Reciprocating Internal Combustion Engines

Prof. Rolf D. Reitz
Engine Research Center
University of Wisconsin-Madison

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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

Motivation

Society relies on IC engines for transportation, commerce and power generation:
utility devices (e.g., pumps, mowers, chain-saws, portable generators, etc.),
earth-moving equipment, tractors, propeller aircraft, ocean liners and ships,
personal watercraft and motorcycles

ICEs power the 600 million passenger cars and other vehicles on our roads today.
250 million vehicles (cars, buses, and trucks) were registered in 2008 in US alone.

50 million cars were made world-wide in 2009, compared to 40 million in 2000.
China became the world's largest car market in 2011.
A third of all cars are produced in the European Union, 50% are powered diesels.
→ IC engine research spans both gasoline and diesel powerplants.

Fuel Consumption

70% of the roughly 86 million barrels of crude oil consumed daily world-wide is used in IC engines for transportation.

10 million barrels of oil are used per day in the US in cars and light-duty trucks
4 million barrels per day are used in heavy-duty diesel engines,
- total oil usage of 2.5 gallons per day per person.

Of this, 62% is imported (at $80/barrel - costs US economy $1 billion/day).

US energy flow chart

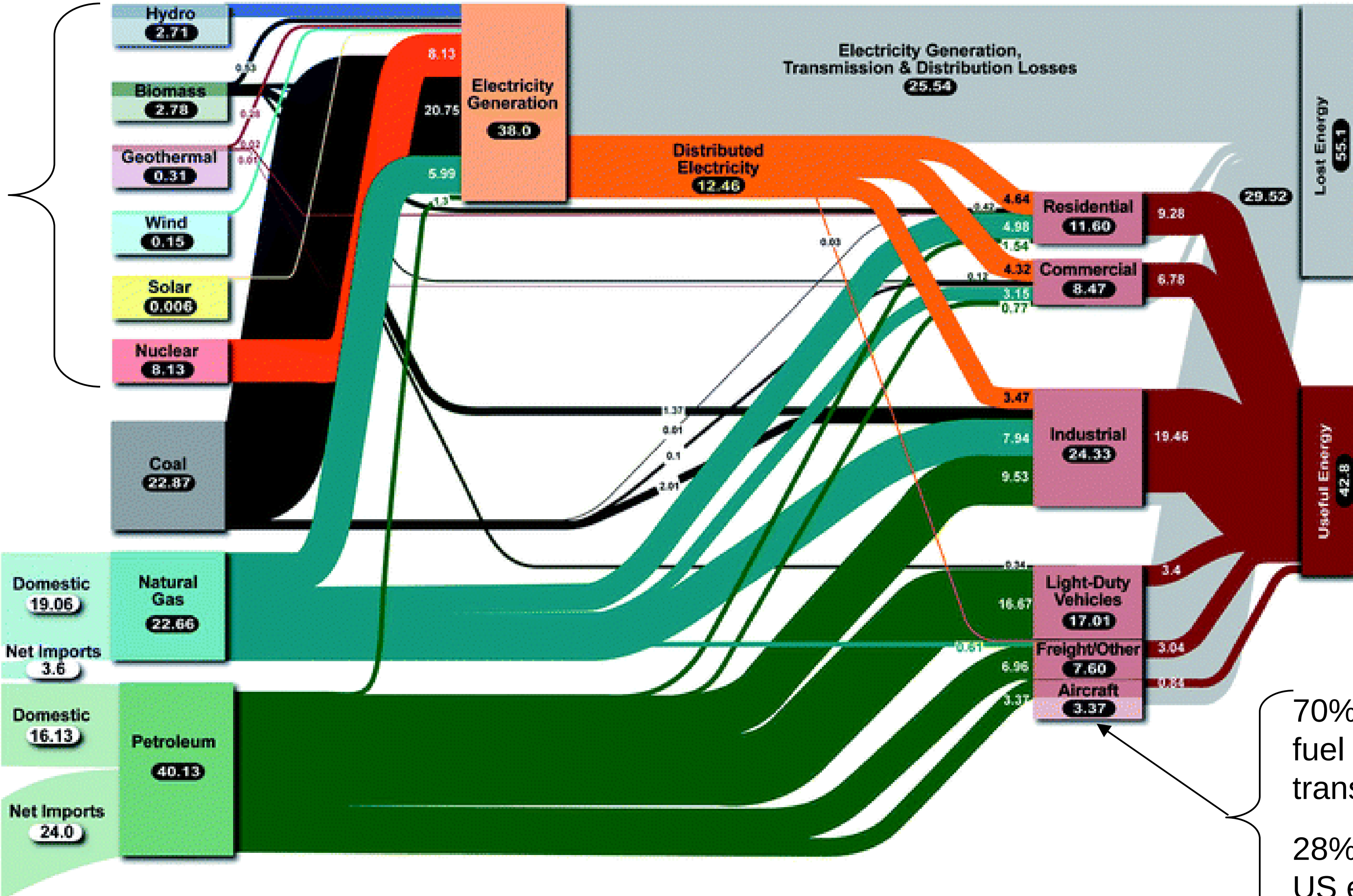
World energy use = 500 x 10^{18} J

14EJ

23EJ

23EJ

40EJ



100x10^{18}J

70% of liquid fuel used for transportation

28% of total US energy consumption

http://www.eia.gov/totalenergy/

Fuel consumption - CO_2 emissions

World oil use: 86 million bbl/day = 3.6 billion gal/day (~0.6 gal/person/day)

Why do we use fossil fuels (86% of US energy supply)?

Large amount of energy is tied up in chemical bonds.

Consider stoichiometric balance for gasoline (octane) in air:

$$C_8H_{18} + 12.5(O_2+3.76N_2) \rightarrow 8CO_2 + 9H_2O+47N_2 \quad (+\ 48\text{x}10^6\ J/kg_{fuel})$$

Kinetic energy of 1,000 kg automobile traveling at 60 mph (27 m/s)
$= 1/2 \cdot 1{,}000 \cdot 27^2$ ($m^2\ kg/s^2$ =Nm) ~$0.46\text{x}10^6$ J
= energy in 10g gasoline ~ 1/3 oz (teaspoon)

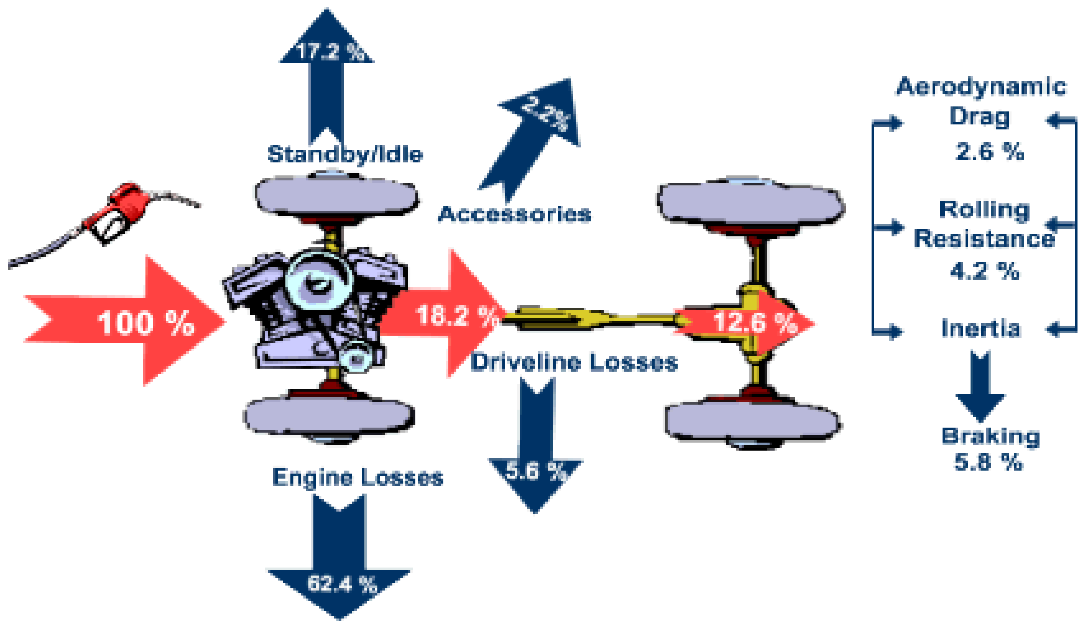
Assume:

1 billion vehicles/engines, each burns 2.5 gal/day (1 gal ~ 6.5lb ~ 3kg)
→ $7.5\text{x}10^9\ kg_{fuel}$/day*$48\text{x}10^6$ J/kg=$360\text{x}10^{18}$ J/yr

1 kg gasoline makes $8 \cdot 44/114$=3.1 kg CO_2
~ $365 \cdot 7.5\text{x}10^9\ kg_{fuel}$/yr ~ $8{,}486\text{x}10^9$ kg-CO_2/year ~ $8.5\text{x}10^9$ tonne-CO_2/year
(Humans exhale ~ 1 kg-CO_2/day = $6\text{x}10^9$ kg-CO_2/year)

Total mass of air in the earth's atmosphere ~ $5\text{x}10^{18}$ kg
So, CO_2 mass from engines/year added to earth's atmosphere
$8.5\text{x}10^{12} / 5\text{x}10^{18}$ ~ 1.7 ppm

Where does the energy go?



Engine Losses – 62.4%

In gasoline-powered vehicles, over 62% of the fuel's energy is lost in the internal combustion engine (ICE). ICE engines are very inefficient at converting the fuel's chemical energy to mechanical energy, losing energy to engine friction, pumping air into and out of the engine, and wasted heat.

Advanced engine technologies such as variable valve timing and lift, turbocharging, direct fuel injection, and cylinder deactivation can be used to reduce these losses.

In addition, diesels are about 30-35% more efficient than gasoline engines,

1%
(Prof. John Heywood, MIT)

Modern gasoline IC engine vehicle converts about 16% of the chemical energy in gasoline to useful work.

The average light-duty vehicle weighs 4,100 lbs.

The average occupancy of a light-duty vehicle is 1.6 persons.

If the average occupant weighs 160 lbs,

$0.16x((1.6x160)/4100) = 0.01$

http://www.fueleconomy.gov/feg/atv.shtml

Pollutant Emissions

37 billion tons of CO_2 (6 tons each for each person in the world) from fossil fuels/yr, plus other emissions, including nitric oxides (NOx) and particulates (soot).

CO_2 contributes to Green House Gases (GHG), implicated in climate change
- drastic reductions in fuel usage required to make appreciable changes in GHG

CO_2 emissions linked to fuel efficiency:
- automotive diesel engine is 20 to 40% more efficient than SI engine.

But, diesels have higher NOx and soot.
- serious environmental and health implications,
- governments are imposing stringent vehicle emissions regulations.
- diesel manufacturers use Selective Catalytic Reduction (SCR) after-treatment for NOx reduction: requires reducing agent (urea - carbamide) at rate (and cost) of about 1% of fuel flow rate for every 1 g/kWh of NOx reduction.

Soot controlled with Diesel Particulate Filters (DPF),
- requires periodic regeneration by richening fuel-air mixture to increase exhaust temperature to burn off the accumulated soot
- imposes about 3% additional fuel penalty.

Need for emissions control removes some of advantages of the diesel engine

Goal of IC engine:

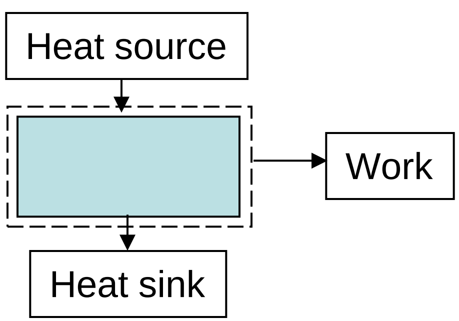
Convert energy contained in a fuel into useful work, as efficiently and cost-effectively as possible.

Identify energy conversion thermodynamics that governs reciprocating engines.
Describe hardware and operating cycles used in practical IC engines.
Discuss approaches used in developing combustion and fuel/air handling systems.

Internal Combustion Engine development
requires control to:
introduce fuel and oxygen, initiate and control combustion, exhaust products

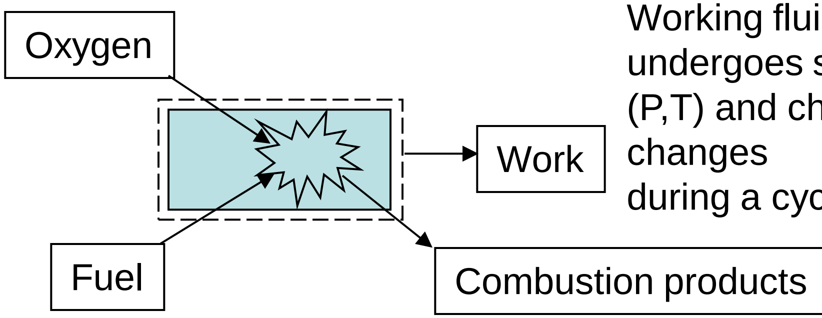
Heat (EC) engine (Carnot cycle)

Energy release occurs External to the system.
Working fluid undergoes reversible state changes (P,T) during a cycle
(e.g., Rankine cycle)



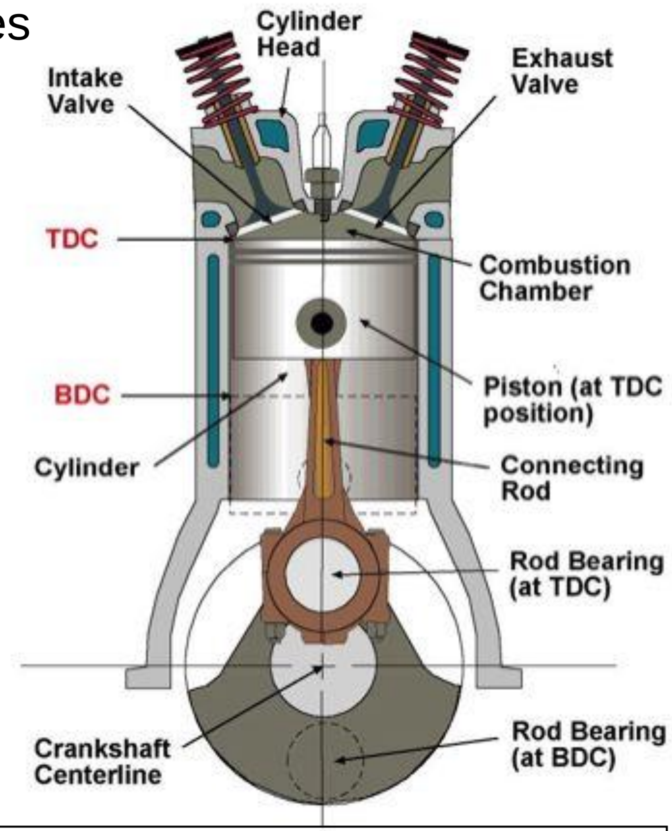
IC engine (Not constrained by Carnot cycle)

Energy release occurs Internal to the system.
Working fluid undergoes state (P,T) and chemical changes during a cycle



Components of piston engine

Piston moves between Top Dead Center (TDC) and Bottom Dead Center (BDC).
Compression Ratio = CR = ratio of BDC/TDC volumes
Stroke = S = travel distance from BDC to TDC
Bore = B = cylinder diameter
D = Displacement = (BDC-TDC) volume.# cylinders
$= \pi B^2 S/4$. # cylinders



Basic Equations

$P = W.N = T.N$
P [kW] = T [Nm].N [rpm].1.047 E-04

BMEP = P.(rev/cyc) / D.N
BMEP [kPa] = P [kW].(2 for 4-stroke) E03 / D [l]. N [rev/s]
BSFC = $\dot{m}_{fuel}$ / P
BSFC = $\dot{m}_{fuel}$ [g/hr] / P [kW]

Brake = gross indicated + pumping + friction
= net indicated + friction

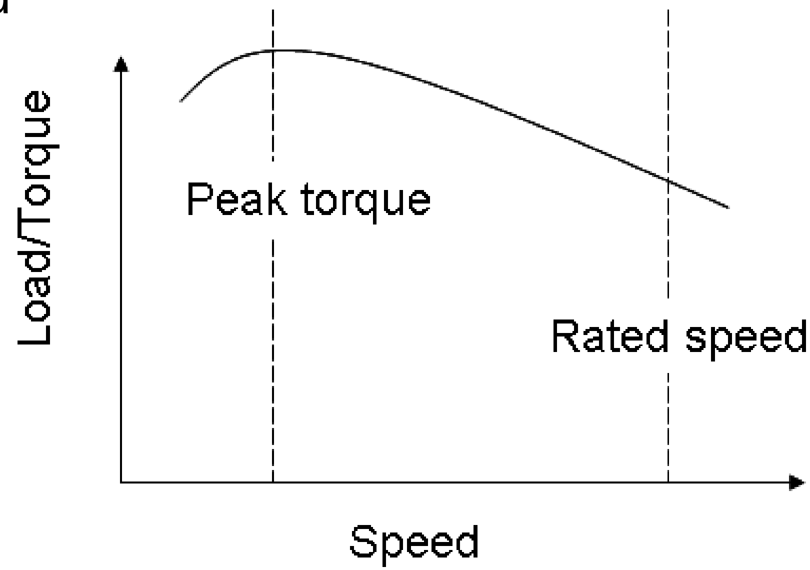
P = (Brake) Power [kW]
T = (Brake) Torque [Nm] = Work = *W*
BMEP = Brake mean effective pressure
$\dot{m}_{fuel}$ = fuel mass flow rate [g/hr]
BSFC = Brake specific fuel consumption

Engine Power

Heywood, 1988

Indicated power of IC engine at a given speed is proportional to the air mass flow rate, $\dot{m}_{air}$

$$P = \eta_f \,.\, \dot{m}_{air}\, N.\, LHV \,.\, (F/A) \,/\, n_r$$



η_f = fuel conversion efficiency
LHV = fuel lower heating value
F/A fuel-air ratio m_f/m_{air}
n_r = number of power strokes / crank rotation
= 2 for 4-stroke

Efficiency estimates:

SI: $270 < bsfc < 450$ g/kW-hr
Diesel: $200 < bsfc < 359$ g/kW-hr

η_f = 1/46 MJ/kg / 200 g/kW-hr = 40-50%

→500 MW GE/Siemens combined cycle gas turbine natural gas power plant ~ 60% efficient


SGT5-8000H ~530MW

4-stroke (Otto) cycle

1. Intake:
piston moves from TDC to BDC with the intake valve open, drawing in fresh reactants

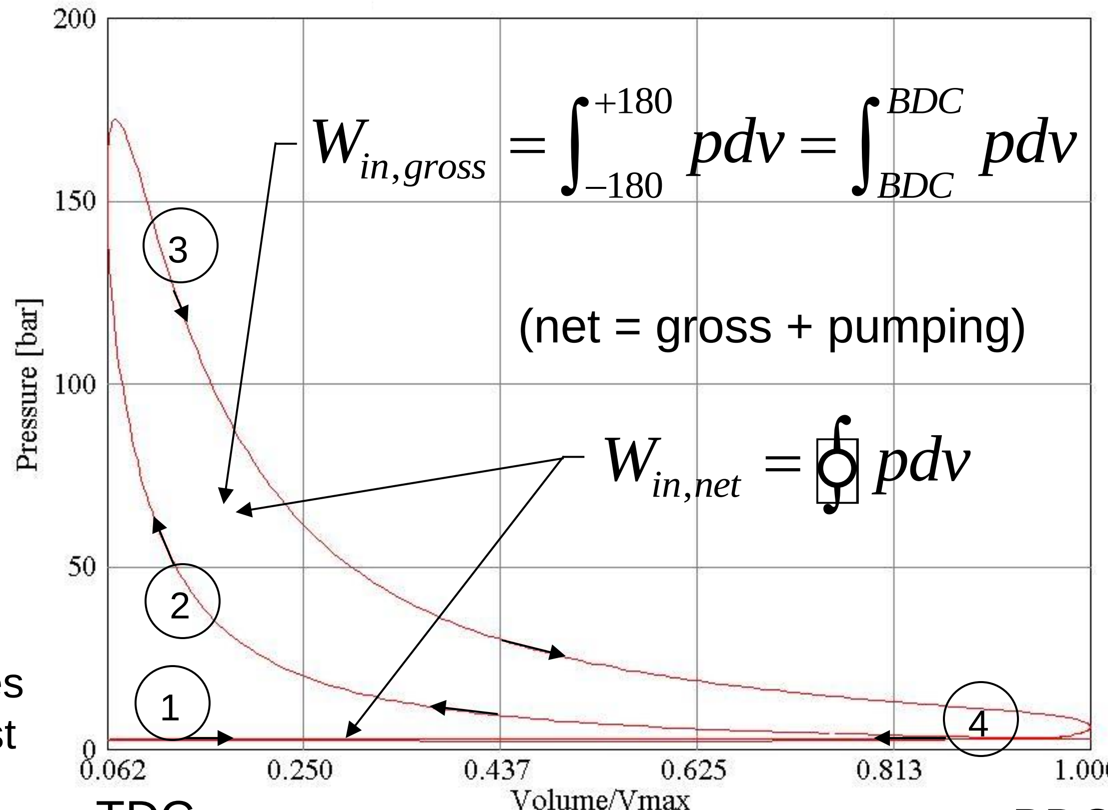
2. Compression:
valves are closed and piston moves from BDC to TDC,
Combustion is initiated near TDC

3. Expansion:
high pressure forces piston from TDC to BDC, transferring work to crankshaft

4. Exhaust:
exhaust valve opens and piston moves from BDC to TDC pushing out exhaust

1,4 Pumping loop – An additional rotation of the crankshaft used to:
- exhaust combustion products
- induct fresh charge

"Suck, squeeze, bang, blow"



$$W_{in,gross} = \int_{-180}^{+180} pdv = \int_{BDC}^{BDC} pdv$$

(net = gross + pumping)

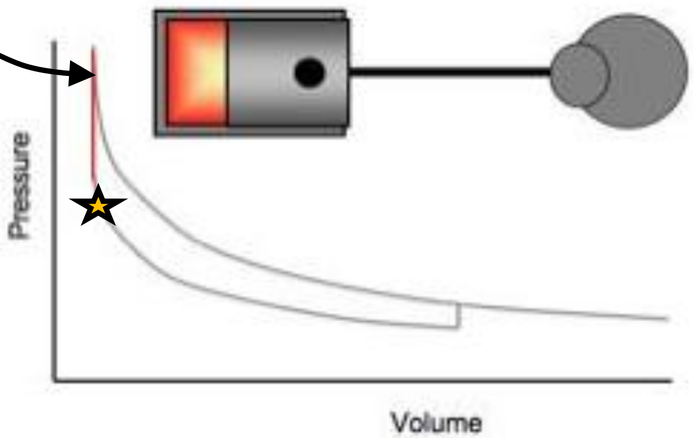
$$W_{in,net} = \oint pdv$$

Four-stroke diesel pressure-volume diagram at full load

<u>Combustion process</u> - initiated near end of compression stroke.

Instantaneous combustion has high theoretical efficiency, but is impractical due to need to manage peak pressures and due to high heat transfer.

<u>Spark-ignition engine:</u>
mixture of air (oxygen carrier) and fuel enters chamber during intake process. Mixture is compressed - combustion initiated using a high-energy electrical spark.



f. Combustion process

<u>Compression-ignition (Diesel) engine:</u>
air alone is drawn into chamber, compressed. Fuel injected directly into chamber near end of compression process.
(Fuel used in compression-ignition engine must easily spontaneously ignite when exposed to high temperature and pressure compressed air.)

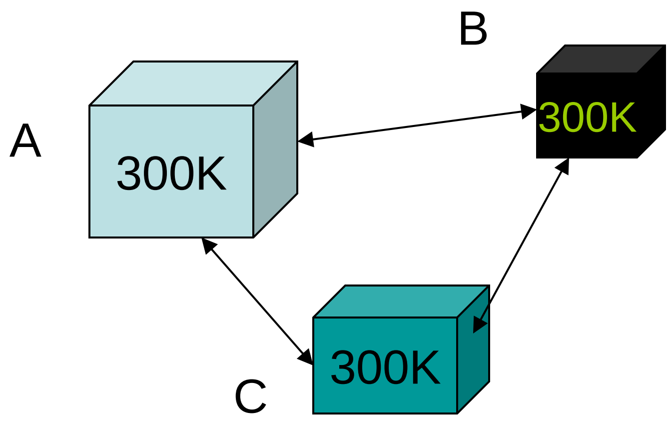
Diesel is often portrayed as having a slower combustion process (constant pressure instead of constant volume)

Goal of rapid combustion near TDC for maximum efficiency is true for both Diesel and spark-ignition engines.

Heywood, 1988

Thermodynamics review – Zero'th law

1. Systems in thermal equilibrium are at the same temperature
2. If two thermodynamic systems are in thermal equilibrium with a third, they are also in thermal equilibrium with each other.

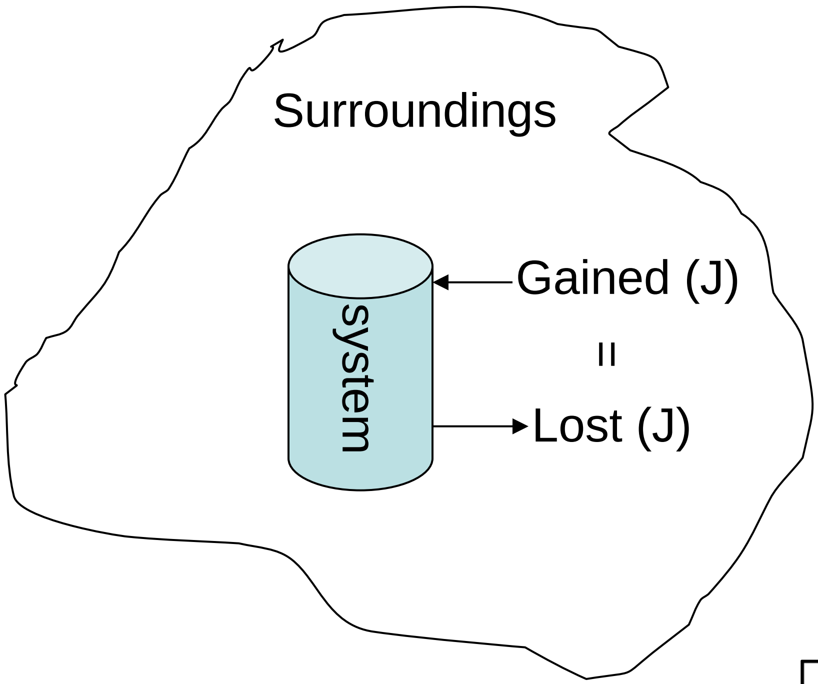


Thermal equilibrium

Heywood, 1988

Thermodynamics review - First law

During an interaction between a system and its surroundings, the amount of energy gained by the system must be exactly equal to the amount of energy lost by the surroundings



Engine System

Gained (input) (J)

- Intake flow
- Energy of fuel combustion

Lost (output) (J)

- Work
- \+ Heat Lost (Cylinder wall, Exhaust gas)
- Friction

$$de = dq - dw$$

Heywood, 1988

Thermodynamics review - Second law

The second law asserts that energy has quality as well as quantity (indicated by the first law)

$$ds = \frac{\delta q}{T} + ds_{irrev}$$

$$ds_{irrev} \geq 0$$

Engine research:

Reduce irreversible losses → Increase thermal efficiency

Heywood, 1988

Equations of State

Thermal: $Pv = RT$ where $R = R_u / W$

Caloric: $de = c_v dT$ and $dh = c_p dT$

Enthalpy: $h = e + Pv$

Ratio of specific heats: $\gamma = \dfrac{c_p}{c_v}$

$$c_v = \frac{R}{\gamma - 1}$$

$$c_p = \frac{\gamma R}{\gamma - 1}$$

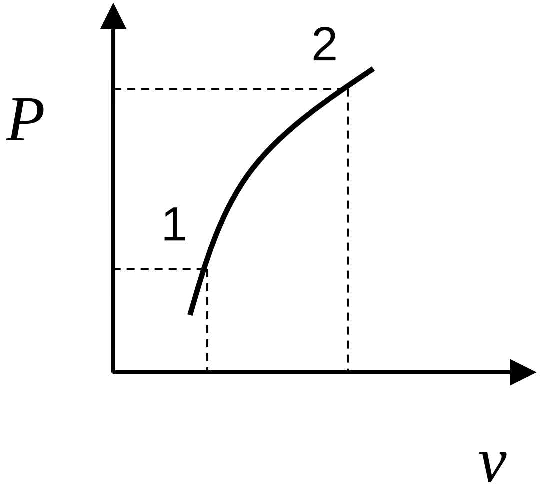
Calculation of Entropy

Gibbs' equation: $Tds = de - vdP$

and

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}$$



Heywood, 1988

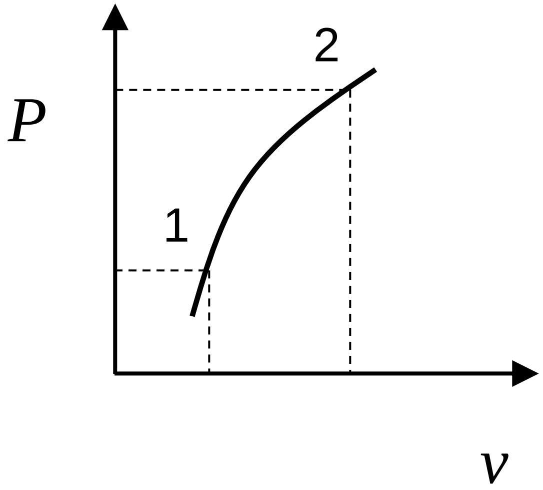
Isentropic process

Adiabatic, reversible ideal reference process

$$0 = s_2 - s_1 = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$$

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = \left(\frac{T_2}{T_1}\right)^{\gamma/(\gamma-1)}$$

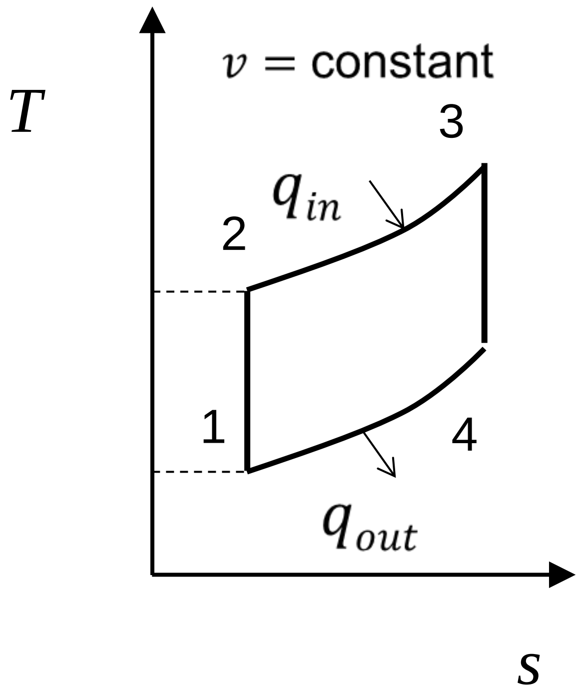
$$0 = s_2 - s_1 = c_v \ln \frac{T_2}{T_1} + R \ln \frac{v_2}{v_1}$$



Heywood, 1988

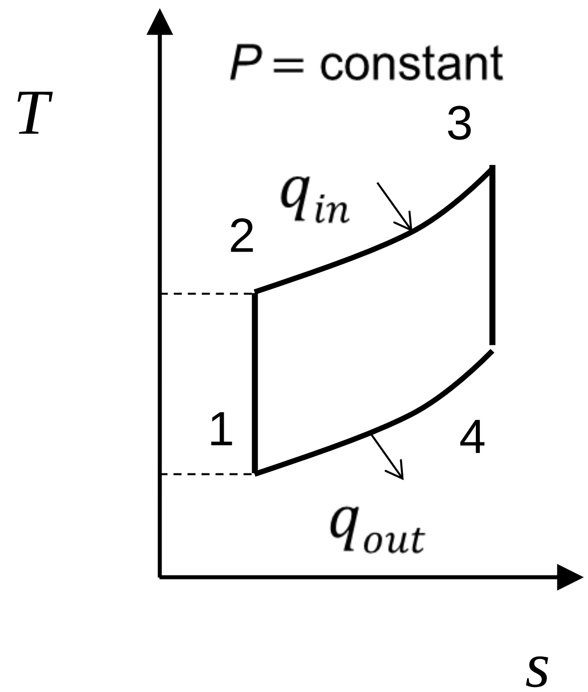
Ideal cycles

Otto



1-2 Isentropic compression
2-3 Constant volume heat addition
3-4 Isentropic expansion
4-1 Constant volume heat rejection

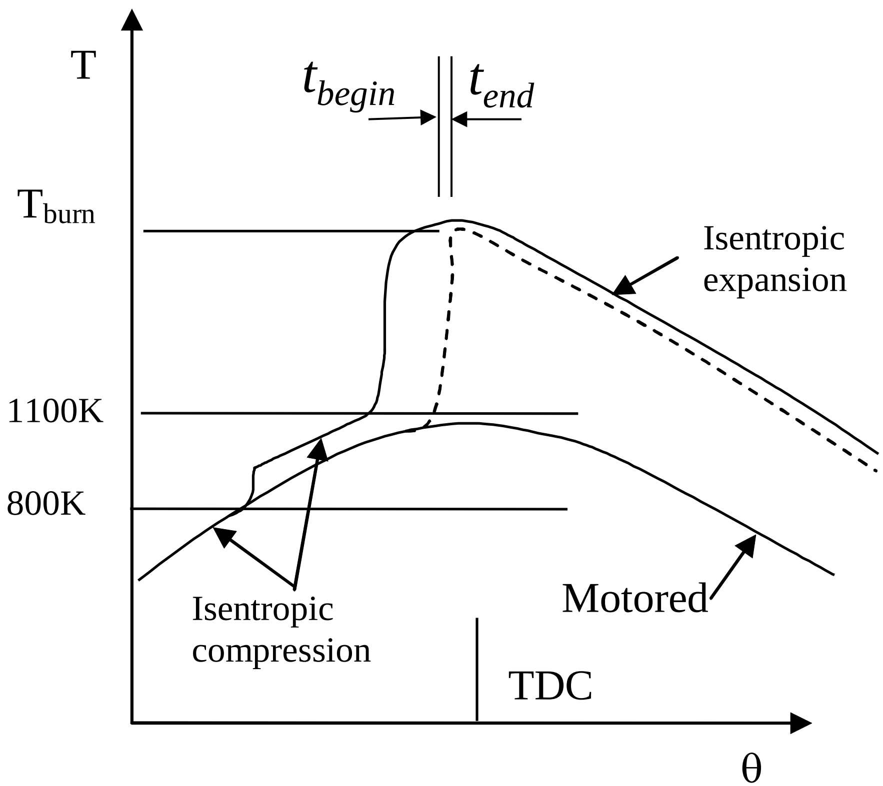
Diesel

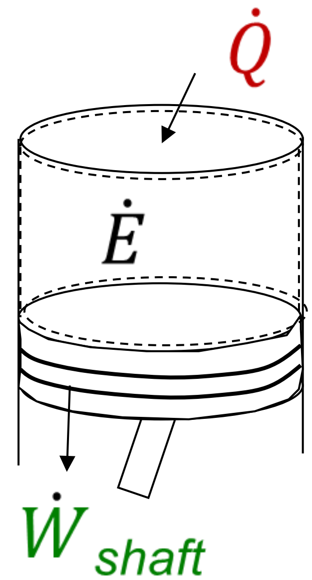


1-2 Isentropic compression
2-3 Constant pressure heat addition
3-4 Isentropic expansion
4-1 Constant volume heat rejection

Constant volume combustion - HCCI:

$$d\dot{E} = d\dot{Q} - d\dot{W}_{shaft}$$





During constant volume combustion process:

$$t_{begin} - t_{end} \longrightarrow 0$$

$$\dot{W}_{Shaft} = \int_{t_{begin}}^{t_{end}} P d\forall = 0$$

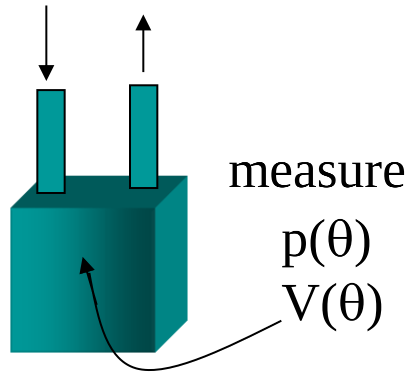
$$Q = \int_{t_{begin}}^{t_{end}} \dot{Q} dt = m_f \cdot Q_{LHV}$$

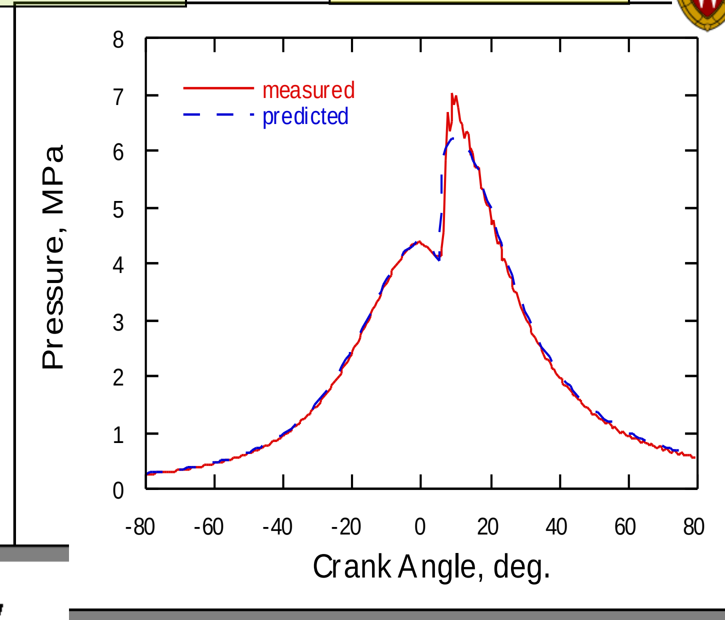
$$T_{burn} = T_{unburn} + \frac{(\gamma - 1)}{R} m_f Q_{LHV}$$

Heywood, 1988

Zero-Dimensional models

Single zone model





1st Law of Thermodynamics

$$mc_v \frac{dT}{dt} + p\frac{dV}{dt} + \sum_j \dot{m}_j h_j = q_{Comb} - q_{Loss} = q_{Net}$$

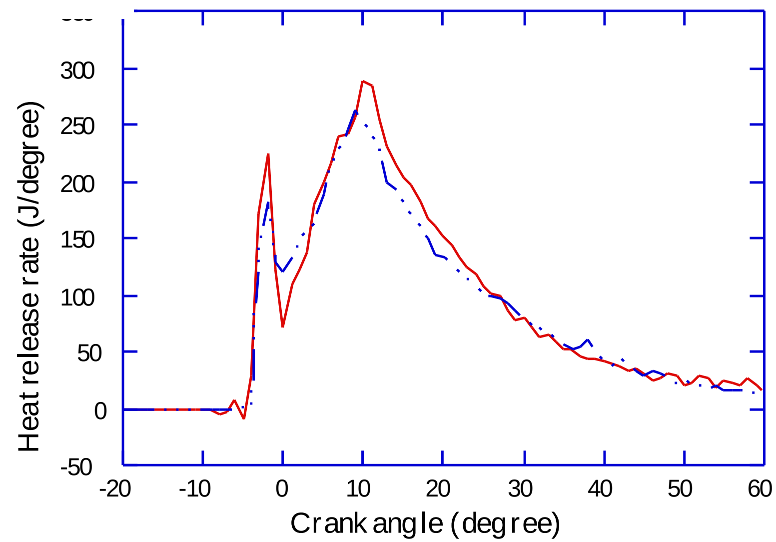
Use the ideal gas equation to relate p & V to T

$$q_{Net} = p\frac{dV}{dt} + \frac{1}{\gamma - 1}\frac{dpV}{dt}$$

where

$$q_{Loss} = hA(T - T_{wall})$$

Assume h and T_{wall}

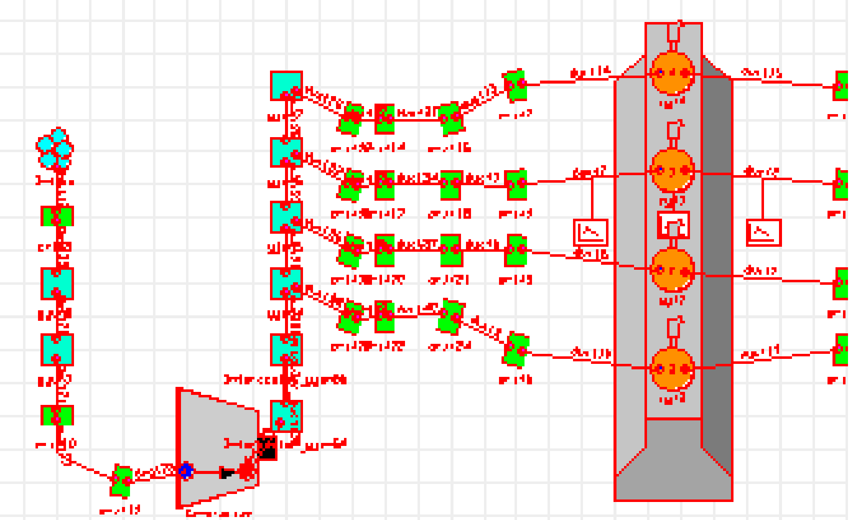


1-D Models

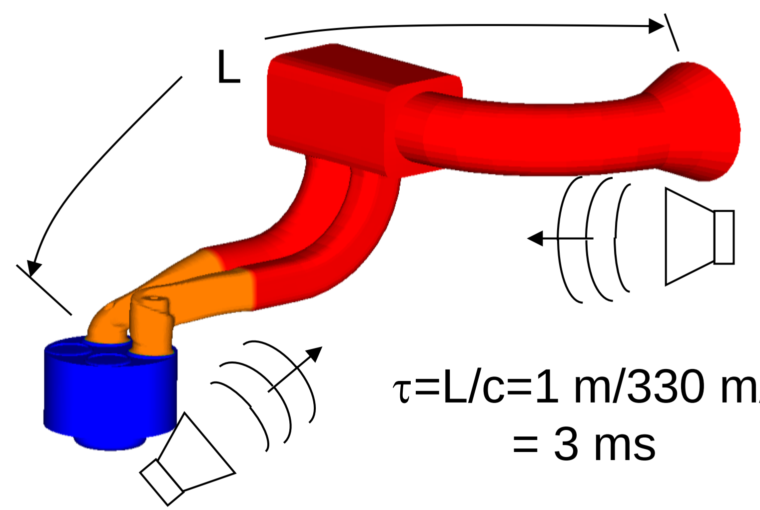
1-D codes (e.g., GT-Power, AVL-Boost, Ricardo WAVE) predict wave action in manifolds
At high engine speed valve overlap can improve engine breathing
→ inertia of flowing gases can cause inflow even during compression stroke.

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



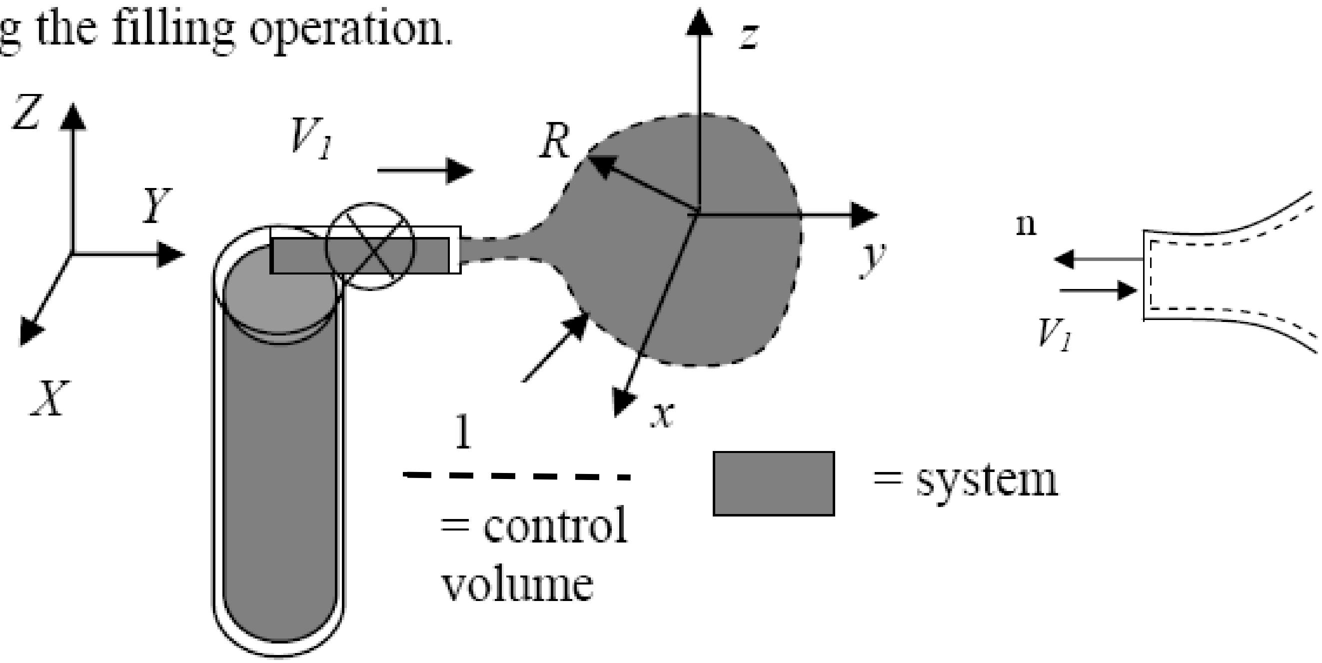
$\tau = L/c = 1$ m/330 m/s
= 3 ms

1 ca deg = 0.1 ms @ 1800 rev/min

Anderson, 1990

Control volumes and systems

Fluid enters a balloon at the valve (Station 1) at velocity V1 and the control volume deforms during the filling operation.



The Reynold's Transport Equation written for a coordinate system placed on the balloon (*xyz*) becomes

$$\left.\frac{dM}{dt}\right)_{system} = \frac{d}{dt}\int_{cv} \rho \, d\forall + \int_{cs} \rho \mathbf{V}_{rel} \cdot \mathbf{n} \, dA = 0$$

$$\frac{d}{dt}\left(\rho \frac{4}{3}\pi R^3\right) = \rho_1 V_1 A_1 .$$

Anderson, 1990

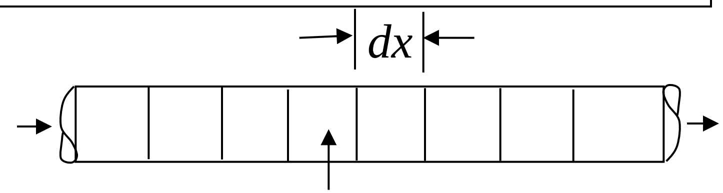
1-D compressible flow

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

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

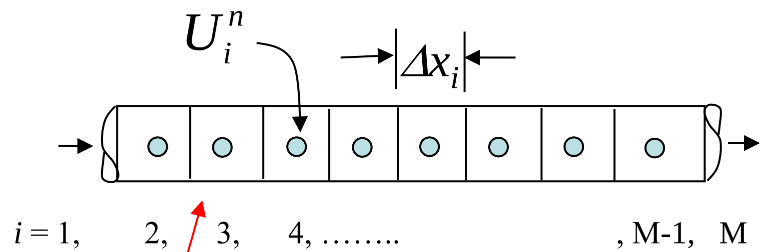
$$\dot{Q} = \dot{q}\rho A dx$$

5 unknowns U: ρ, V, e, P, and T

5 equations for variation of flow variables in space and time

Need to evaluate derivatives $\partial/\partial x, \partial/\partial t$

Numerical solutions



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

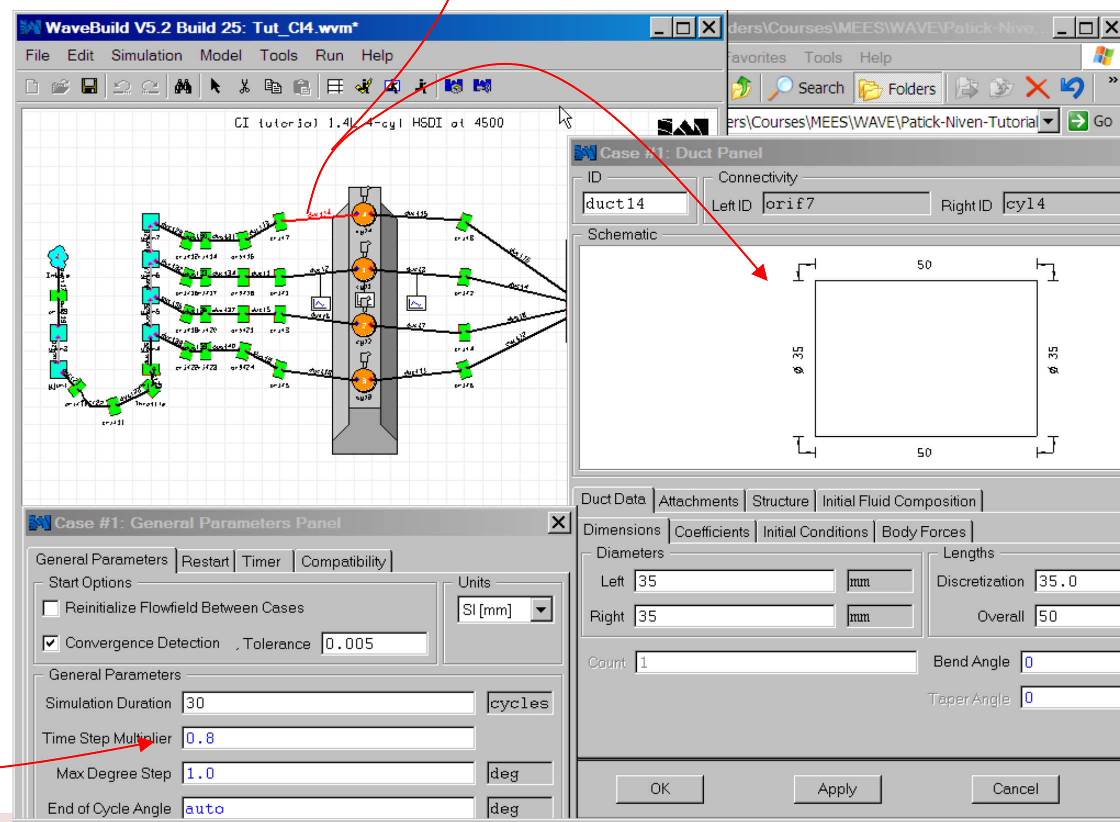
$$t = n\Delta t \quad n = 0, 1, 2, 3, \ldots.$$

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

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

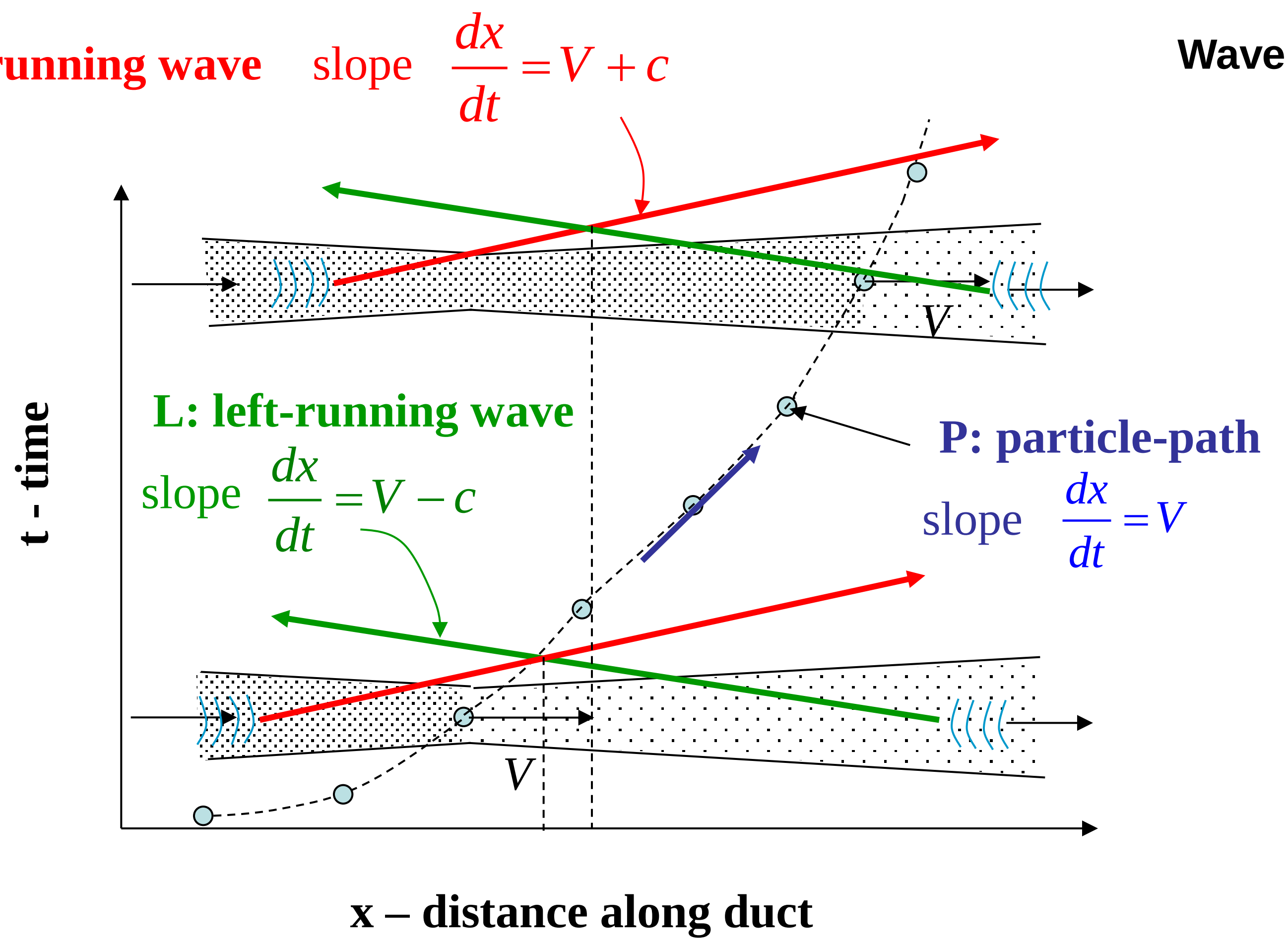
Considerations of stability require the Courant-Friedrichs-Levy (CFL) condition

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



Anderson, 1990

Analytical solutions – Method of Characteristics



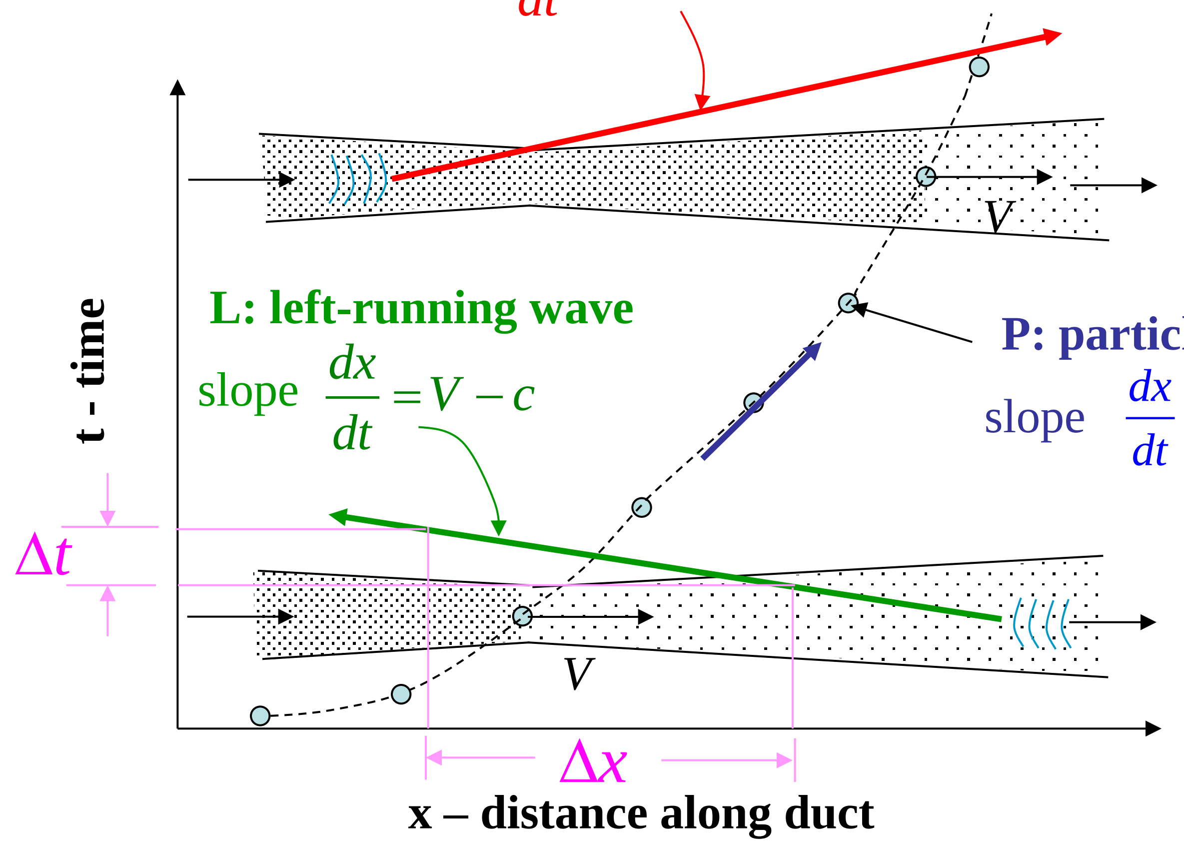
All points continuously receive information about both upstream and downstream flow conditions from both left and right-running waves. These waves originate from all points in the flow.

Anderson, 1990

Analytical solutions – Method of Characteristics

R: right-running wave slope $\frac{dx}{dt} = V + c$

Wave diagram



L: left-running wave slope $\frac{dx}{dt} = V - c$

P: particle-path slope $\frac{dx}{dt} = V$

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

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

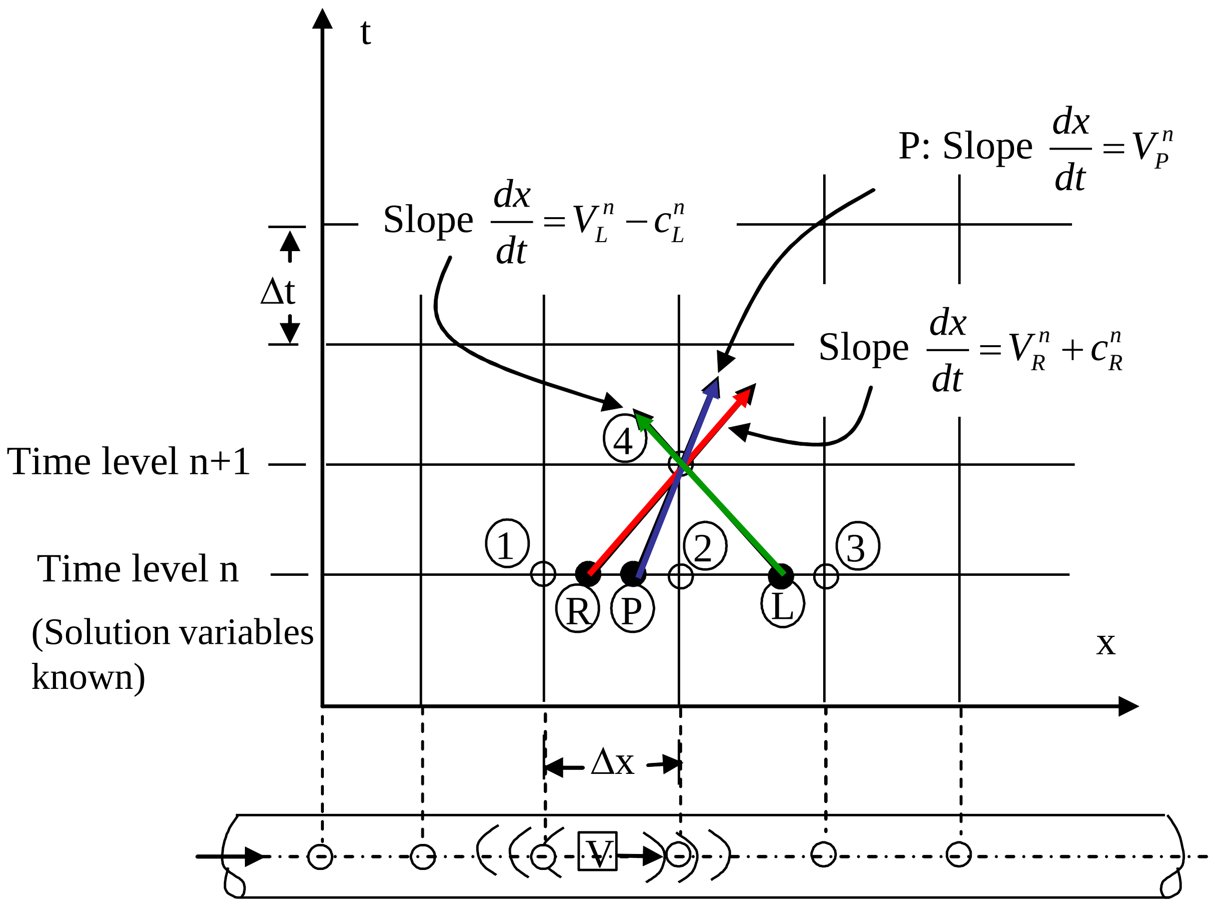
Moody, 1989

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

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

Moody, 1989
Anderson, 1990

Analytical solutions – Method of Characteristics

In the special case of isentropic flow, *F=G=H=0*, and P: equation is not needed

$$dP/\rho c \pm dV = 0$$: along R and L characteristic lines

Integrating gives

$$\int \frac{dP}{\rho c} \pm V = J_{R,L} = \frac{2}{\gamma - 1} c \pm V$$

$$M = V/c$$

where $J_{R,L}$ are the Riemann Invariants

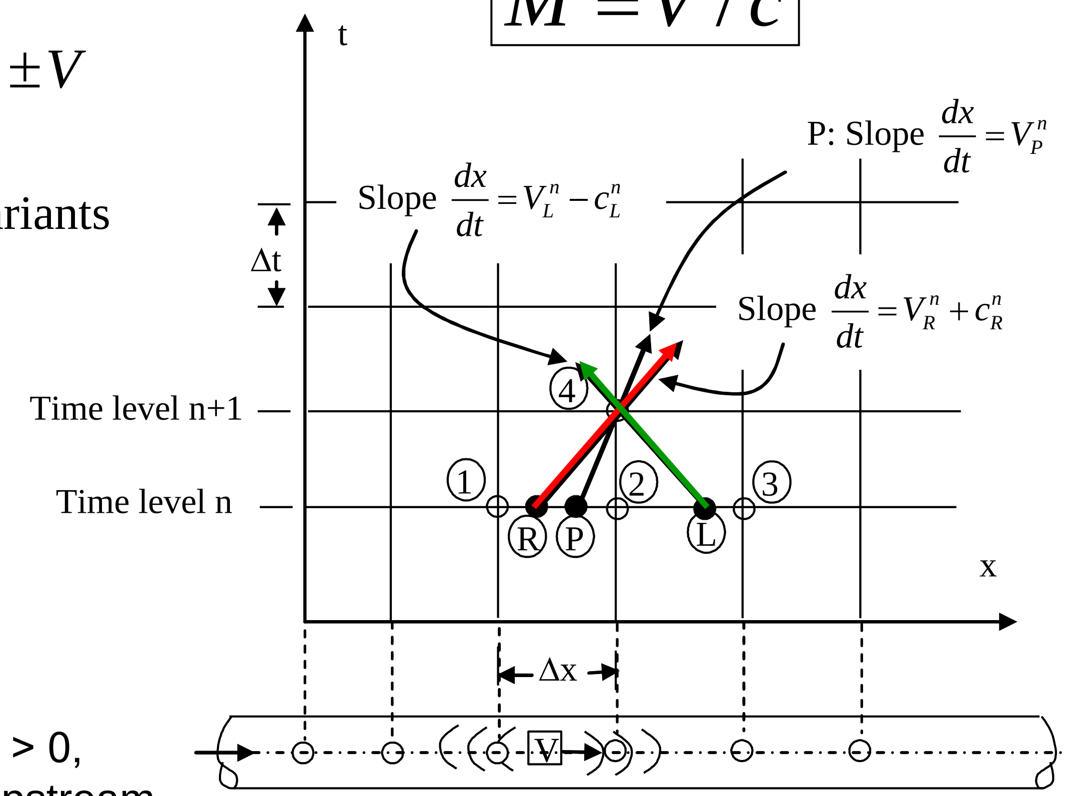
(2 equations in 2 unknowns)

or, along R:

$$V + \frac{2}{\gamma - 1} c = J_R$$

and along L:

$$V - \frac{2}{\gamma - 1} c = J_L$$



When V>c "left-running" wave's slope > 0,
and information does not propagate upstream

Moody, 1989
Anderson, 1990

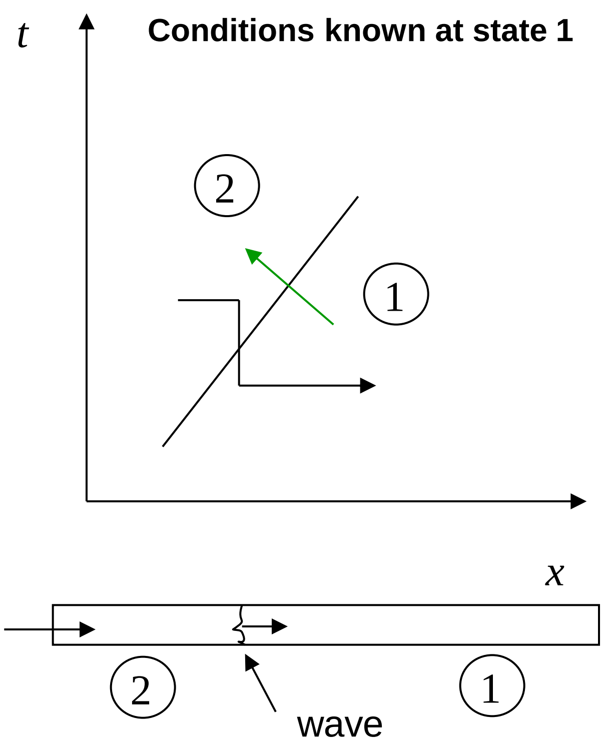
Analytical solutions – Method of Characteristics

Example: A weak wave with pressure ratio P_2/P_1=1.25 propagates down a tube filled with air at rest with T_1= 500K and P_1=500 kPa.
Find the gas velocity behind the wave using MOC.

$$c_1 = \sqrt{\gamma R T_1} = \sqrt{1.4 \cdot 287 \cdot 500} = 448.2 m/s$$

→ For isentropic flow:

$$c_2 / c_1 = (P_2 / P_1)^{(\gamma-1)/2\gamma}$$



Along L: $V_1 - \frac{2}{\gamma-1} c_1 = J_L = V_2 - \frac{2}{\gamma-1} c_2$ (V_1 =0)

$$\therefore V_2 = 72.6 m/s$$

Thompson, 1972

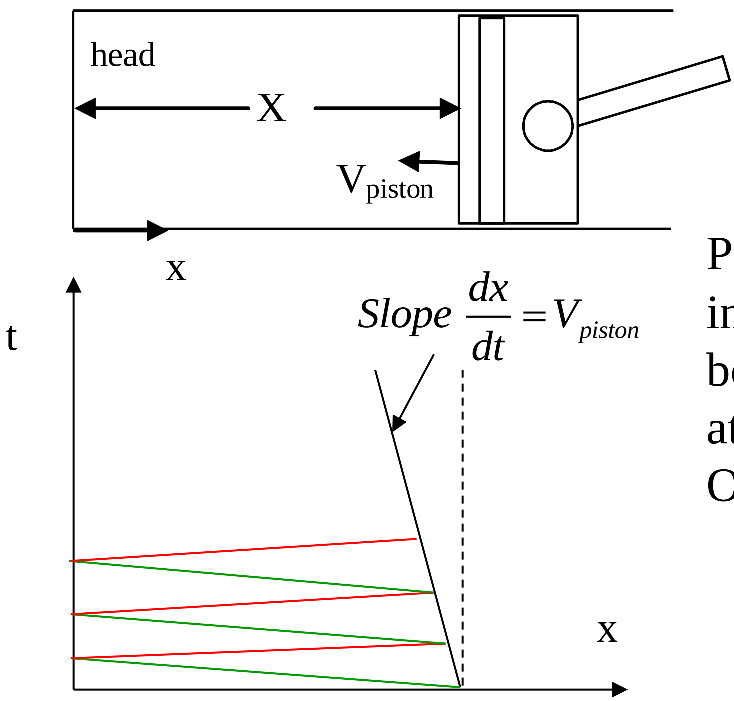
Lagrange ballistics

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.

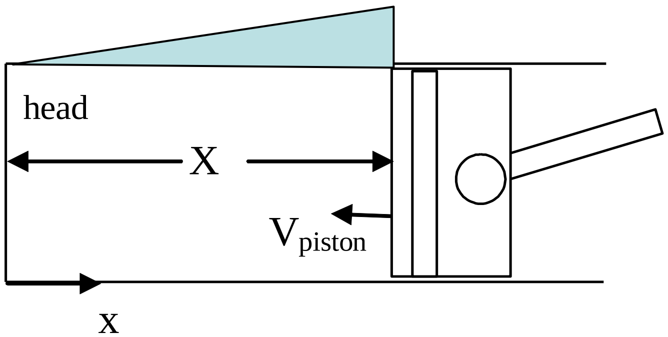
Order of magnitude analysis of **L:, R:,** and **P:** gives

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

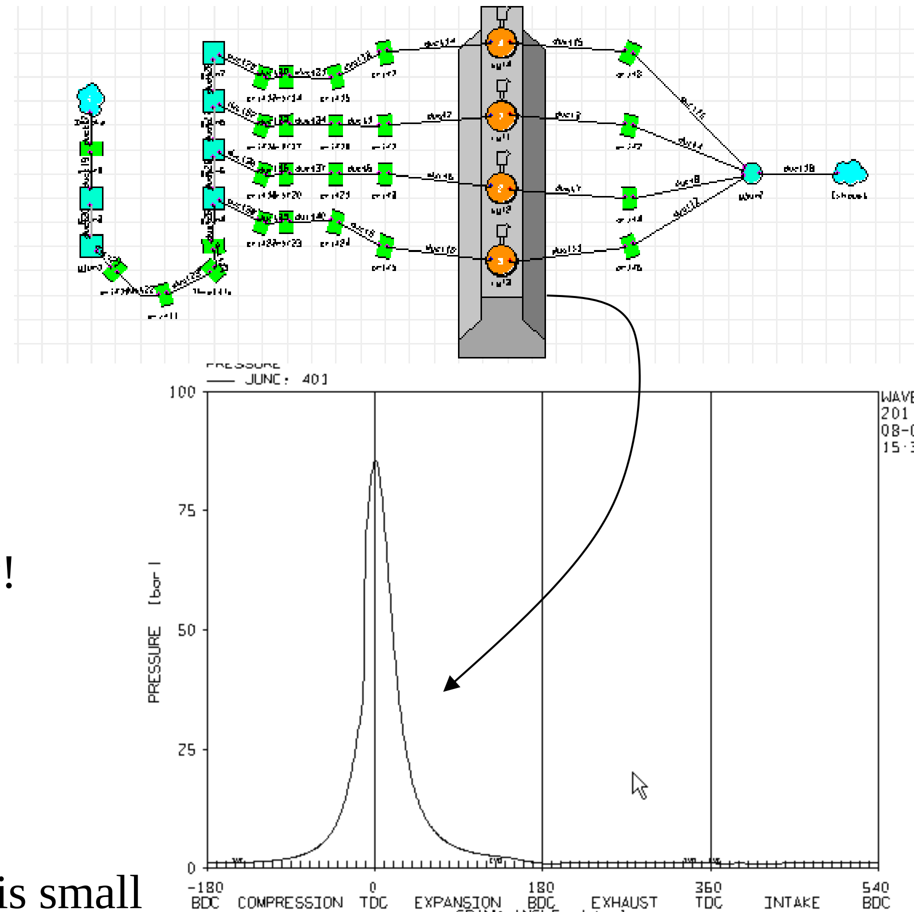
For $dV<<c$ relative density change is small– so density changes only in time

Thompson, 1972

Lagrange ballistics



$$\rho(x,t) \sim \rho(t)$$



Mass conservation shows that

$$\frac{d\rho}{dt} + V\frac{\partial\rho}{\partial x} + \rho\frac{\partial V}{\partial x} = 0$$

$$V = \frac{x}{X} V_{piston}$$ Gas stretches linearly!

Momentum conservation gives

$$P(x,t) = P(x=0,t) - \rho \ddot{X} x^2 / 2X$$

or $P=P(t)$ as long as piston acceleration $\ddot{X}$ is small

Location of pressure transducer unimportant

Gas exchange process

Gasoline engine intake system:

air filter, carburetor and throttle plate or port fuel injector, intake manifold, intake port, intake valves.

Supercharging – increases inducted air mass (in both gasoline and diesel engines).

Intake and exhaust manifold designed to maximize cylinder filling and scavenging.

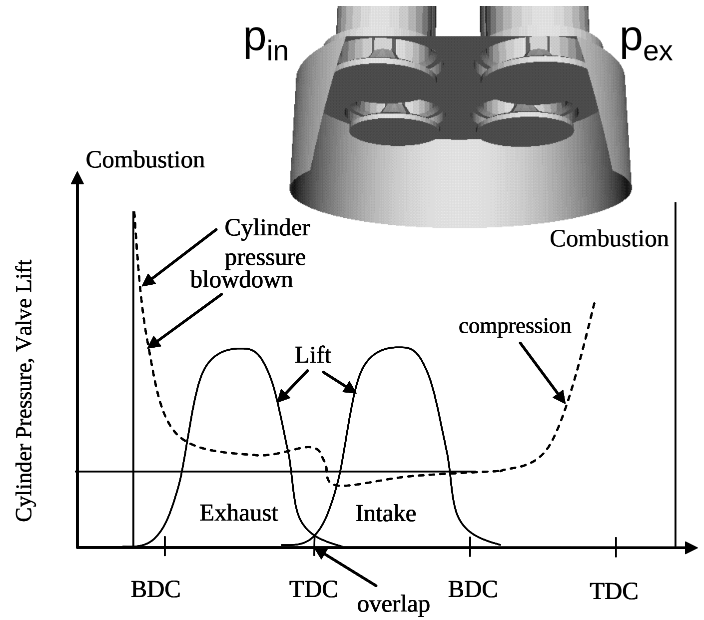
Intake system pressure drops (losses) occur due to quasi-steady effects (e.g., flow resistance), and unsteady effects (e.g., wave action in runners).

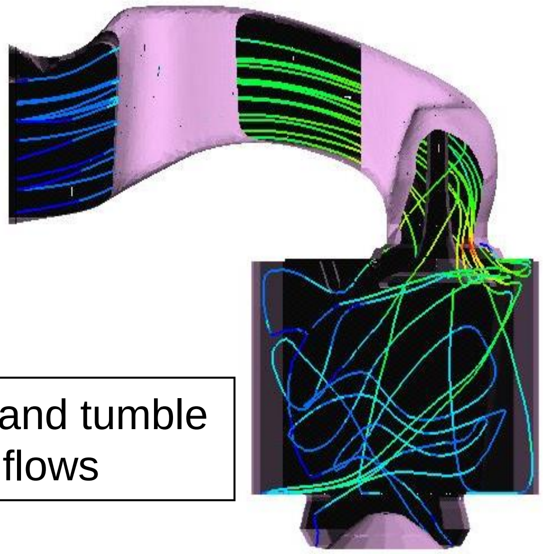
Engine breathing affected by intake/exhaust valve lifts and open areas (most of the losses).

Valve overlap can cause exhaust gases to flow back into intake system, or intake gases can enter the exhaust (depending on p_{in}/ p_{ex})

Intake also generates large scale flow structures that can be used to promote turbulent mixing

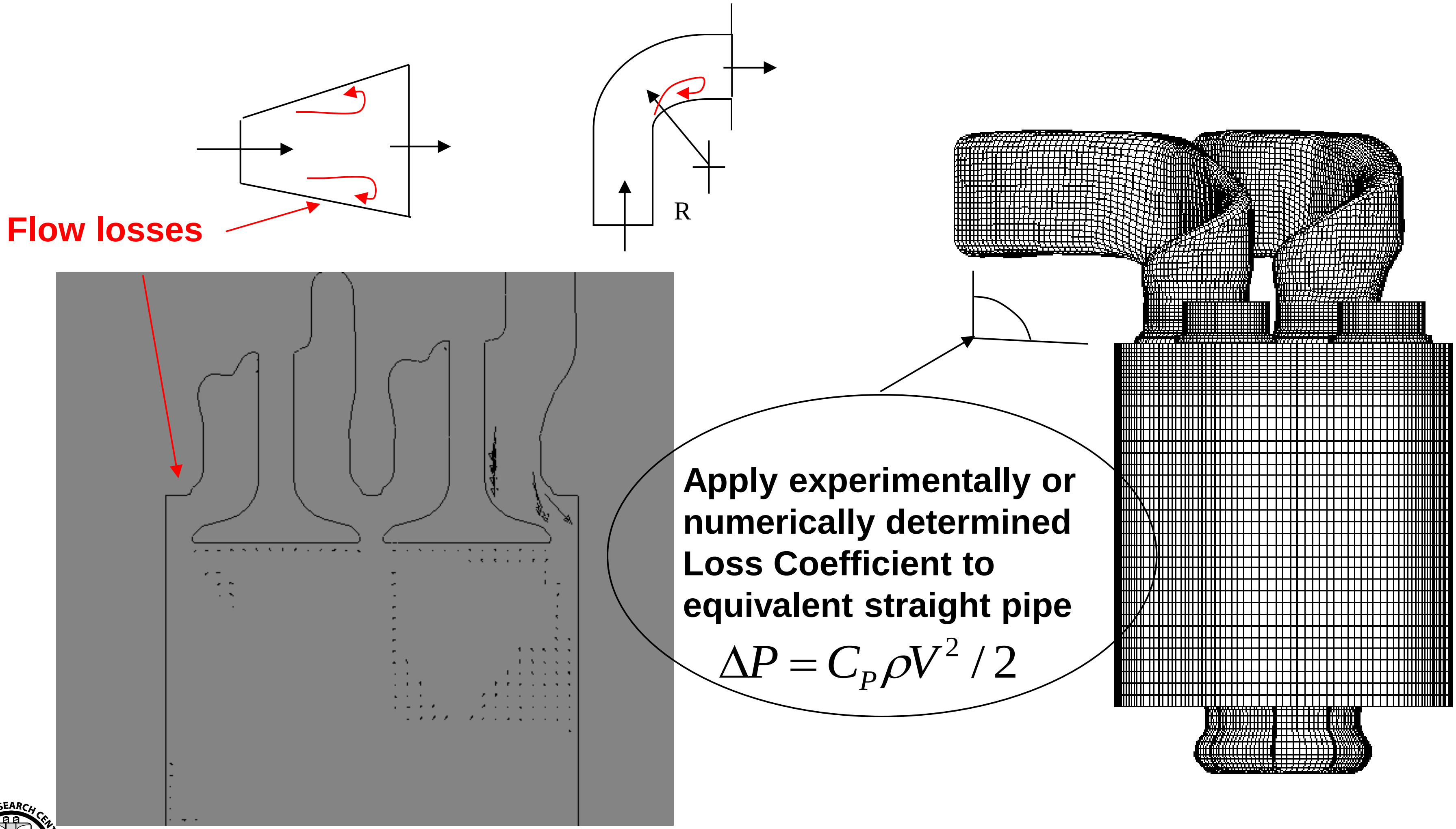
- requires 3-D CFD modeling





Swirl and tumble flows

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



Ricardo WAVE friction models

WAVE Knowledge Center v5.2

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ports where ... should be set to 0.0 because wall friction is already included in the valve flow coefficients.

2.3.5 Duct Bend Losses

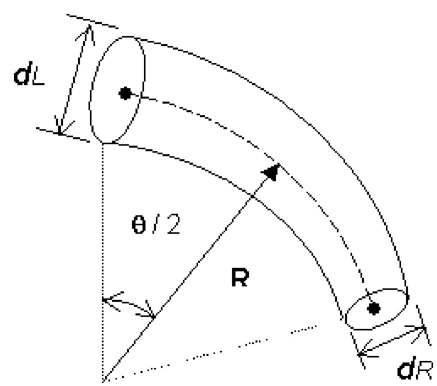
Pressure loss coefficients due to bends can usually be taken from handbook values. However, the handbook loss measurements must be adjusted to subtract friction losses due to bend length, which WAVE calculates separately. This is especially true for large-radius bends, which are long and thus can have a large friction loss[1].

Experience seems to indicate that bends in intake runners for engines require higher Cp values than those given by handbooks. This may be due to the highly pulsing nature of the flow. Tables of loss coefficients always give steady flow values. There are not many authoritative sources for losses in pulsing flow.

A distributed pressure loss can be imposed in a duct via the Cp parameter in the DUC:DUCT block where

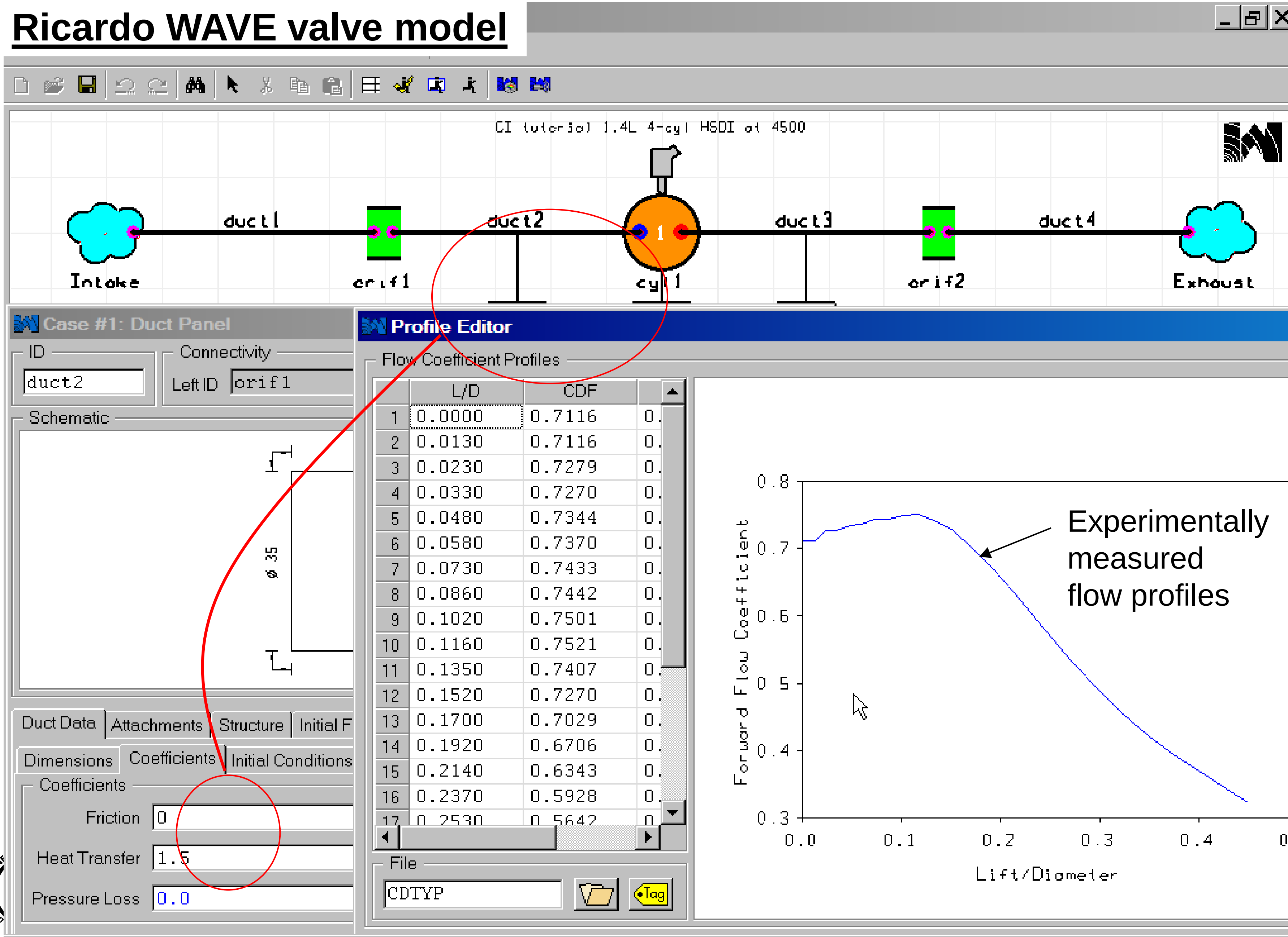
$$Cp = \Delta p \Big/ \left(\frac{1}{2}\right) \rho v^2$$

To simplify the setup of a model, the Cp value for a curved duct can be calculated by specifying the bend angle in the DUC:BENDS block. The bend loss coefficient is calculated as follows:

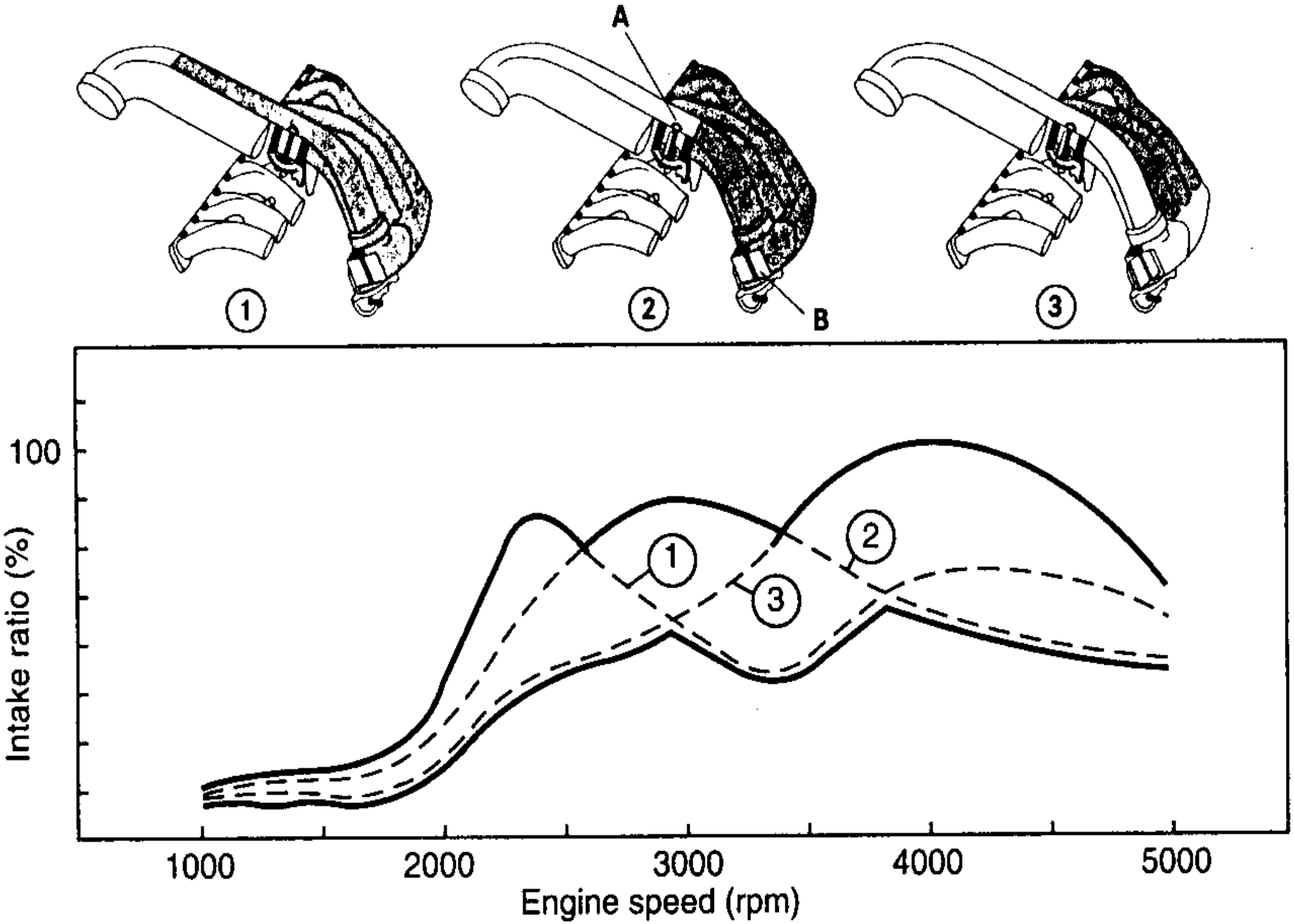


http://www.ricardo.com/en-GB/What-we-do/Software/Products/WAVE/

Ricardo WAVE valve model



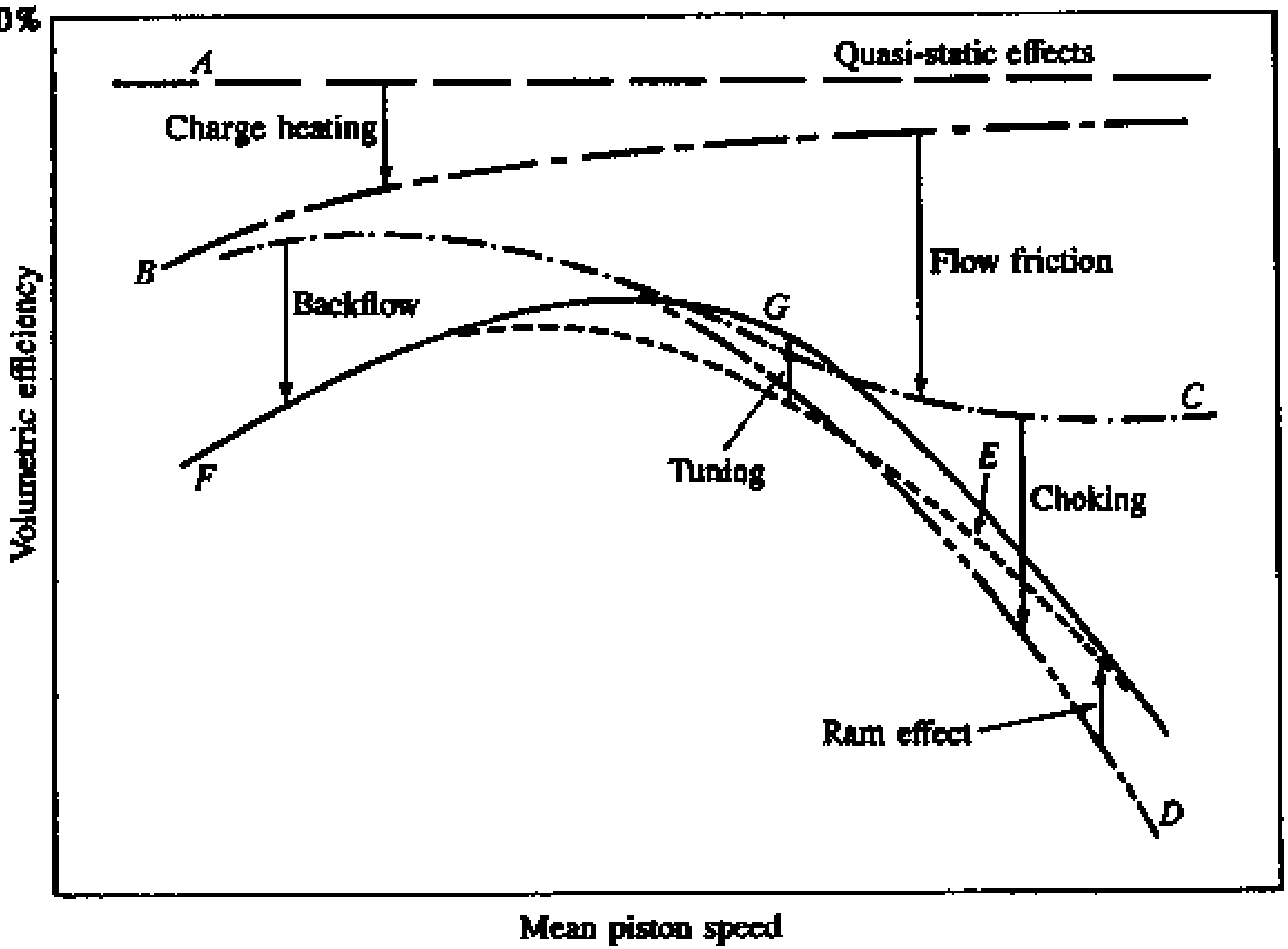
Optimization: Volumetric efficiency



Mercedes-Benz three stage resonance intake system

Heywood, Fig. 6.9

Volumetric efficiency parameters (SI engine < CI engine)



A → B → C → D → E → F → G

Losses in Carburetor, Intake manifold heating (rho), Fuel vapor displaces air

MAP Pin~Pex in diesel

Lower CR - SI more residual

Diesel - more residual is air

$$\eta_v = \frac{m_{air}}{\rho_{a,i} V_d}$$

Volumetric efficiency

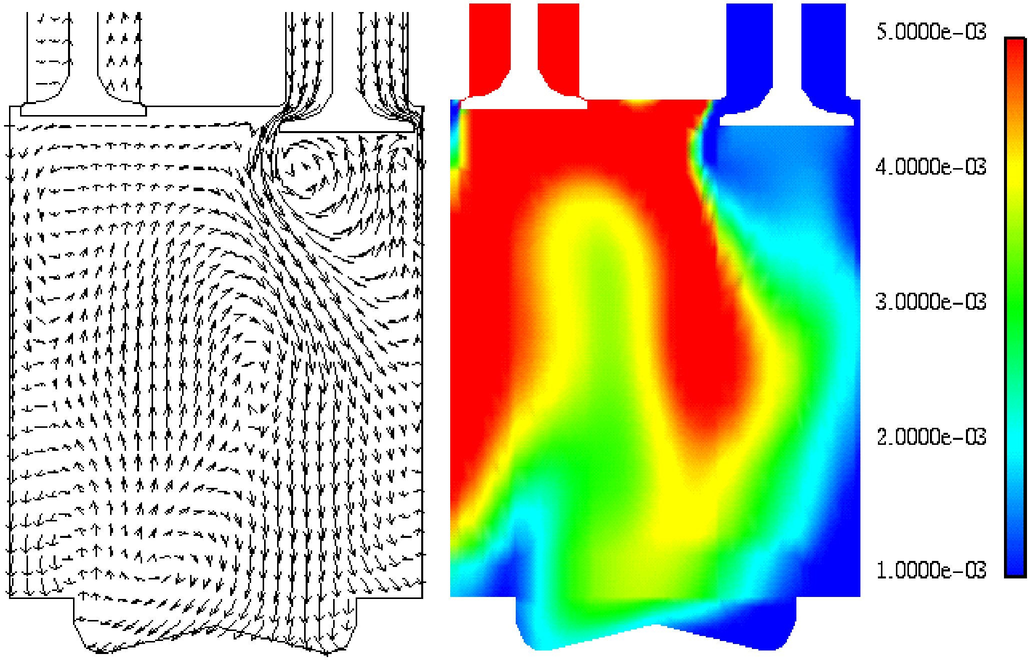
m_{air} is the mass of air trapped in the cylinder at intake valve closure (IVC),

$\rho_{a,i}$ is the intake air density, and

V_d is the volume displaced by the piston.

Accurate descriptions of valve flow losses require consideration of multi-dimensional flow separation phenomena and their effect initial conditions at intake valve closure (IVC).

Highest mixing of incoming fresh charge and combustion products occurs when intake flow velocities are largest due to high flow turbulence (half-way through stroke).



CFD flow velocity and residual gas distribution during gas exchange in plane of valves (intake valves about to close 144 degrees ATDC - 1600 rev/min)

Part 1: IC Engine Review, 0, 1 and 3-D modeling

Brief history of engine CFD

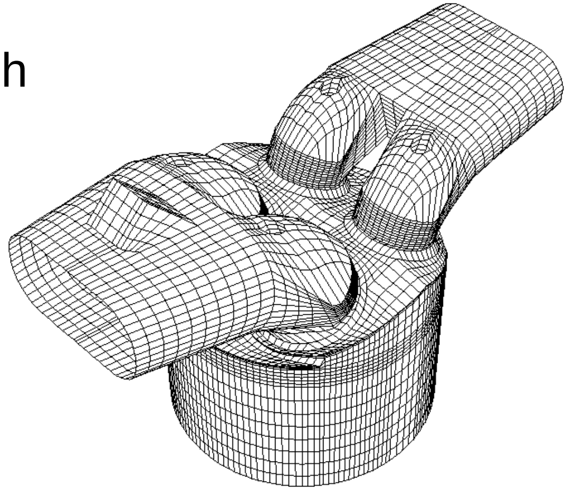
Arab oil crisis ~ 1973: US DOE

- Open source codes
  - Los Alamos National Lab, Princeton Univ., UW-ERC
  - 1970's – RICE → REC → APACHE → CONCHAS
  - 1980's – CONCHAS-SPRAY → KIVA family
  - 1985 – KIVA ;1989 – KIVA-II; 1993 – KIVA-3;
  - 1997 – KIVA-3V; 1999 – KIVA-3V Release 2; 2006 - KIVA-4
  - 2004 – OpenFOAM (2011 SGI)
- Commercial codes
  - 1980's Imperial College & others
  - Computational Dynamics, Ltd. → commercialize: STAR-CD
  - 1990's—other commercial codes: AVL FIRE, Ricardo VECTIS
  - 2005– FLUENT (with moving piston and in-cylinder models)
  - 2010 – CONVERGE (CSI), FORTE (Reaction Design)…..

Annual IMEM-User group meeting: Cray/UW-ERC/Iowa State
SAE Congress Multidimensional Modeling Session…..

Amsden, 1989, 1997

3-Dimensional models



Solve conservation equations on (moving) numerical mesh

Mass

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = \dot{\rho}^s$$

spray source terms

Species

$$\frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m \mathbf{u}) = \nabla \cdot \left[\rho D \nabla \left(\frac{\rho_m}{\rho} \right) \right] + \dot{\rho}_m^c + \dot{\rho}_m^s$$

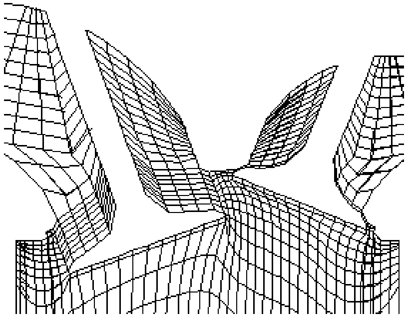
combustion source terms

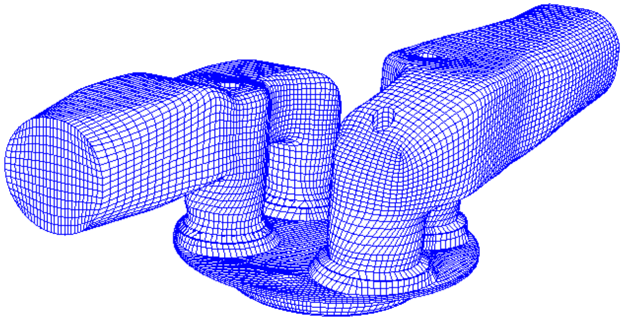
Momentum

$$\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = \rho \mathbf{g} + \mathbf{F}^s - \nabla p + \nabla \cdot \bar{\sigma}$$

Energy

$$\frac{\partial (\rho I)}{\partial t} + \nabla \cdot (\rho \mathbf{u} I) = -\nabla \cdot \mathbf{J} + \dot{Q}^c + \dot{Q}^s - p \nabla \cdot \mathbf{u} + \bar{\sigma} : \nabla \mathbf{u}$$

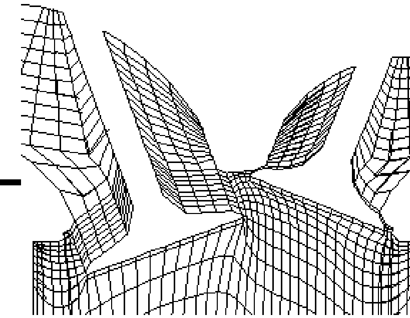




Amsden, 1989, 1997

KIVA-3V CFD code: flow solver

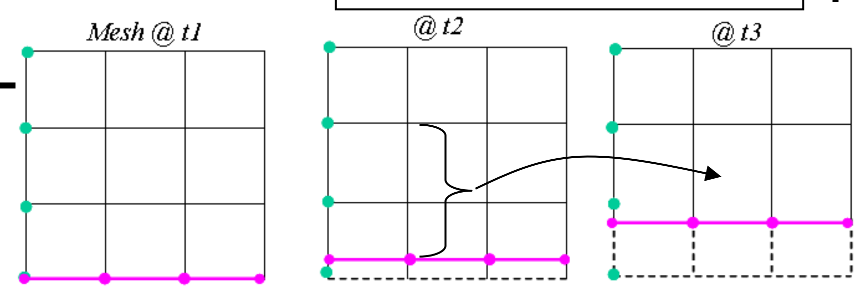
Main program and approximately 50 subroutines



Initialization	Read input data Calculate gas viscosity Initialize time step, piston velocity
Phase A	Spray modeling (injection, drop breakup, collision, evaporation…) Combustion chemistry Emission modeling Mass and energy contribution due to spray and combustion
Phase B	Fluid phase calculation Mass, momentum, velocity, temperature, pressure, turbulence properties (Implicit solver, iterations) Update droplet velocity
Phase C	Snapping/Rezoning grids Remapping fluid properties to new grids Update cell properties

Big Iteration

"Snapper"
add/delete grid cells

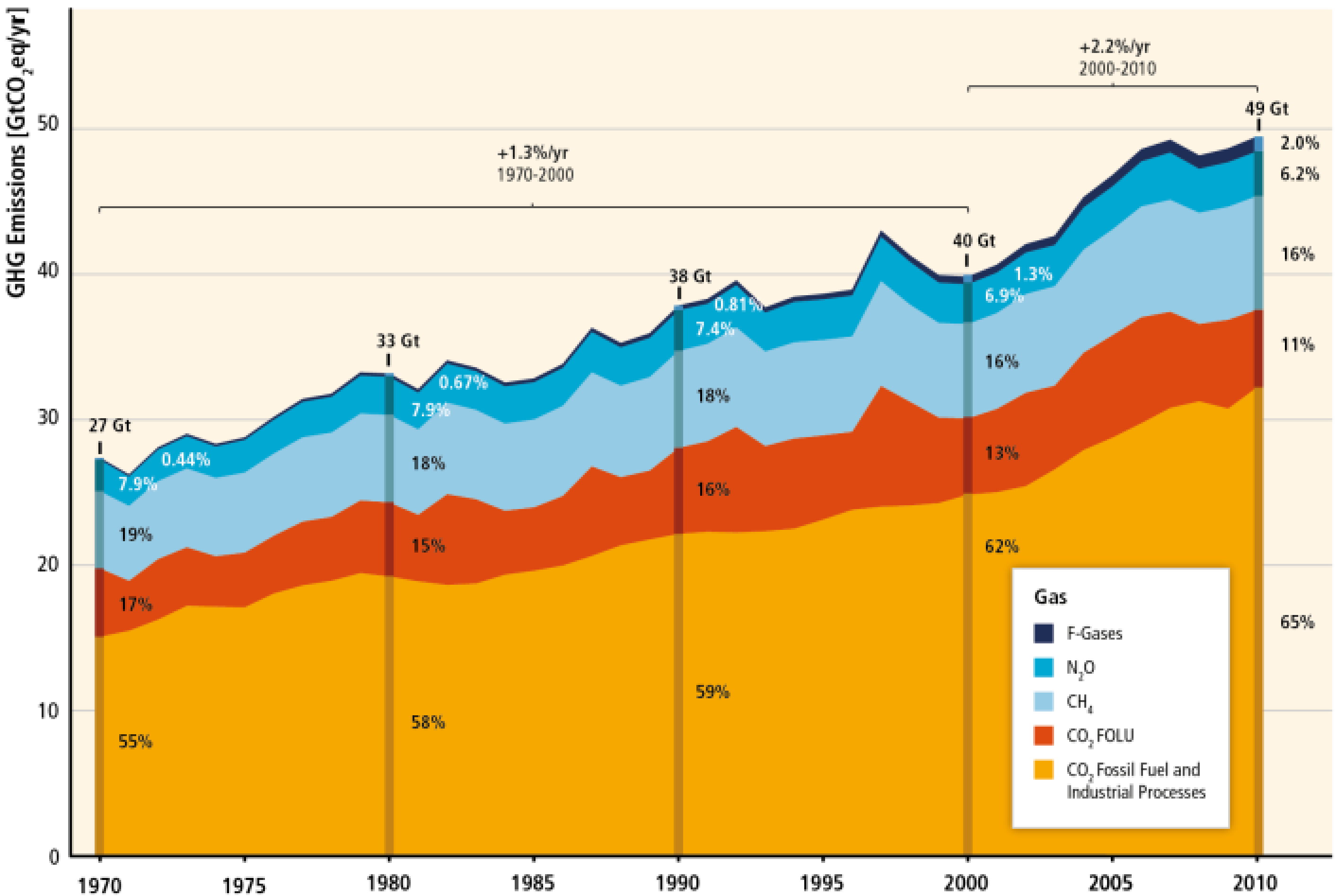


IPCC, 2014

Summary – CO_2 = fuel efficiency

~50% of cumulative anthropogenic CO_2 emissions between 1750 and 2010 have occurred in the last 40 years

Total Annual Anthropogenic GHG Emissions by Groups of Gases 1970-2010



transportation

Charmley, 2004

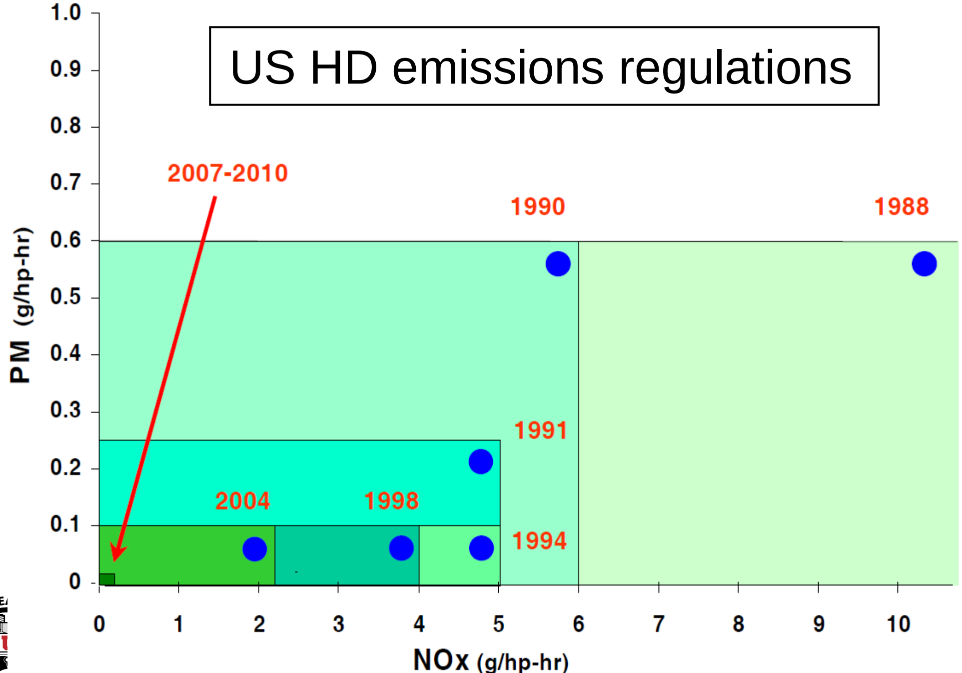
Summary

Transportation is ~1/3 of the total energy use in the US

Internal combustion engines are among the most efficient power plants known to man, but research is needed to improve them further

The industry faces significant challenges to meet emissions regulations, but great progress has been made in the last 20 years.

Modeling tools are available to help quantify engine performance and to provide directions for improved efficiency



Oil Consumption, 2010:	
US	21.1%
Total Europe & Eurasia	22.9%
China	10.6%