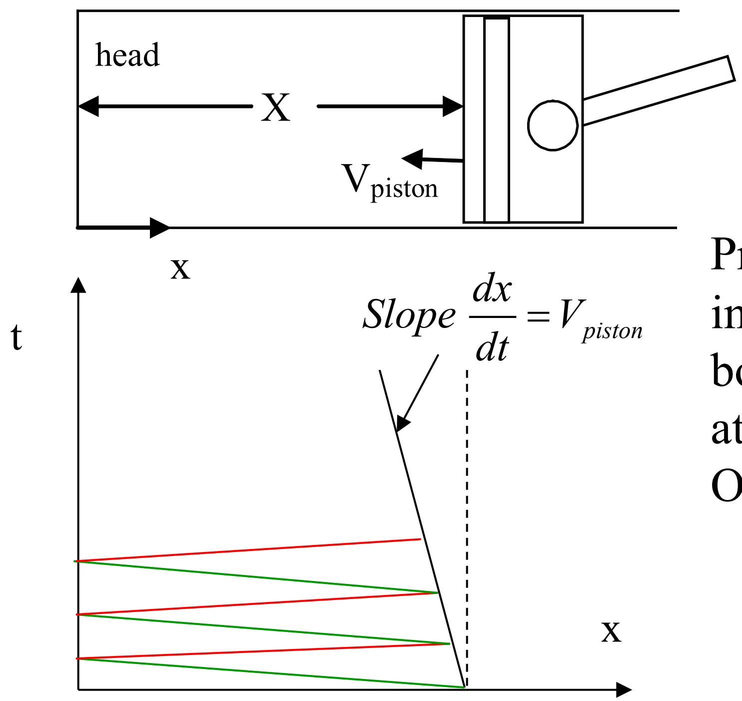
Lagrange ballistics

Flow velocities in IC engine cylinders are usually << than the speed of sound. Lagrange ballistics shows that cylinder pressure and density is the same at all points within the combustion chamber.

L: $P_4 = P_L + (\rho c)_L (0 - V_L)$

R: $P_4 = P_R - (\rho c)_R (V_{piston} - V_R)$

P: $\rho_4 = \rho_P + (P_4 - P_P) / c^2)_P$



Pressure increases by *dP* each wave reflection (*dV<0*) in order to alternately ensure that the flow meets the boundary conditions: *V=0* at head, and $V=V_{piston}$ at piston.

Order of magnitude analysis of **L:, R:,** and **P:** gives

$$dP \sim \rho c dV \quad \text{and} \quad \boxed{\frac{d\rho}{\rho} \sim \frac{dV}{c}}$$

For *dV<<c* relative density change is small– density and pressure changes only in time

Anderson, 1990

Steady Compressible flow – A review

Gibbs ➔ $T\cancel{ds} = dh - dp/\rho$

Energy ➔ $dh = -VdV$

Euler ➔ $dP = -\rho V dV$

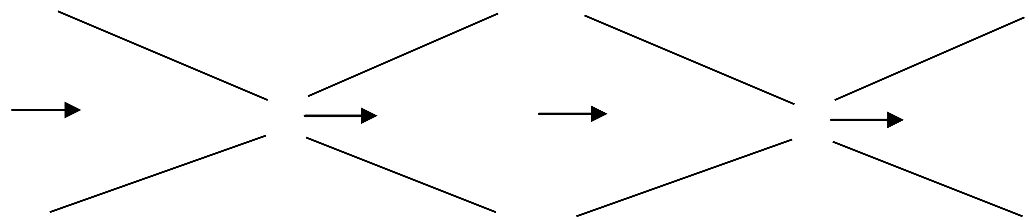
$\rho A V = Const$ ➔ $\dfrac{d\rho}{\rho} + \dfrac{dA}{A} + \dfrac{dV}{V} = 0$

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

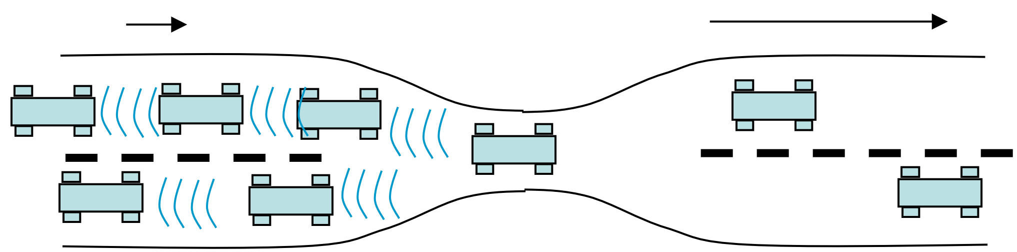
$$\boxed{\frac{dA}{A} = \frac{(1-M^2)}{\rho V^2}dP}$$

Area-velocity relations

for $M<1$ — for $M>1$



	Subsonic nozzle	Subsonic diffuser	Supersonic diffuser	Supersonic nozzle
	$dA<0$	$dA>0$	$dA<0$	$dA>0$
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



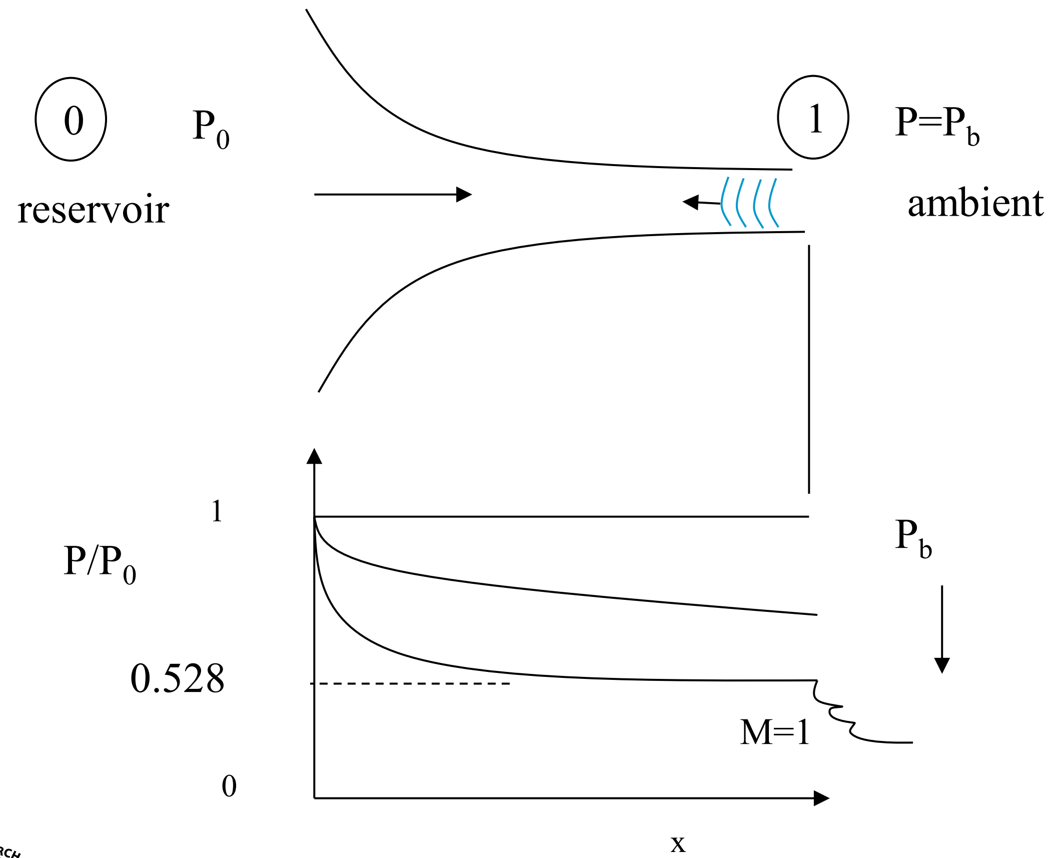
Traffic flow behaves like a supersonic flow!

Anderson, 1990

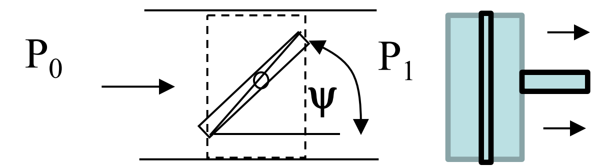
Isentropic nozzle flows

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

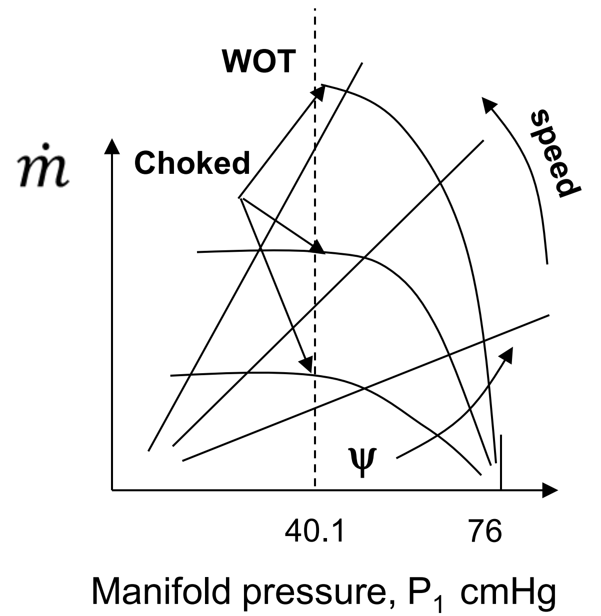
$$\frac{P_0}{P_1} = \left(1 + \frac{\gamma - 1}{2} M_1^{\ 2}\right)^{\frac{\gamma}{\gamma - 1}}$$



Ex. Flow past throttle plate



Choked flow for P_2 < 53.5 kPa = 40.1cmHg



Anderson, 1990

Model passages as compressible flow in converging-diverging nozzles

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

$$= P_0 \sqrt{\frac{\gamma}{RT_0}} AM(P/P_0)/(T/T_0)^{-1/2}$$

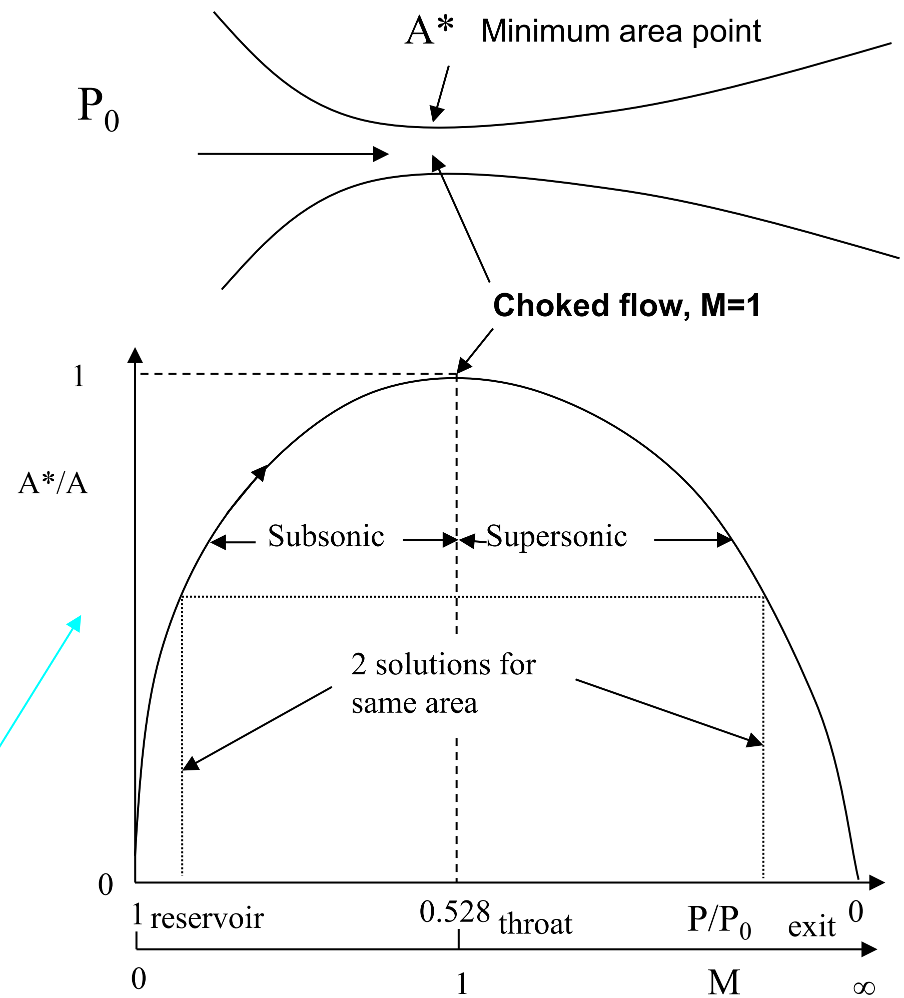
With *M=1*: Fliegner's formula

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

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

Application to turbomachinery

Fliegner's Formula:

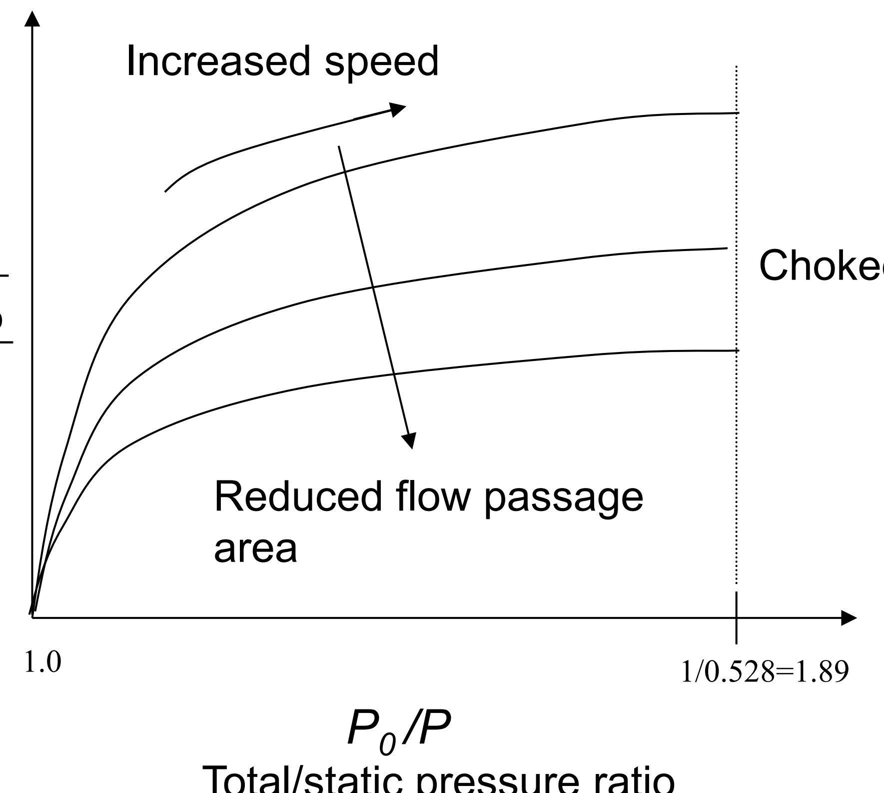
$$\dot{m}_{M=1} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \sqrt{\frac{\gamma}{RT_0}} P_0 A^*$$

Variable Geometry Compressor/ turbine performance map

"Corrected mass flow rate"

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

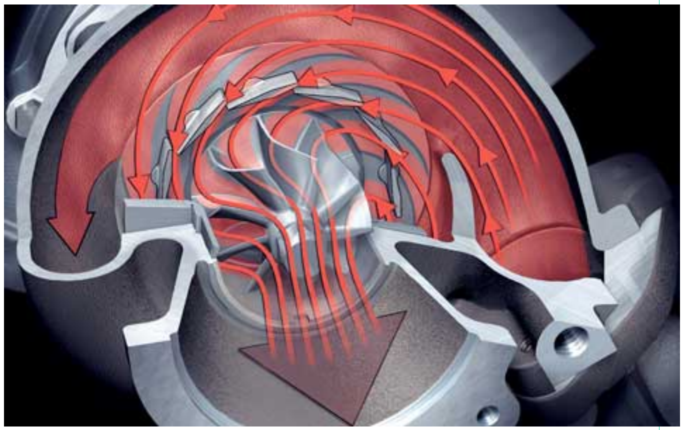
A measure of effective flow area

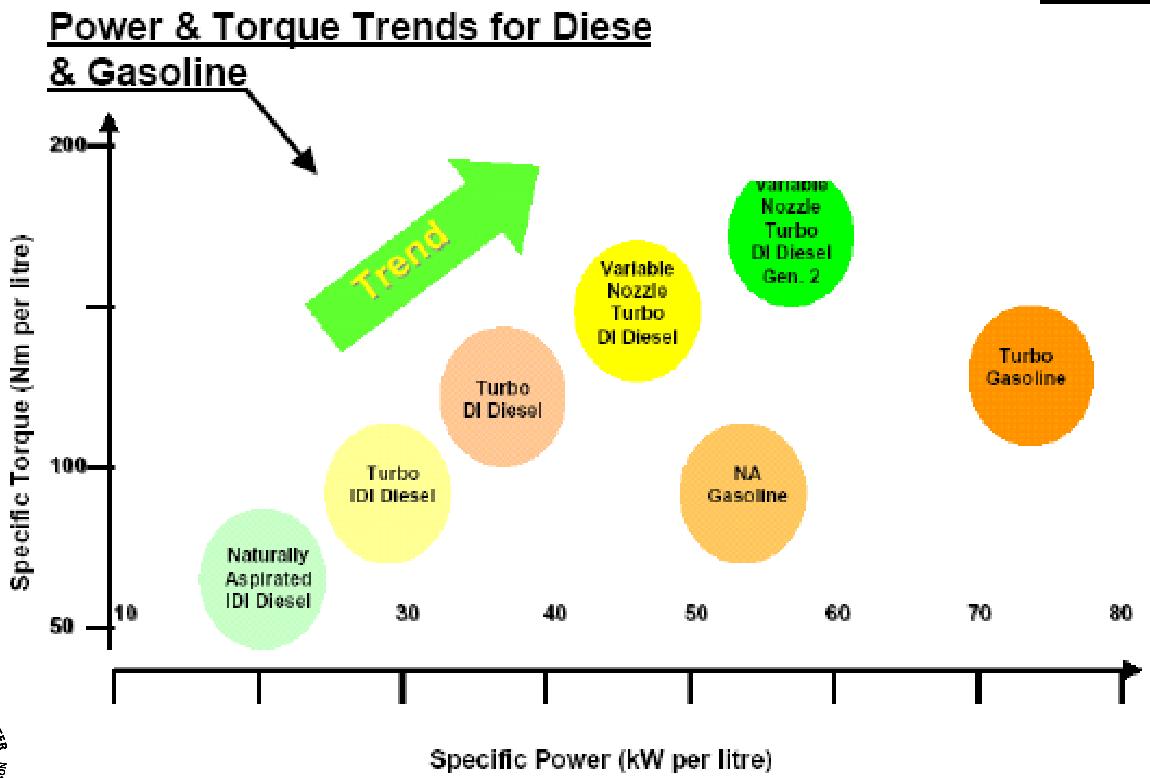


Turbocharging

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





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.

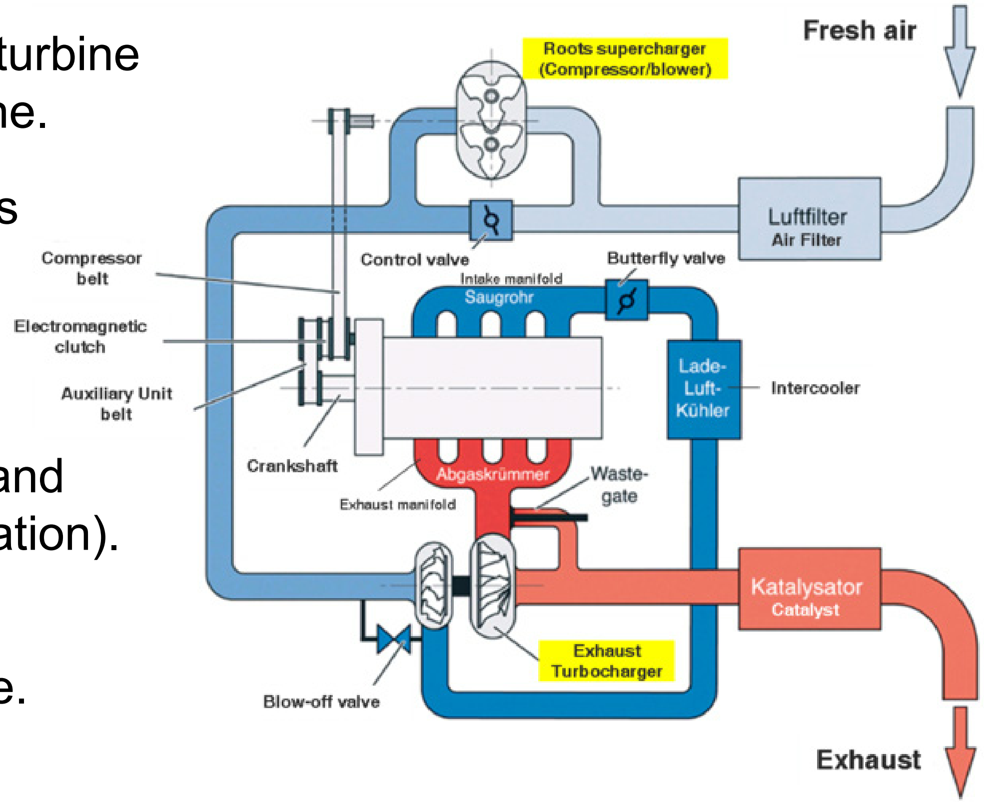
Turbocharger - connected compressor/turbine
- energy in exhaust used to drive turbine.

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).

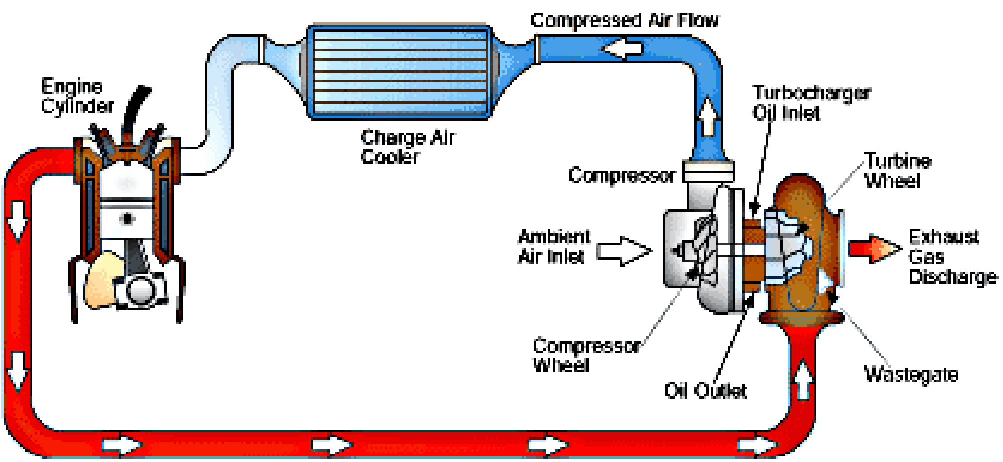
Intercooler after compressor
- controls combustion air temperature.

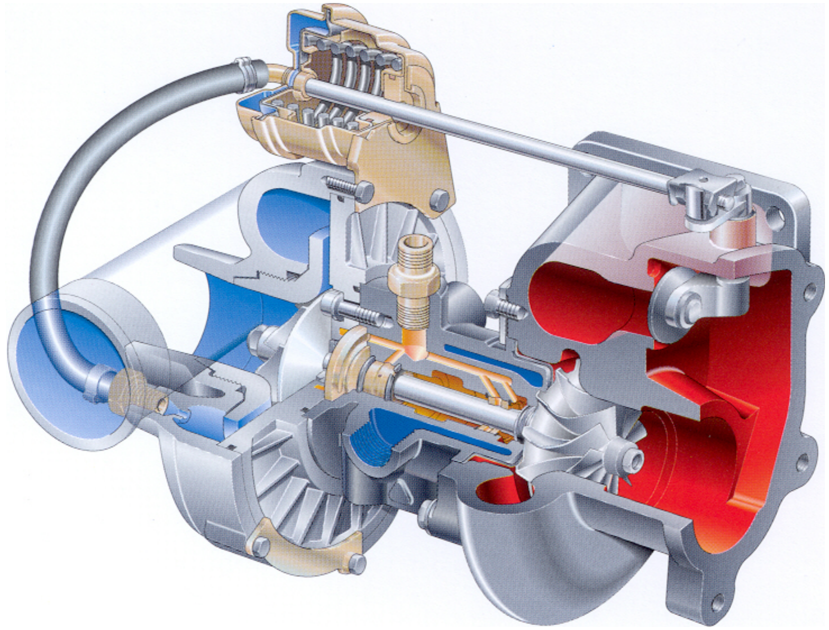
Air Flow in the VW Twincharged TSI

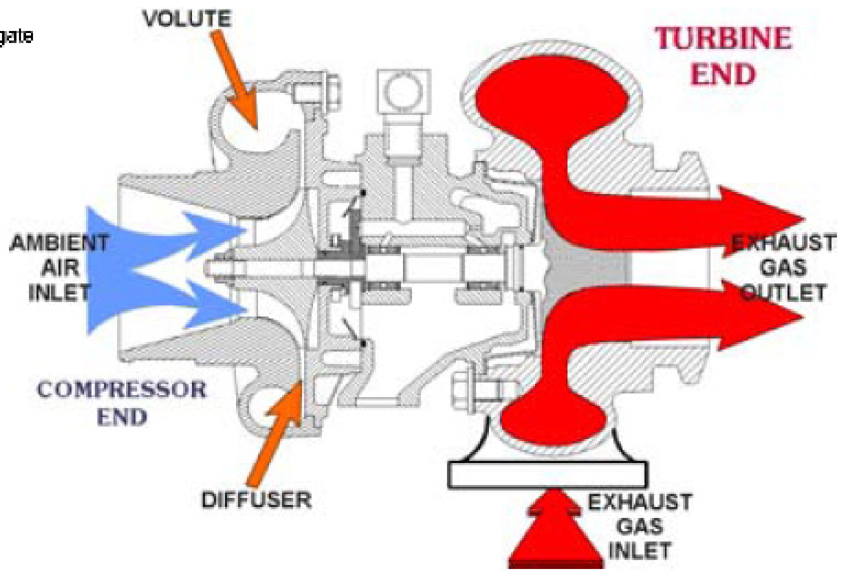


Turbocharging

Energy in exhaust is used to drive turbine which drives compressor





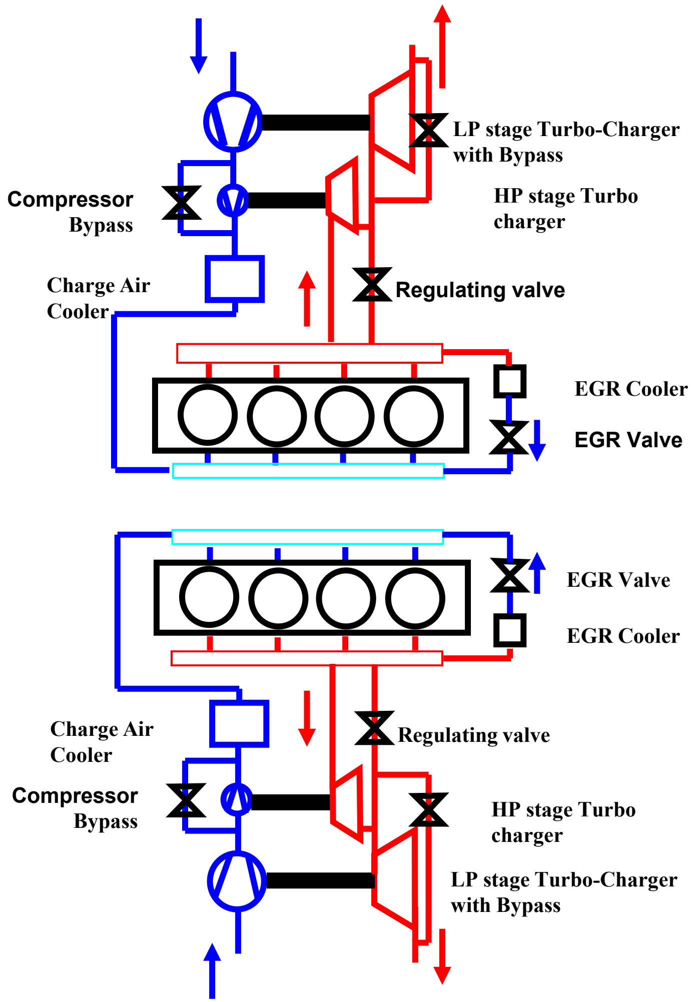


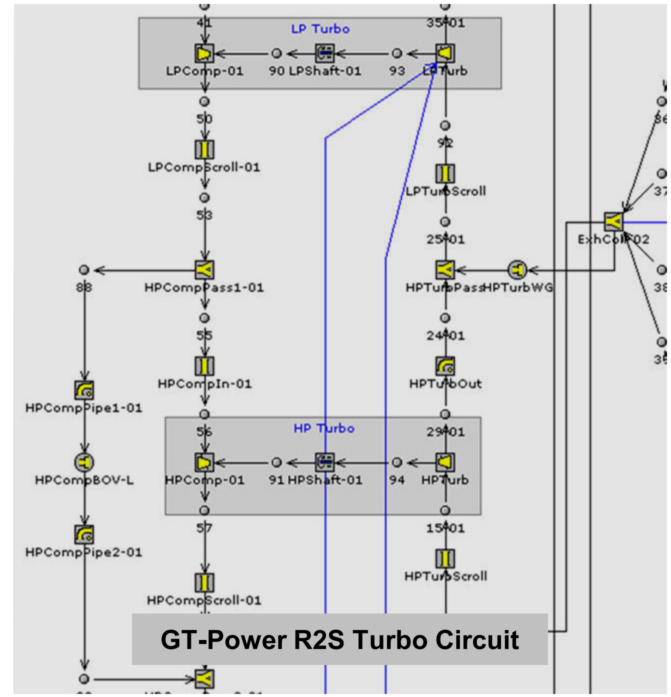
Wastegate used to by-pass turbine

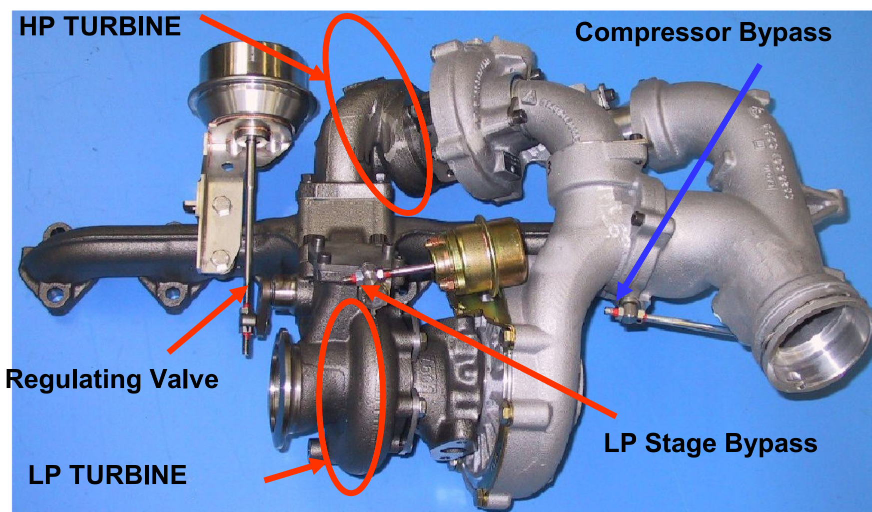
Charge air cooling after compressor further increases air density
- more air for combustion

Regulated two-stage turbocharger

Duplicated Configuration per Cylinder Bank







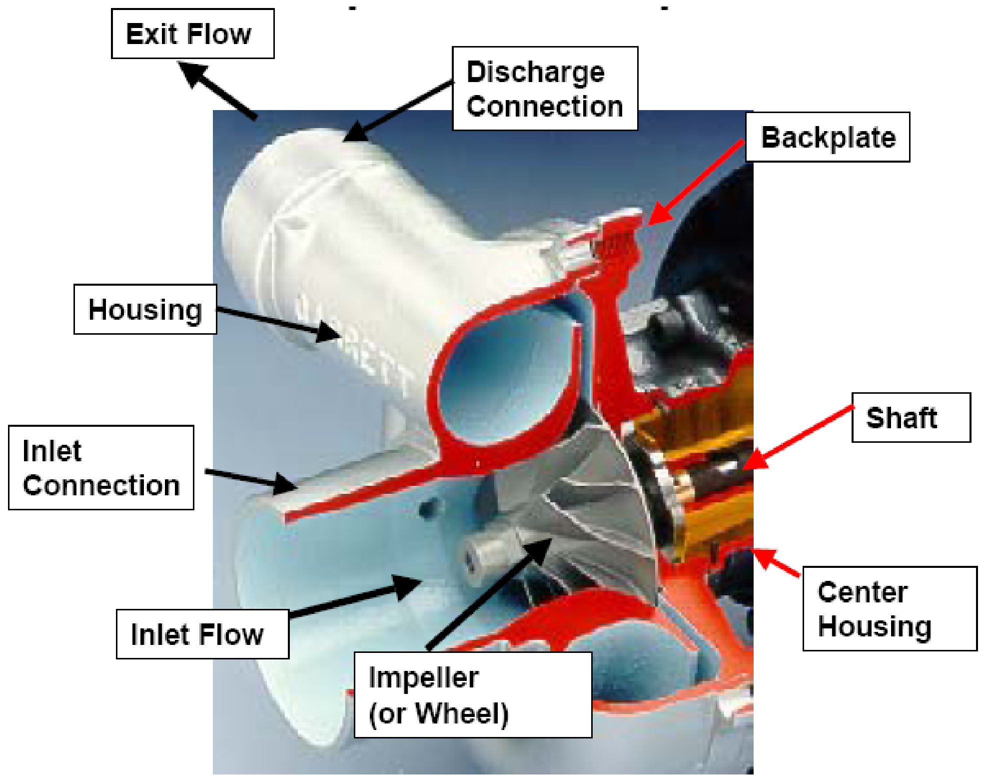
Automotive compressor

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

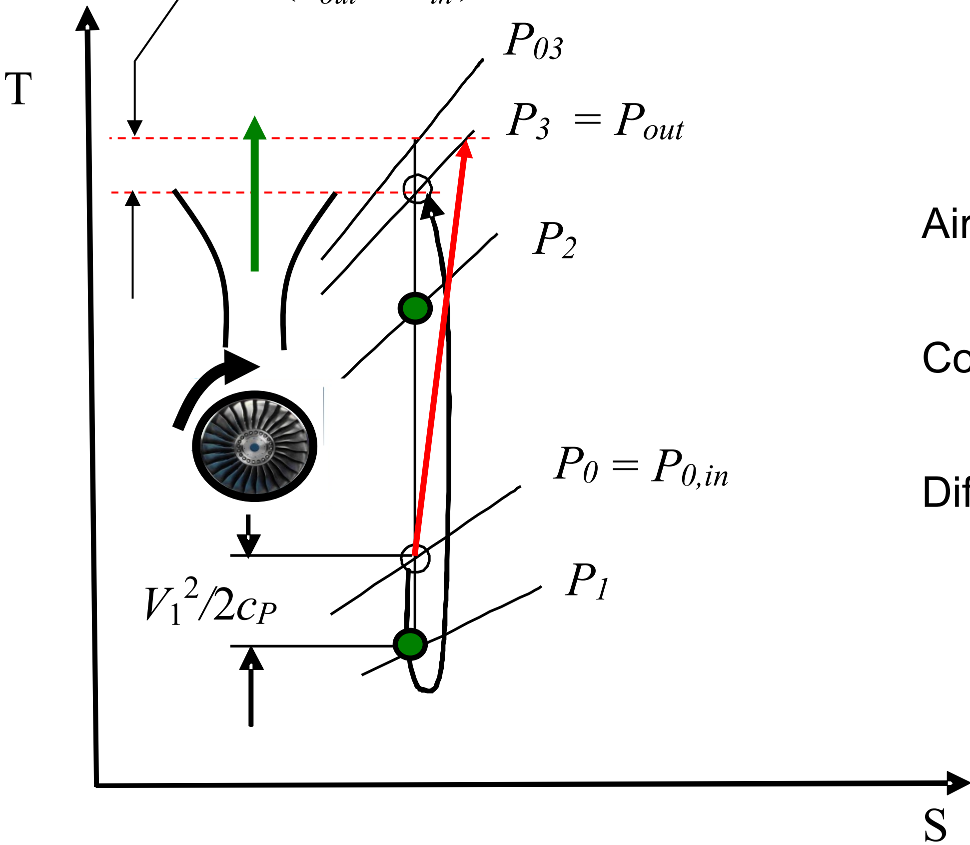
Consists of:
stationary inlet casing,
rotating bladed impeller,
stationary diffuser (w or w/o vanes)
collector - connects to intake system

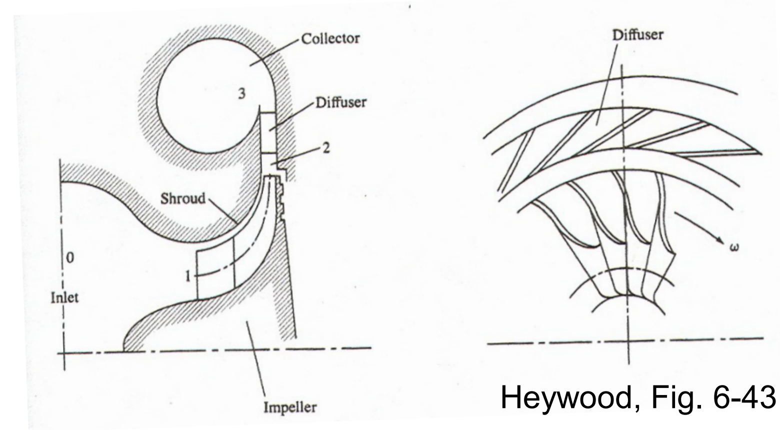


Heywood, 1988

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}

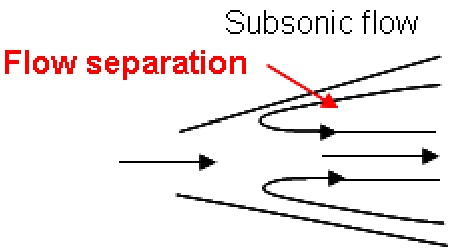
$$\dot{W}_c = \dot{m}\left(h_{out} - h_{in}\right)$$

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

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

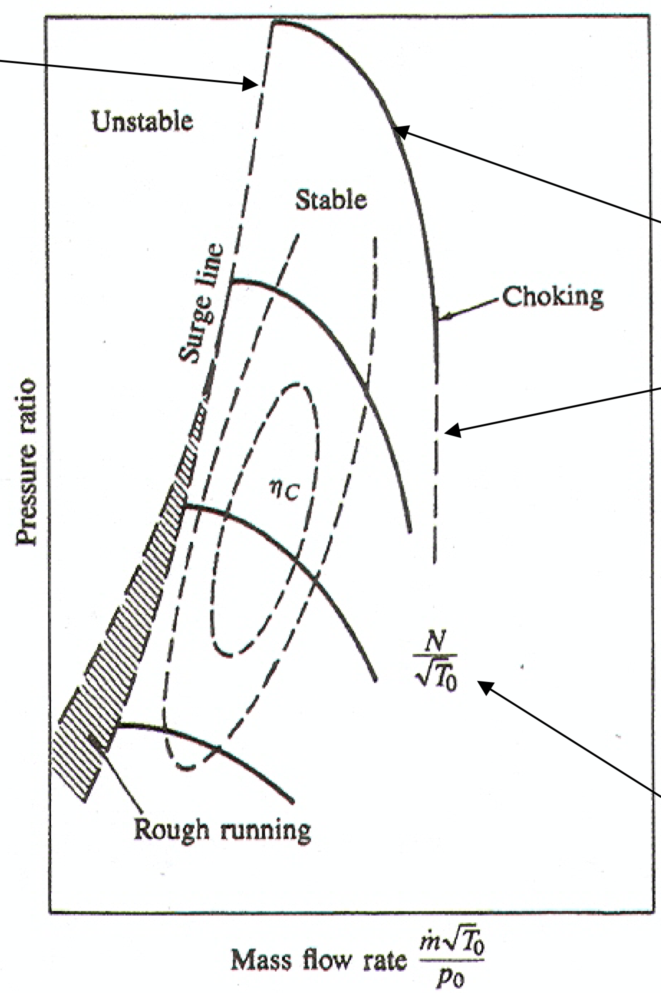
Compressor maps

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

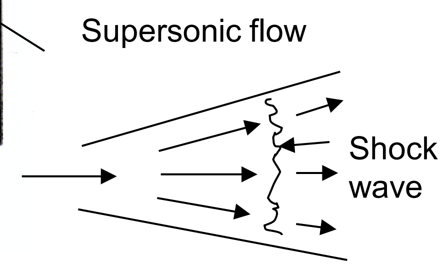


Heywood, Fig. 6-46

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

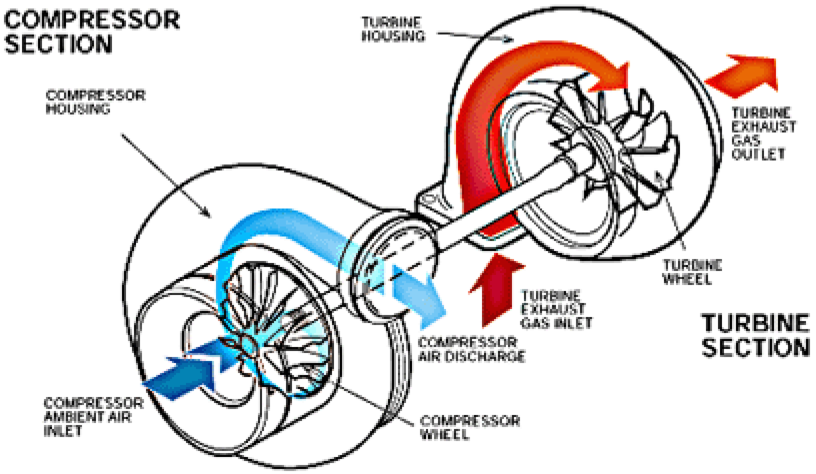


Compressor selection

To select compressor, first determine engine breathing lines.

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

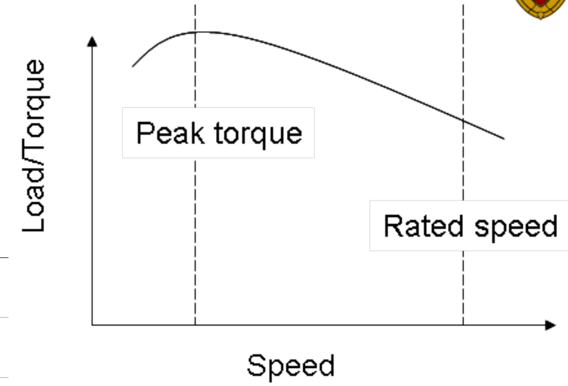
$$\dot{m}_{intake} = \left[\frac{\eta_{vol} \times D * N \times P_{ref}}{2 \times R \times T_{ref}} \right]$$

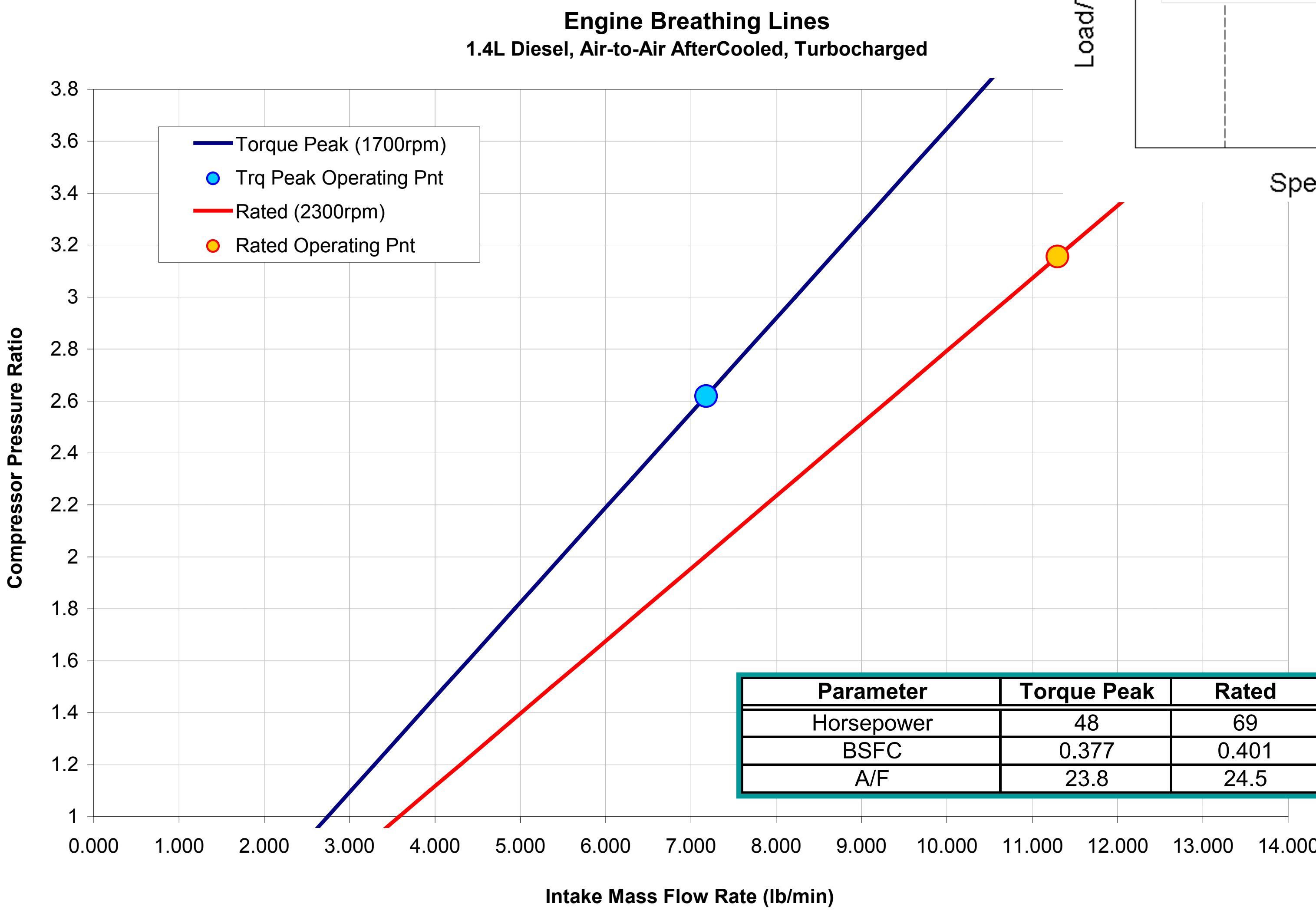


Where:

$\dot{m}_{intake}$ = 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



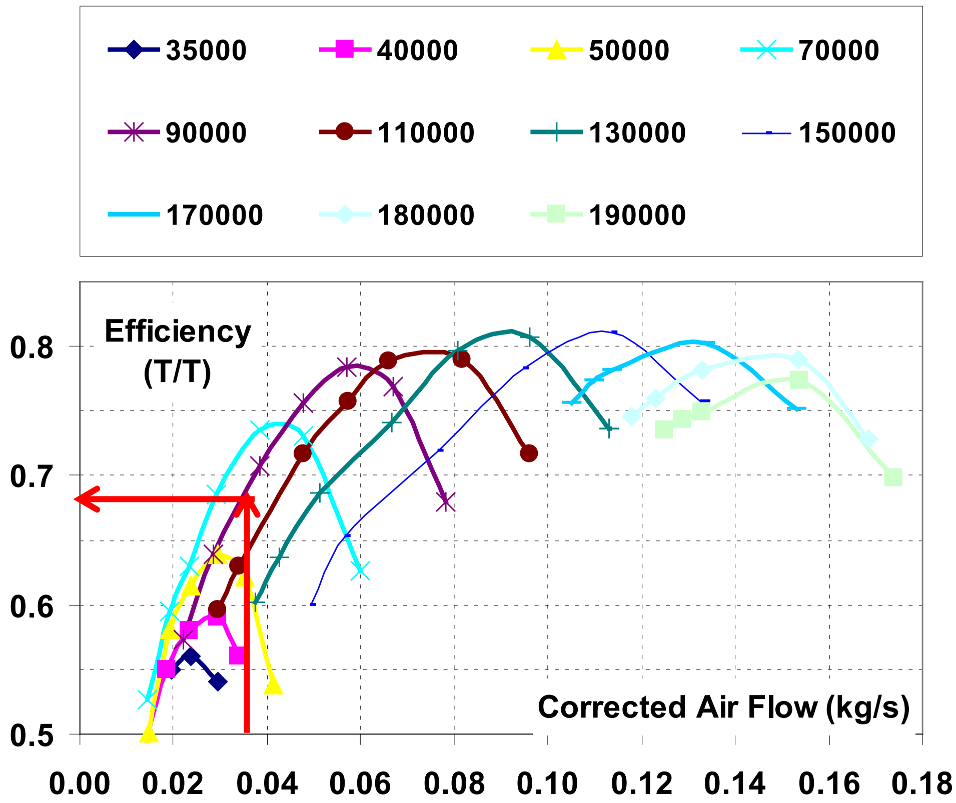


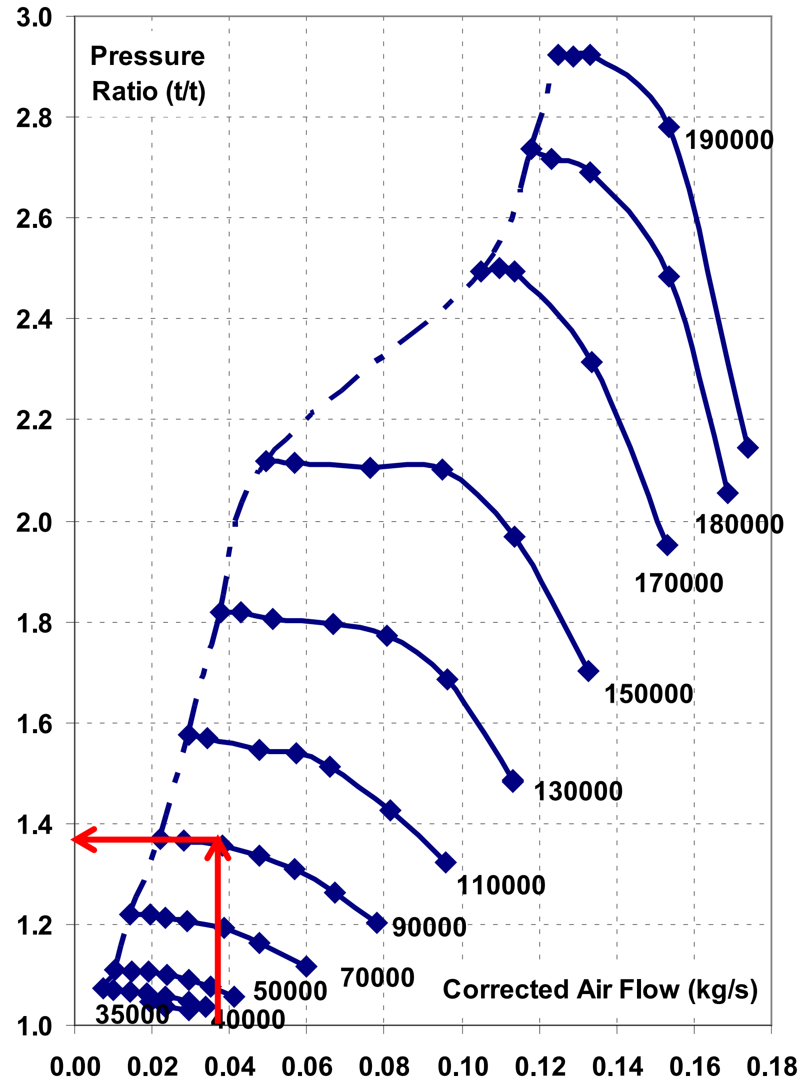
Parameter	Torque Peak	Rated	Units
Horsepower	48	69	hp
BSFC	0.377	0.401	lb/hp-hr
A/F	23.8	24.5	none

Serrano, 2008

Compressor maps

GM 1.9L diesel engine





Reitz, 2007

Automotive turbines

Naturally aspirated:

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

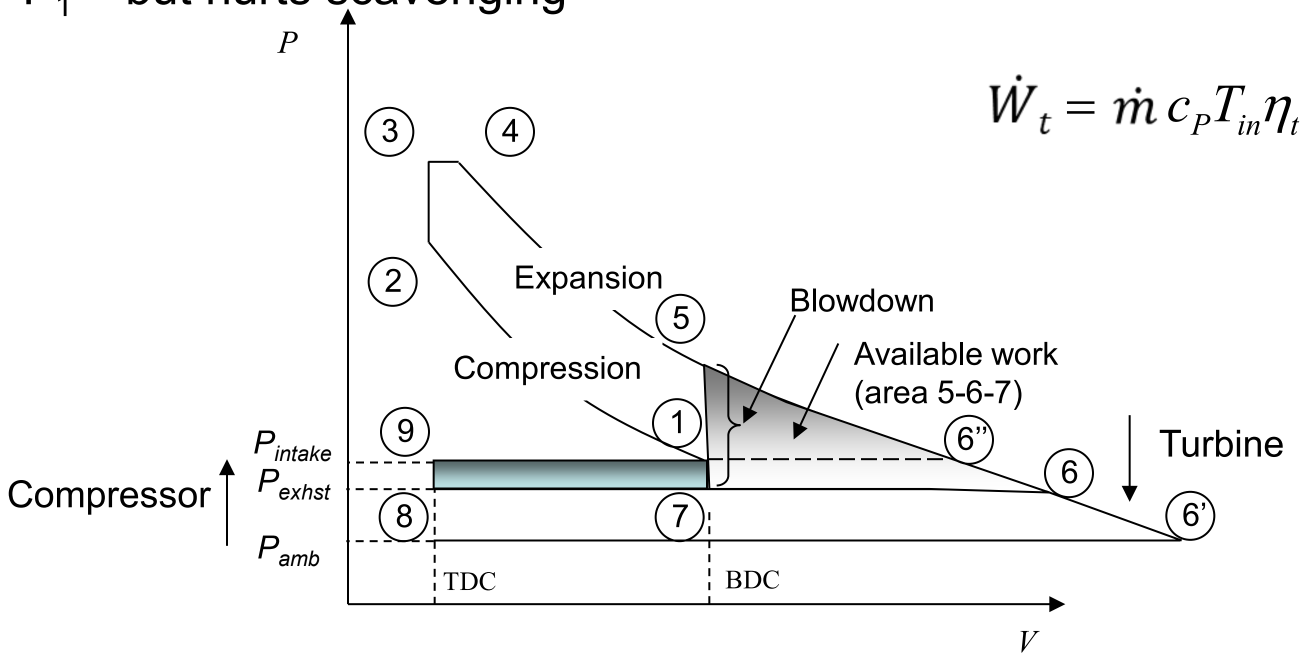
Boosted operation:

Negative pumping work:

$P_7<P_1$ – but hurts scavenging

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

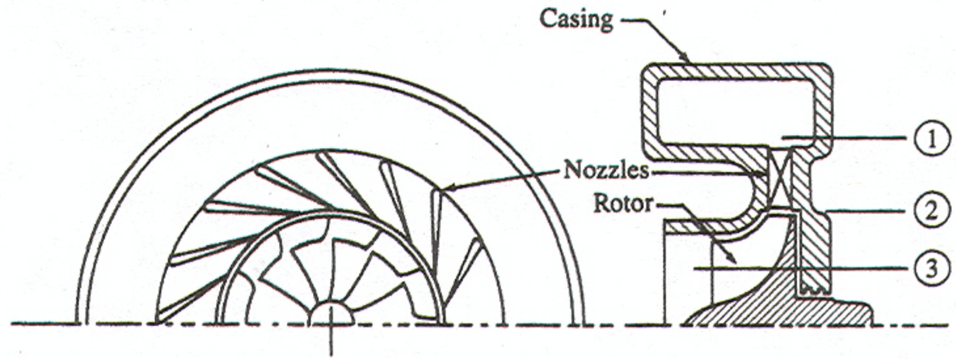
$$\dot{W}_t = \dot{m}\,c_P T_{in}\eta_t \left\{1 - \left[\frac{P_{0,out}}{P_{in}}\right]^{\frac{\gamma_g - 1}{\gamma_g}}\right\}$$

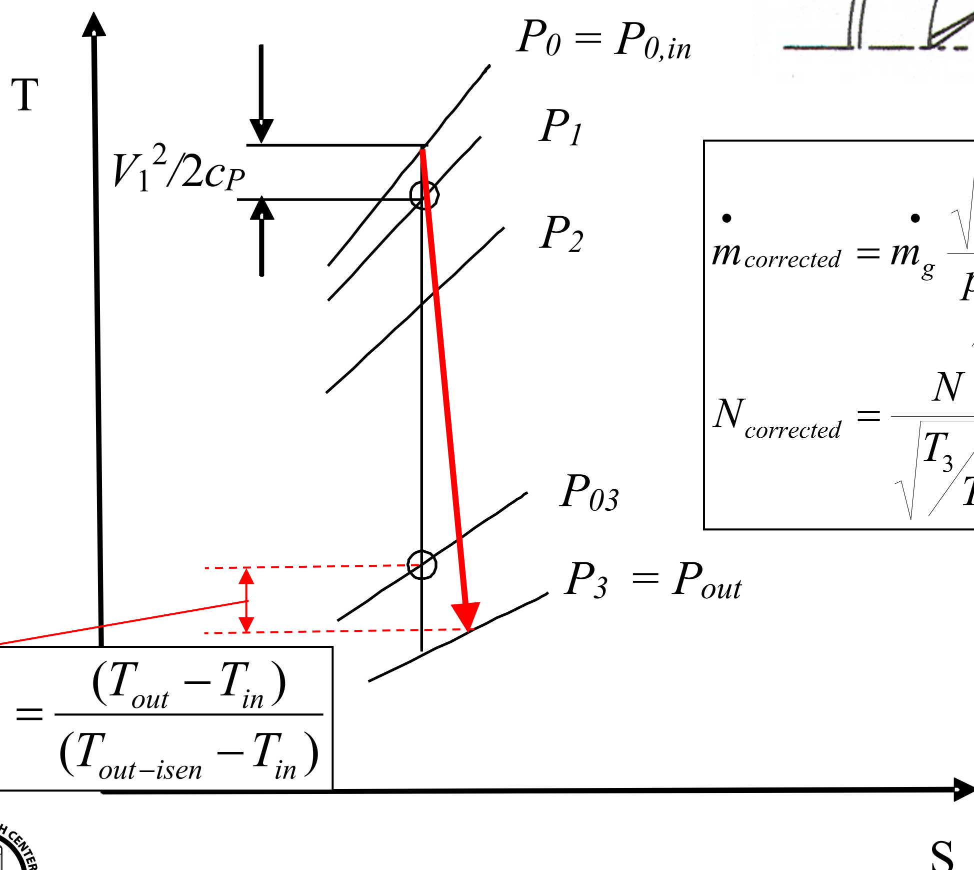


P-V diagram showing available exhaust energy
- turbocharging, turbocompounding, bottoming cycles and thermoelectric generators further utilize this available energy

Turbochargers

Radial flow – automotive;
axial flow – locomotive, marine

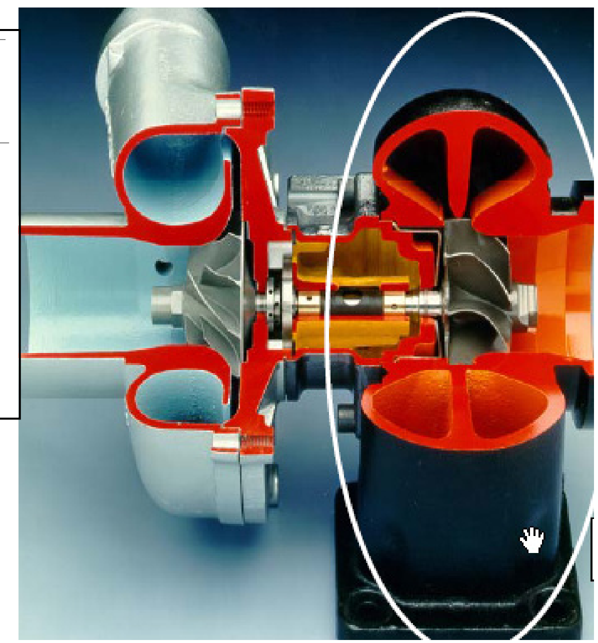




$$\eta_t = \frac{(T_{out} - T_{in})}{(T_{out-isen} - T_{in})}$$

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

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



out

Heywood, 1988

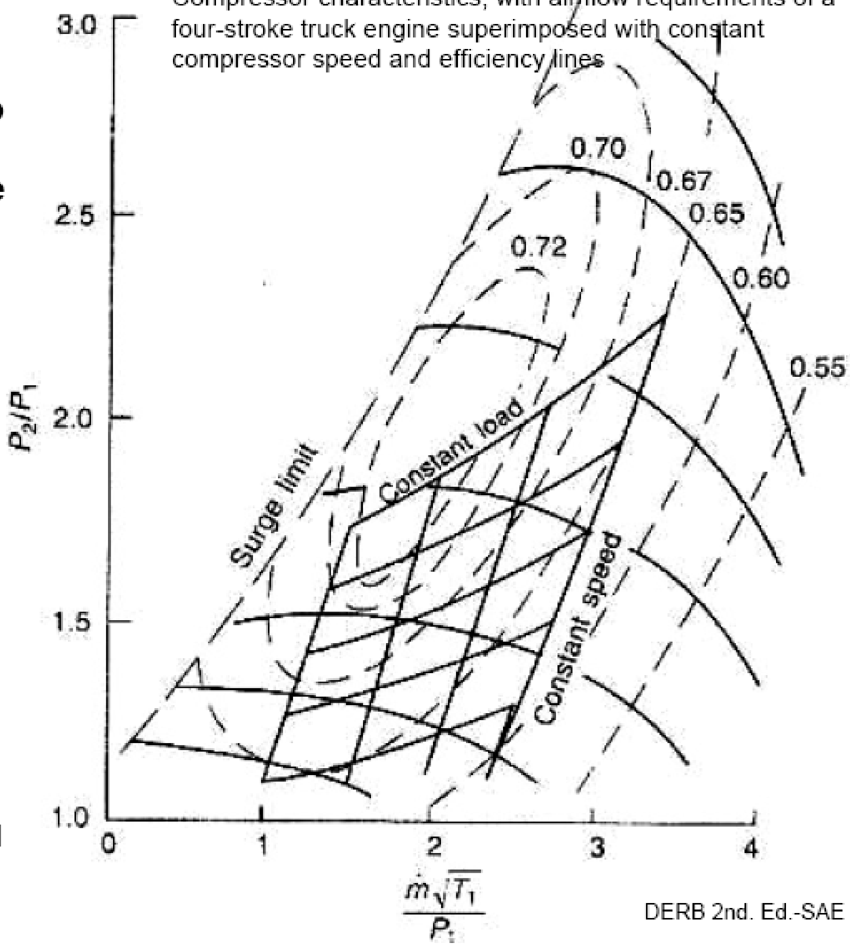
Matching

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



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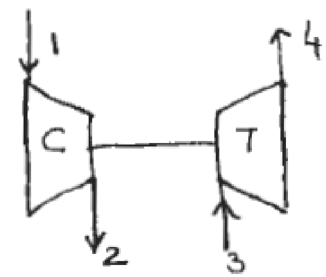
$$\dot{W}_t = \dot{W}_c$$



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

Summary

1-D models/codes based on thermodynamic models are available, and they are very useful for understanding charge preparation and engine breathing.

But, 1-D models require calibration against engine or theoretical data.

Turbocharging increases overall engine efficiency by using available energy in exhaust and by reducing pumping work.

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