INTERNAL COMBUSTION ENGINE

Chapter 5

Engine performance

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3.3 COMBUSTION WITH AIR

The general formula for the fuel used in IC engines can be taken as $C_mH_nO_p$, where m, n, and p represent the number of moles of carbon, hydrogen and oxygen atoms in a mole of fuel.

The basic stoichiometric (chemically correct) equation for fuel-air reaction is

$$C_mH_nO_p + Y_{cc}O_2 + 3.76Y_{cc}N_2 \rightarrow mCO_2 + \frac{n}{2}H_2O + 3.76Y_{cc}N_2 \tag{3.1}$$

where Y_{cc} is the chemically correct moles of O_2 per mole of fuel. The N_2 does not take part in the reaction. There are 3.76 moles of N_2 per mole of O_2 and since Y_{cc} moles of O_2 are theoretically necessary for the oxidation of the fuel, $3.76Y_{cc}$ moles of N_2 are present.

By balancing the number of moles of O_2 on both sides of Eq. (3.1),

$$\frac{p}{2} + Y_{cc} = m + \frac{n}{4}$$

$$\therefore \qquad Y_{cc} = m + \frac{n}{4} - \frac{p}{2} \tag{3.2}$$

The stoichiometric equation now becomes

$$C_mH_nO_p + \left(m + \frac{n}{4} - \frac{p}{2}\right)O_2 + 3.76\left(m + \frac{n}{4} - \frac{p}{2}\right)N_2 \rightarrow mCO_2 + \frac{n}{2}H_2O + 3.76\left(m + \frac{n}{4} - \frac{p}{2}\right)N_2 \tag{3.3}$$

The stoichiometric equation for pure hydrocarbons (C_mH_n), where the oxygen atom is not present ($p = 0$) can be written as

$$C_mH_n + \left(m + \frac{n}{4}\right)O_2 + 3.76\left(m + \frac{n}{4}\right)N_2 \rightarrow mCO_2 + \frac{n}{2}H_2O + 3.76\left(m + \frac{n}{4}\right)N_2 \tag{3.4}$$

The combustion equation for octane (C_8H_{18}) can be written with $m = 8$ and $n = 18$ as

$$C_8H_{18} + 12.5O_2 + 3.76(12.5)N_2 \rightarrow 8CO_2 + 9H_2O + 3.76(12.5)N_2$$

or

$$C_8H_{18} + 12.5O_2 + 47N_2 \rightarrow 8CO_2 + 9H_2O + 47N_2 \tag{3.5}$$

Power and torque

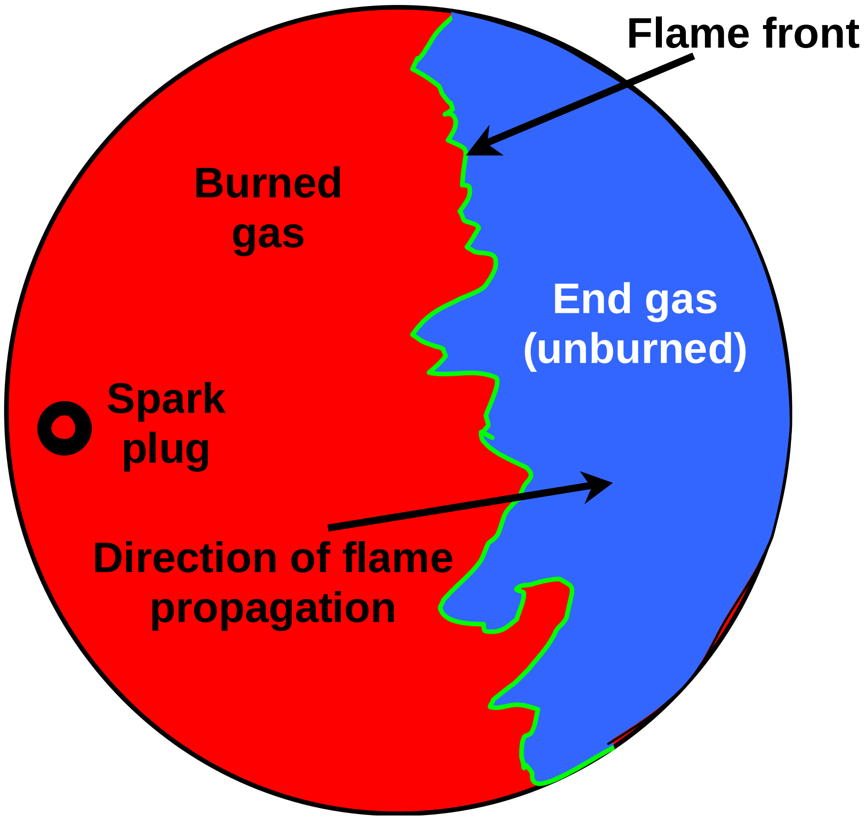
- Engine performance is specified in both in terms of power and engine torque - which is more important?
  - Wheel torque = engine torque x gear ratio tells you whether you can climb the hill
  - Gear ratio in transmission typically 3:1 or 4:1 in 1st gear, 1:1 in highest gear; gear ratio in differential typically 3:1
    - Ratio of engine revolutions to wheel revolutions varies from 12:1 in lowest gear to 3:1 in highest gear
  - Power tells you how fast you can climb the hill
  - Torque can be increased by transmission (e.g. 2:1 gear ratio ideally multiplies torque by 2)

$$\text{P (in horsepower)} \equiv \frac{\text{N (revolutions per minute, RPM) x Torque (in foot pounds)}}{5252}$$

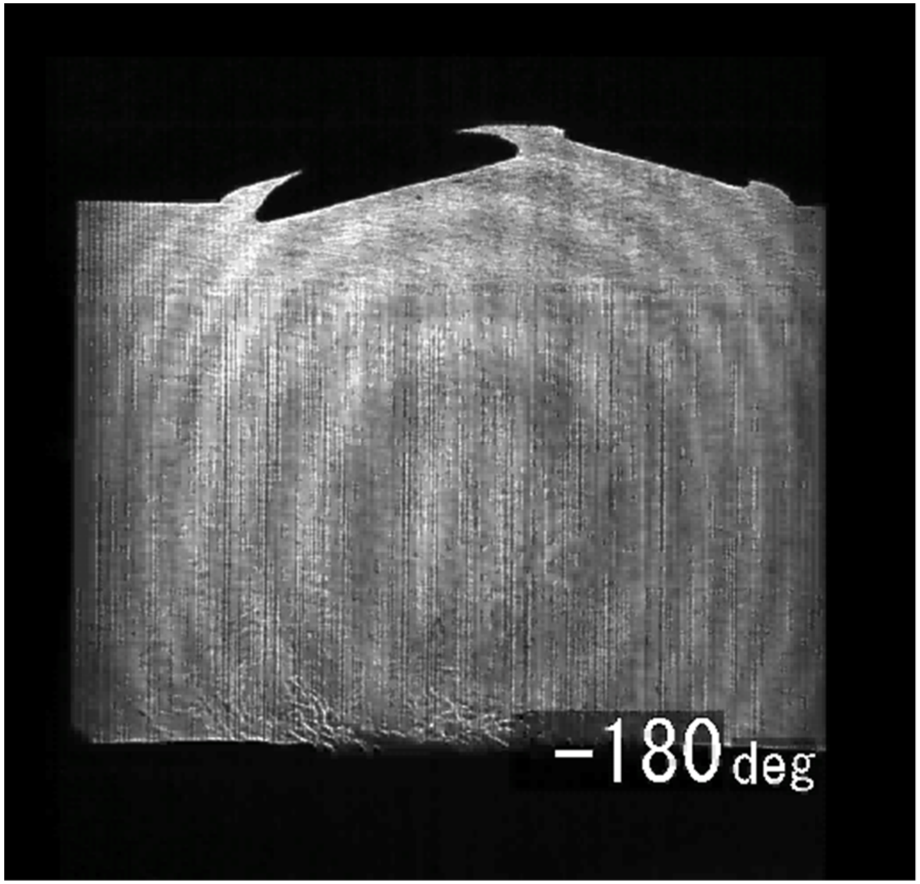
  - Power can't be increased by transmission; in fact because of friction and other losses, power will decrease in transmission
  - Power tells how fast you can accelerate or how fast you can climb a hill, but power to torque ratio ~ N tells you what gear ratios you'll need to do the job

Knock - what is it?

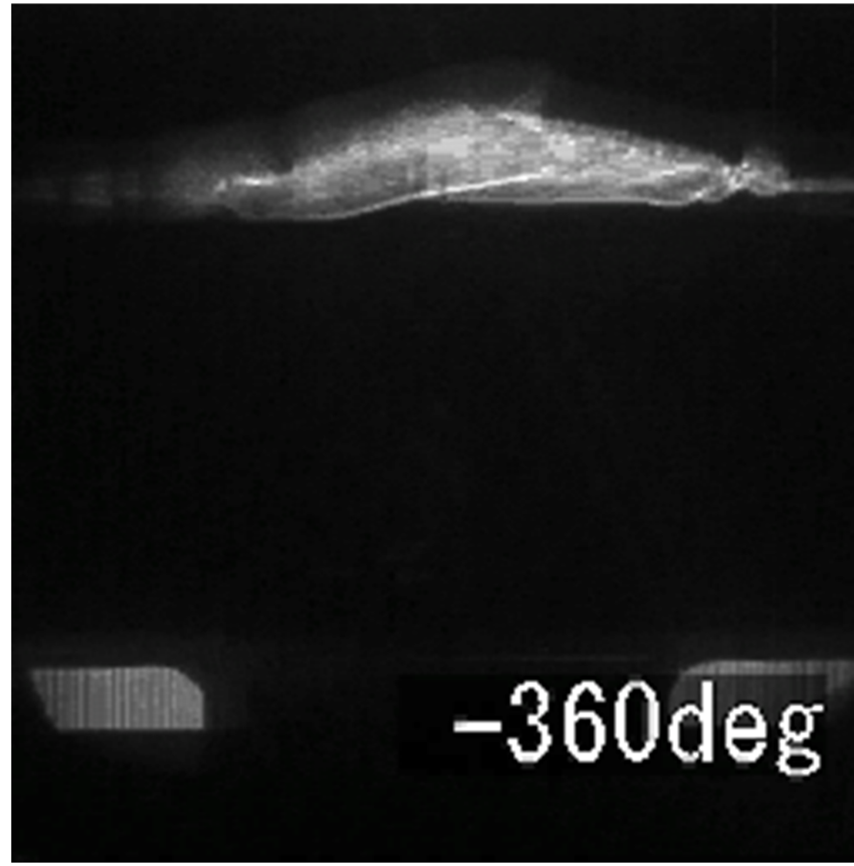
- Occurs when the combination of piston compression + “flame compression” increases temperature and pressure of the end gas until a very rapid explosion
- Engine combustion is always “horse race” between flame propagation (good horse) and knock (bad horse)



Knock - movies



No knock

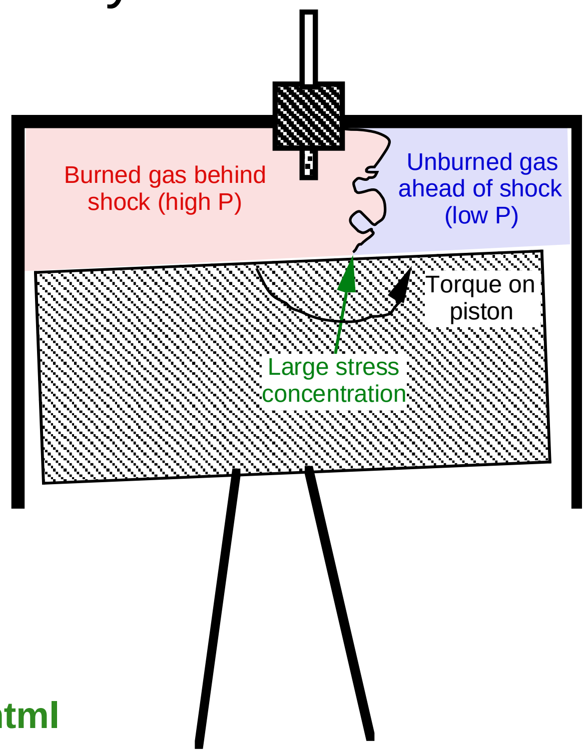


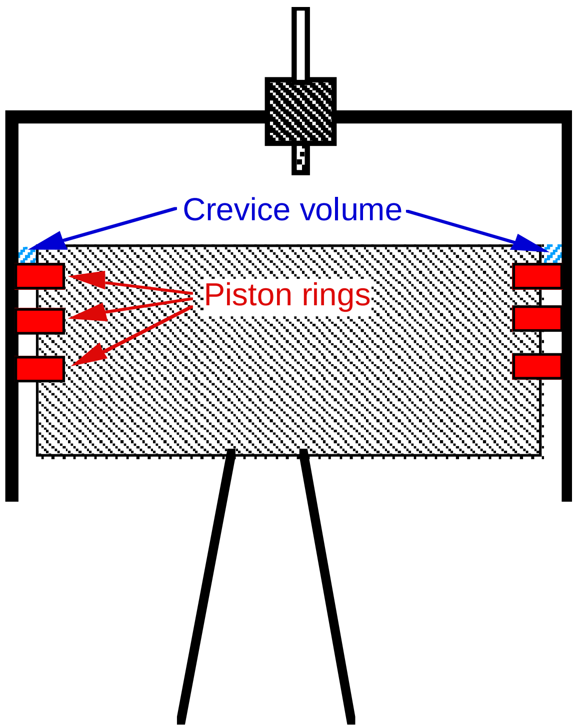
Knock

Videos courtesy Prof. Yuji Ikeda, Kobe University

Knock - why is it bad?

- Pressure gradients cause enormous stresses on the piston
- As the shocks propagate into the narrow region between the piston and cylinder wall (the "crevice volume"), the shock strength increases, causing locally even more severe damage

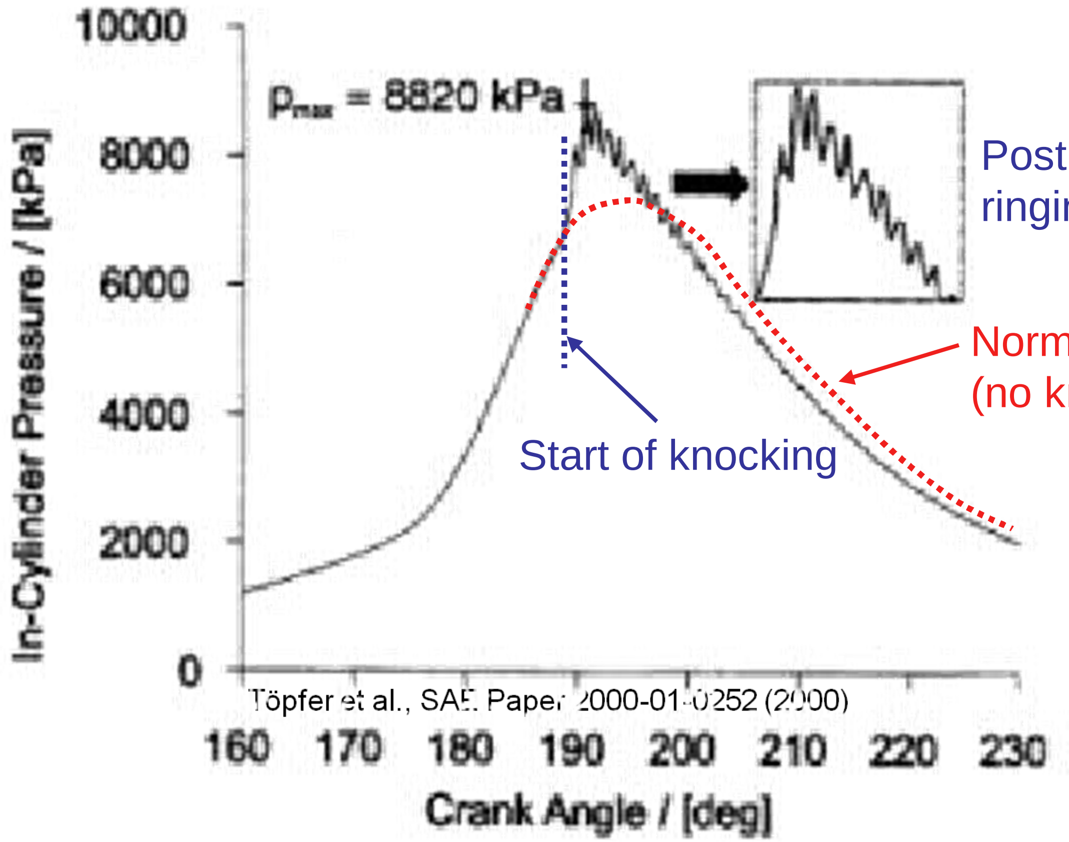




http://www.llnl.gov/str/Westbrook.html

Knock

- Shock formation causes “ringing” of pressure waves back & forth across cylinder - sounds like you're hitting piston with a hammer, which isn't too far from the truth



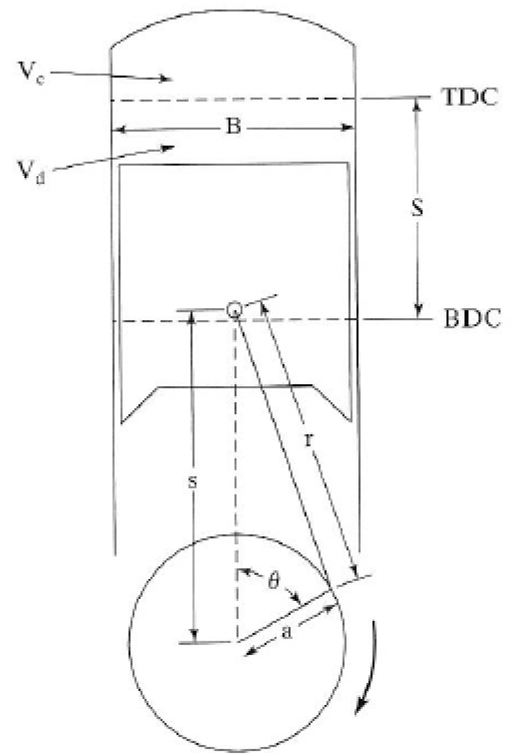
Töpfer et al., SAE Paper 2000-01-0252 (2000)

ANALYSIS OF ENGINE PERFORMANCE

How to achieve higher Performance?

Cylinder Geometry

- B = Bore
- S = Stroke
- s = Piston position
- r = Connecting rod length
- a = Crank radius
- θ = Crank angle,
- TDC = Top Dead Center, position of the crankshaft where $\theta = 0^{\circ}$
- BDC = Bottom Dead Center, position of the crankshaft where $\theta = 180^{\circ}$
- V_c= Clearance volume
- V_d= Displacement volume.



Engine Parameters

- Stroke $S = 2a$
- Average Piston Speed $\overline{U}_p = 2SN$ $N \rightarrow$ in rpm

$\overline{U}_p$ in the range of 5 ~ 20 m/s

Large diesel engines *High performance engines*

- Bore to Stroke Ratio = B/S
  - Small engines is usually from 0.8 ~ 1.2.
  - Square Engine → $B = S$
  - Under Square Engine → $B < S$
  - Over Square Engine → $B > S$

Engine Parameters

- Distance between the crank axis and wrist pin axis

$$s = a\cos\theta + \sqrt{r^2 - a^2\sin^2\theta}$$

$a =$ crankshaft offset

$r =$ connecting rod length

$\theta =$ crank angle

- When s is differentiated with time, the instantaneous piston speed U_p is:

$$U_p = ds/dt$$

- Ratio of instantaneous piston speed divided by the average piston speed cab be written as:

$$U_p/\bar{U}_p = (\pi/2)\sin\theta\left[1 + \left(\cos\theta/\sqrt{R^2 - \sin^2\theta}\right)\right]$$

where;

$$R = r/a$$

R values → 3 ~ 4 for small engines.
→ 5 ~ 10 for the largest engines.

Engine Parameters

- **Displacement, Displacement Volume** or **Swept Volume**, V_d, is the volume displaced by the piston as it travels from BDC to TDC.

$$V_d = V_{BDC} - V_{TDC}$$

- Displacement for the engine:

$$V_d = N_c(\pi/4)B^2\, S$$

$B =$ cylinder bore
$S =$ stroke
$N_c =$ number of engine cylinders

- Minimum cylinder volume occurs when the piston is at TDC. It is called the Clearance Volume, V_c.

$$V_c = V_{TDC}$$
$$V_{BDC} = V_c + V_d$$

Engine Parameters

- Compression Ratio, r_c , of the engine is defined as:

$$r_c = V_{BDC} / V_{TDC} = (V_c + V_d) / V_c$$

- The cylinder volume at any crank angle is:

$$V = V_c + (\pi B^2 / 4)(r + a - s)$$

- This can also be written in a non-dimensional form by dividing it with V_c :

$$V/V_c = 1 + \frac{1}{2}(r_c - 1)\left[R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta}\right]$$

Engine Parameters

- The cross-sectional area of a cylinder is:

$$A_p = (\pi/4)B^2$$

- The combustion chamber surface are is:

$$A = A_{ch} + A_p + \pi B(r + a - s)$$

A_{ch} → cylinder head surface area.

→Larger than A_p

- Then, can be rewritten as:

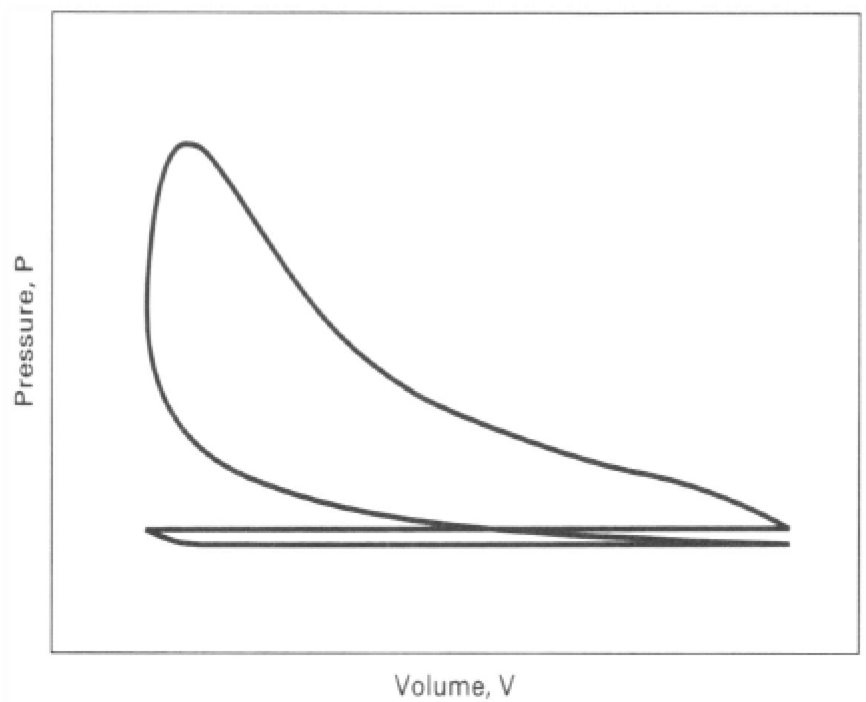
$$A = A_{ch} + A_p + (\pi B S/2)\left[R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta}\,\right]$$

- Combustion Chamber Work on Piston

$$W = \int F\, dx = \int P A_p\, dx$$

$$W = \int P\, dV$$

This is Indicated Work → Work done on Piston by Gases



Cylinder Pressure Curve → (P-V diagram)

Definitions

- Indicated Work, W_i :
  - Work done by Gases on Piston
  - Area in a P-V diagram

- Gross Indicated Work – $W_{i,\ gross}$
  - Work done during compression and expansion strokes

- Pumping Indicated Work – $W_{i,\ pump}$
  - Work done during exhaust and intake
  - Negative for Natural Aspirated Engines
  - Positive for Supercharged and Turbo Charged Engines

Definitions

- Net Indicated Work – $W_{i,\,net}$:

$$W_{i,net} = W_{i,gross} + W_{i,pump}$$

Definitions

- Friction Work (converted to heat) – W_f :
  - – Losses due to friction & parasitics:
  - – Friction losses due to Piston, Rods, Crankshaft
  - – Parasitic Losses required to drive:
    - Water Pump
    - Alternator
    - Air Conditioner
    - Cam Shaft

Definitions

- Brake Work, W_b :
  - Work available at Flywheel after friction losses:

$$\boxed{W_b = W_i - W_f}$$

Specific Work and Specific Volume

- *Work per unit mass of gas in Cylinder*

$$w = W / m \qquad v = V / m$$

$$w = \int P \, dv$$

m = mass of gases in cylinder

Mean Effective Pressure

- Definition:

$$mep = w / \Delta v$$

$$mep = W / V_d$$

$$\Delta v = v_{BDC} - v_{TDC}$$

$$V_d = V_{BDC} - V_{TDC}$$

Mean Effective Pressure

- Brake Mean Effective Pressure:

$$bmep = W_b / V_d = w_b / \Delta v$$

- Indicated Mean Effective Pressure:

$$imep = W_i / V_d = w_i / \Delta v$$

$$imep_{gross} = w_{i,gross} / \Delta v$$

$$imep_{net} = w_{i,net} / \Delta v$$

Mean Effective Pressure

- Pumping MEP:

$$pmep = w_{pump} / \Delta v$$

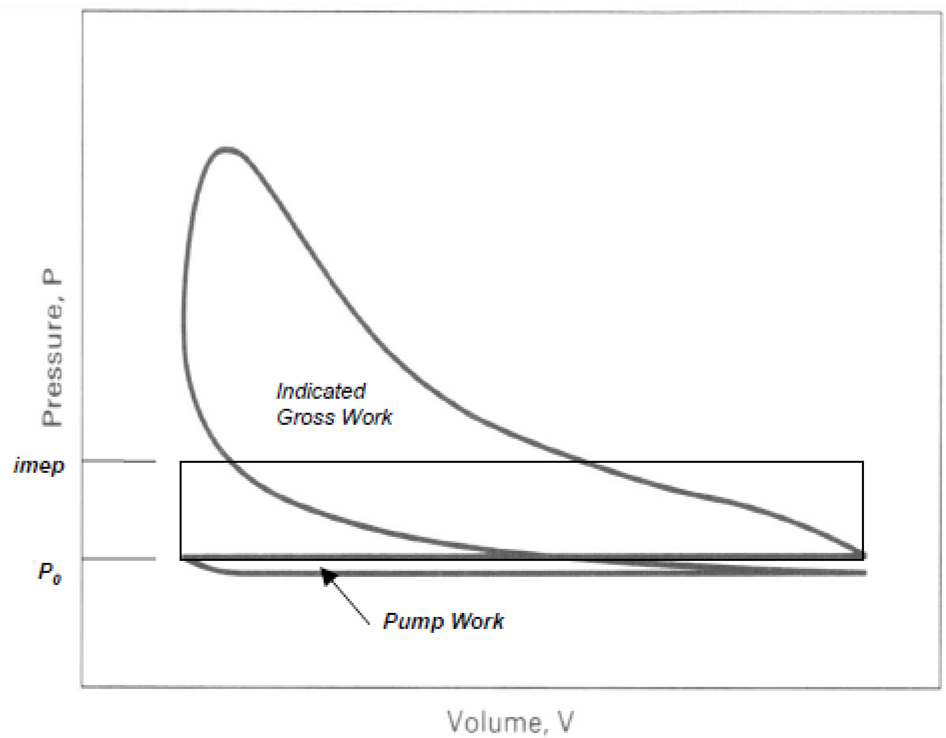
- Friction MEP:

$$fmep = w_f / \Delta v$$

mep Concept

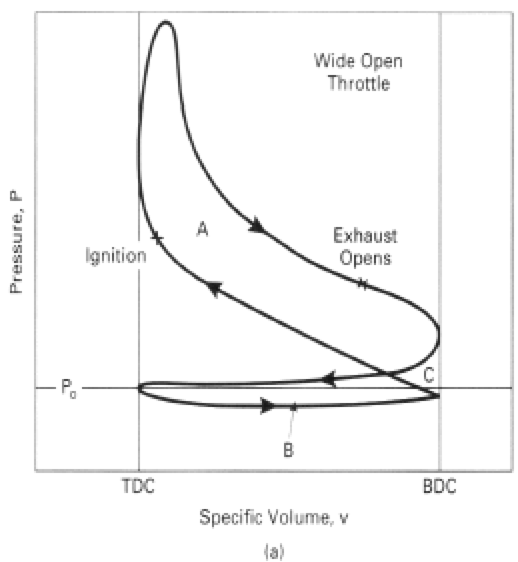
- *mep* is the average pressure when exerted over the stroke would produce the same work as the varying pressure.
- Example → Brake MEP (*bmep*):

$$bmep = W_b / V_d$$

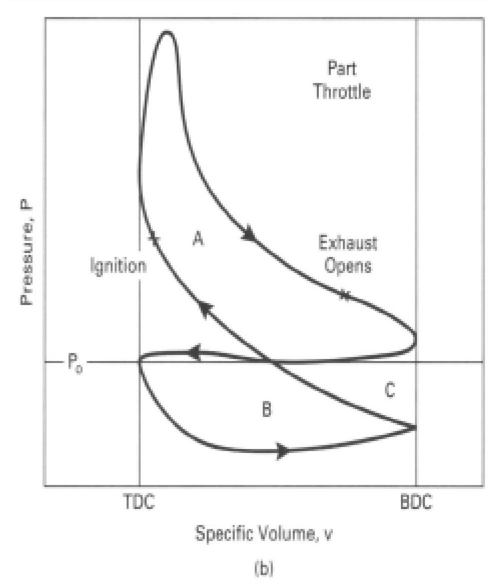

Pressure, P
imep
P_0
Indicated
Gross Work
Pump Work
Volume, V

Actual WOT vs. Part Throttle

Wide Open Throttle



Part Throttle



Mean Effective Pressure

Following from previous definitions:

$$imep_{net} = imep_{gross} + imep_{pump}$$

$$bmep = imep_{net} - fmep$$

$$bmep = \eta_m \, imep$$

$$bmep = imep_{net} - fmep$$

Mean Effective Pressure

- Typical Values (indicates “Loading”):
  - SI Engines at WOT:
    - bmep: 120 – 150 psia (850 – 1050 kPa)
  - CI Engines at WOT:
    - bmep: 100 – 130 psia (700 – 900 kPa)
  - Super &Turbo Charged Engines at WOT:
    - bmep: 145 – 175 psia (1000 – 1200 kPa)

Torque - τ

$$2\pi\tau = W_b = (bmep)V_d / n$$

n = number of revolutions per cycle

= 2 for 4 cycle

= 1 for 2 cycle

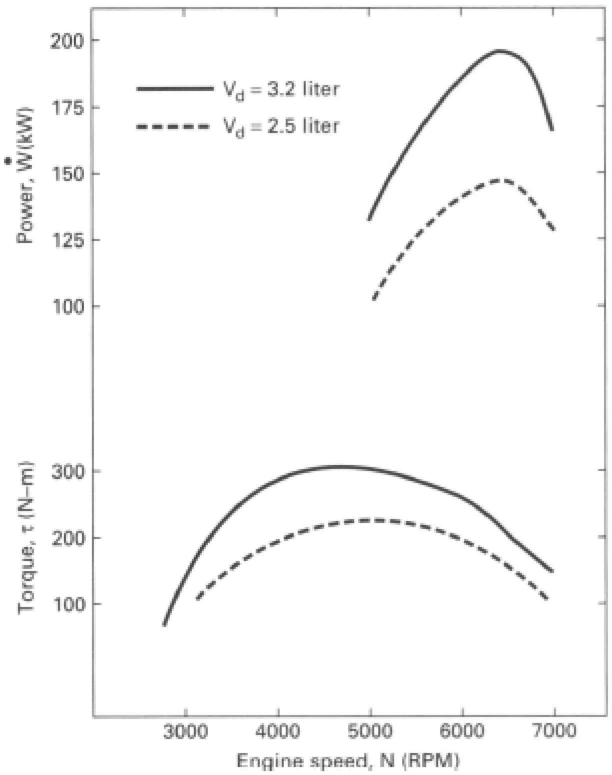
$$\tau = bmep\ V_d / 2\pi n$$

- **Power** - $\dot{W}$

$$\dot{W} = WN / n$$

$$\dot{W} = 2\pi N \tau$$

$$\dot{W} = (1/2n)(mep)A_p\overline{U}_p$$



200
175
150
125
100
Power, $\dot{W}$(kW)
V_d = 3.2 liter
V_d = 2.5 liter
300
200
100
Torque, τ (N–m)
3000
4000
5000
6000
7000
Engine speed, N (RPM)

- Following from definitions:

$$\left(\dot{W}_i\right)_{net} = \left(\dot{W}_i\right)_{gross} - \left(\dot{W}_i\right)_{pump}$$

$$\dot{W}_b = \dot{W}_{i,net} - \dot{W}_f$$

$$Units: \ 1 \ \ kW = 1.341 \ \ hp$$

- **Power Parameters:**

Specific Power: $SP = \dot{W}_b / A_p$

Output per Displacement: $OPD = \dot{W}_b / V_d$

Displacement per unit Power: $SV = V_d / \dot{W}_b$

Specific Weight: $SW = (engine \quad weight) / \dot{W}_b$

where: $\dot{W}_b$ = brake power

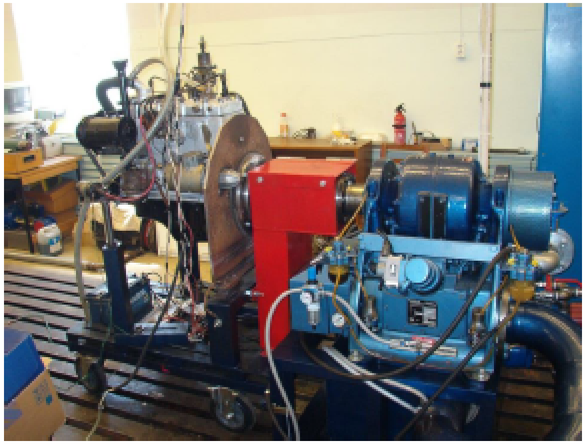
A_p = piston face area of all pistons

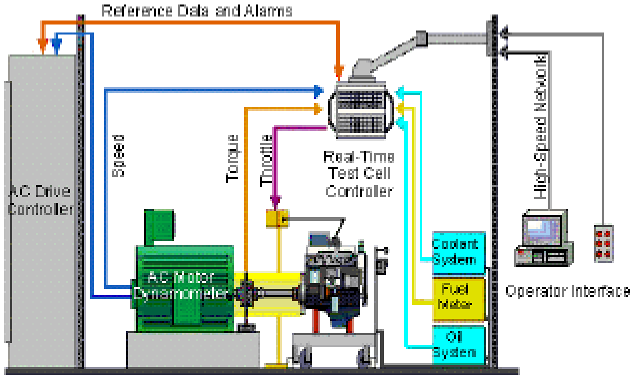
V_d = displacement volume

- Typical Values:
  - OPD: 0.5 – 2 hp/in3 (25 – 100 kW/L)
  - Racing Engines: up to 3 hp/in3
    - (150 kW/L)

Engine Dynamometers

- Dynamometers Measure Torque and Power
- Fluid Dynamometers
  - Water pump absorbing energy forces water through small orifices – Throttling Process
- Eddy Current Dynamometers
  - Generator with internal dissipation of internally generated currents
- Electric Dynamometer
  - Motor-Generator with accurate control of load or rpm and torque power required or absorbed





Reference Data and Alarms
AC Drive Controller
Speed
Torque
Throttle
Real-Time Test Cell Controller
High-Speed Network
AC Motor Dynamometer
Coolant System
Fuel Meter
Oil System
Operator Interface

Example 1

John's automobile has a 3 liter SI V6 engine that operates on a 4-stroke cycle at 3600 rpm. The compression ratio is 9.5, the length of the connecting rods is 16.6 cm, and the engine is square ($B=S$). At this speed, combustion ends at 20^0 ATDC. Calculate:

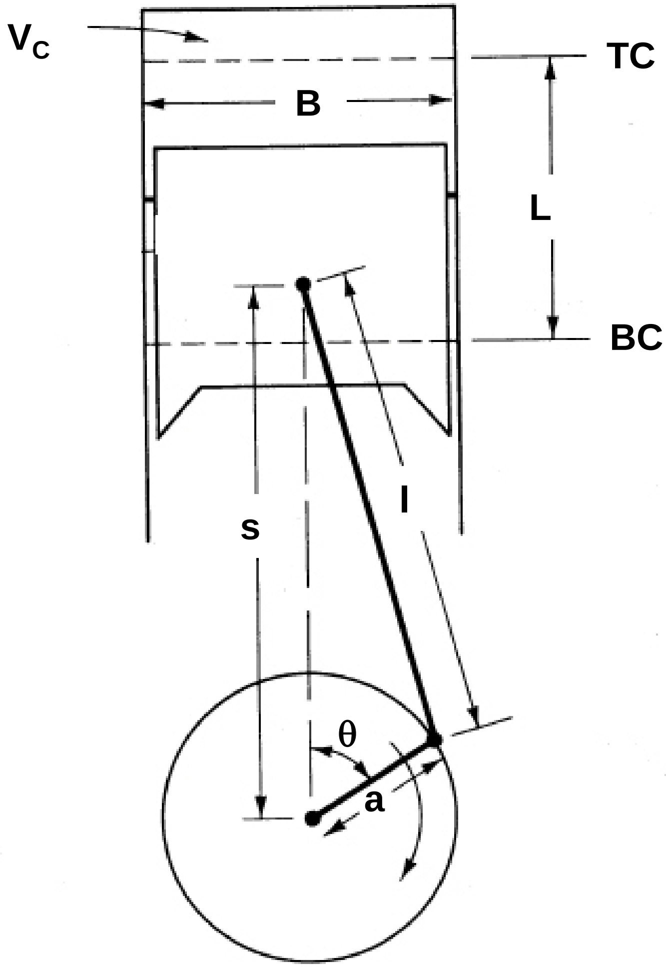
1. Cylinder bore and stroke length
2. Average piston speed
3. Clearance volume of one cylinder
4. Piston speed at the end of combustion
5. Distance the piston has traveled from TDC at the end of combustion
6. Volume in the combustion chamber at the end of combustion

Example 2

The engine in Example 1 is connected to a dynamometer which gives a brake output torque reading of 205 Nm at 3600 rpm. At this speed air enters the cylinders at 85 kPa and 60^0C, and mechanical efficiency of the engine is 85%. Calculate:

1. Brake power
2. Indicated power
3. *bmep*
4. *imep*
5. *fmep*
6. Power lost to friction
7. Brake power per unit mass of gas in the cylinder
8. Brake specific power
9. Brake output per displacement
10. Engine specific volume

Engine Geometry



$$s(\theta) = a\cos\theta + \left(l^2 - a^2\sin^2\theta\right)^{1/2}$$

Cylinder volume when piston at TC (*s*=*l*+*a*) defined as the clearance volume V_c

The cylinder volume at any crank angle is:

$$V(\theta) = V_c + \frac{\pi B^2}{4}(l + a - s(\theta))$$

Maximum displacement, or swept, volume:

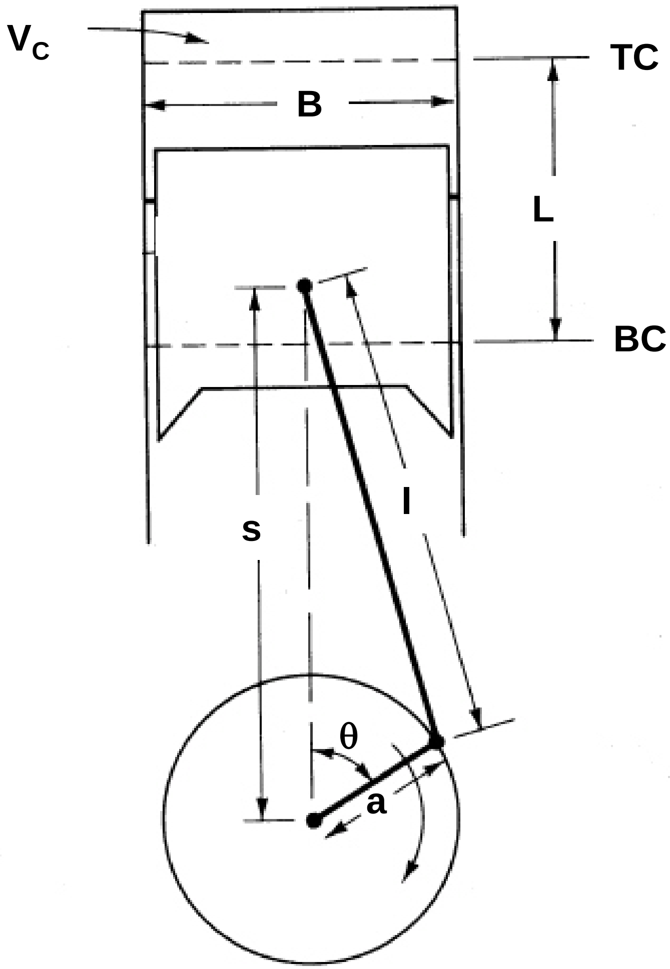
$$V_d = \frac{\pi B^2}{4} L$$

Compression ratio:

$$r_c = \frac{V_{BC}}{V_{TC}} = \frac{V_c + V_d}{V_c}$$

For most engines B ~ L (square engine)

Mean and Instantaneous Piston Speeds



$$s = a\cos\theta + \left(l^2 - a^2\sin^2\theta\right)^{1/2}$$

Average and **instantaneous** piston speeds are:

$$\bar{S}_p = 2LN$$

$$S_p = \frac{ds}{dt}$$

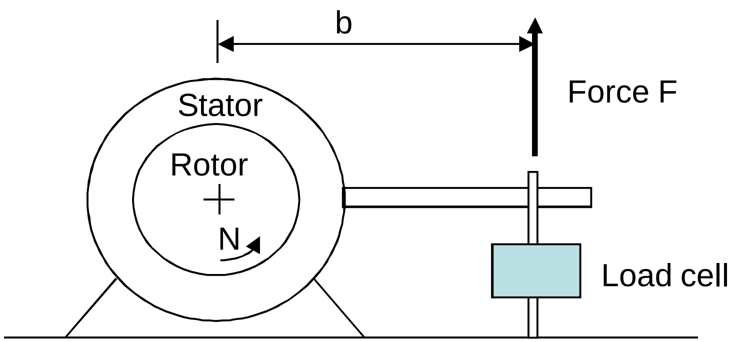
Where N is the rotational speed of the crank shaft in units revolutions per second

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\cos\theta}{\left((l/a)^2 - \sin^2\theta\right)^{1/2}}\right]$$

Average piston speed for a standard auto engine is ~15 m/s. Ultimately limited by material strength. Therefore engines with large strokes run at lower speeds those with small strokes can run at higher speeds.

Engine Torque and Power

Torque is measured using a dynamometer.



The **torque** exerted by the engine is: $T = F\,b$ *with units: J*

The **power** P delivered by the engine turning at a speed N and absorbed by the dynamometer is:

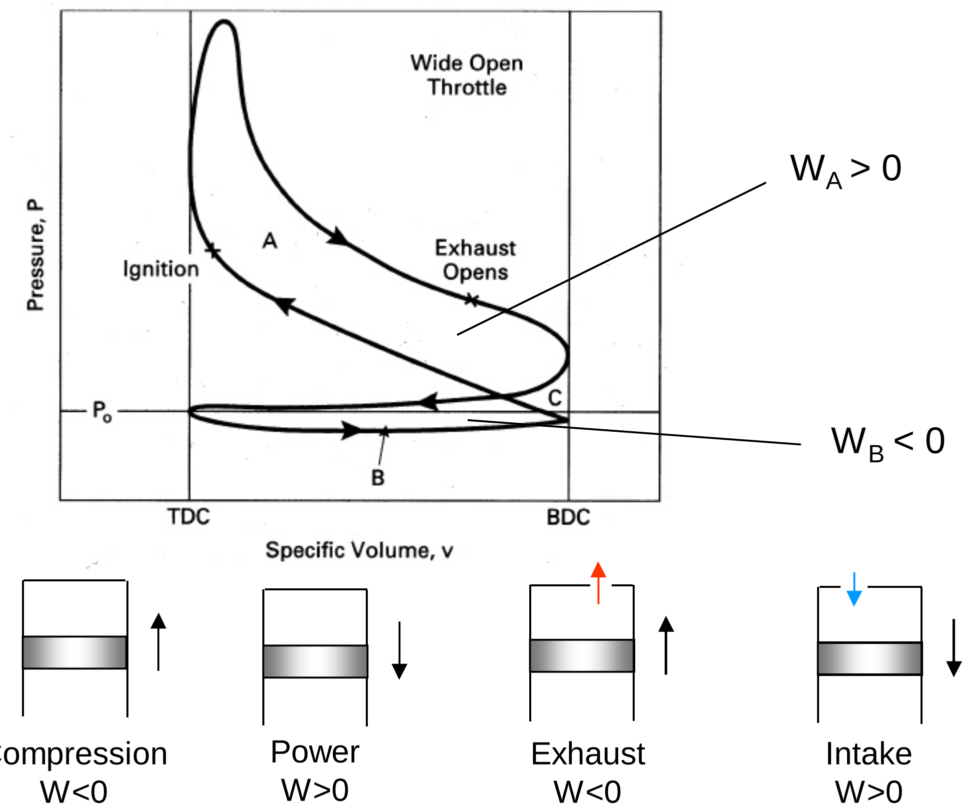
$$P = \omega T = (2\pi N)\,T \quad w/units: (rad/rev)(rev/s)(J) = Watt$$

Note: ω is the shaft angular velocity with units: rad/s

Indicated Work

Given the cylinder pressure data over the operating cycle of the engine one can calculate the work done by the gas on the piston.

The indicated work per cycle is $$W_i = \oint p dV$$



Indicated Power

$P_i = W_i \, N / n_R$ w/units: (kJ/cycle) (rev/s) / (rev/cycle)

where N – crankshaft speed in rev/s

n_R – number of crank revolutions per cycle

= 2 for 4-stroke

= 1 for 2-stroke

Power can be increased by increasing:

- the engine size, V_d
- compression ratio, r_c
- engine speed, N

Mechanical Efficiency

Some of the power generated in the cylinder is used to overcome engine friction. The **friction power** is used to describe these losses:

$$P_f = P_i - P_b$$

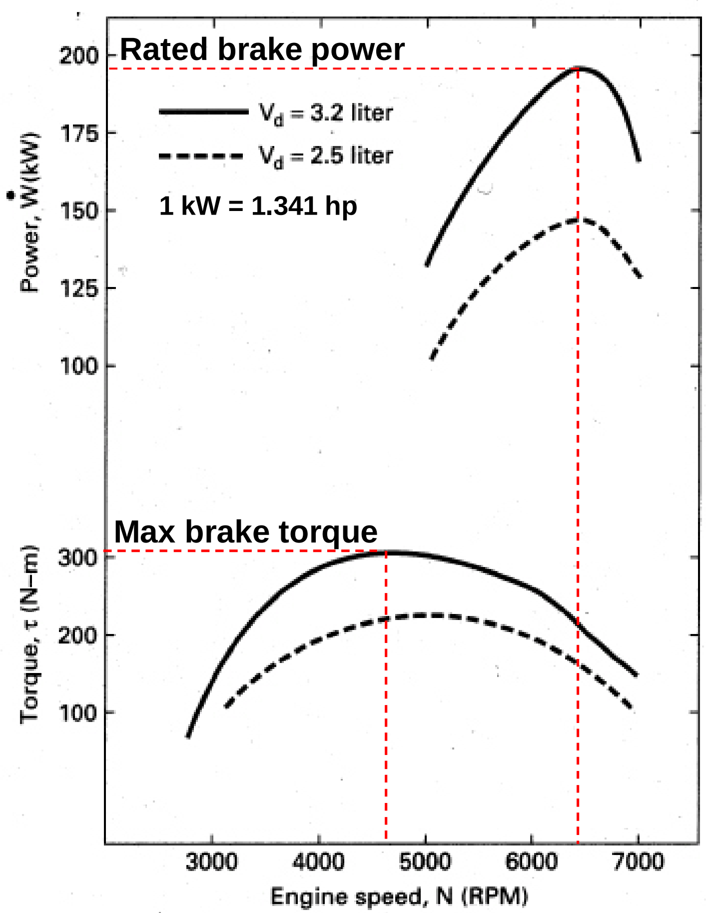
Friction power can be measured by motoring the engine.

The mechanical efficiency is defined as:

$$\eta_m = P_b / P_i = 1 - (P_f / P_i)$$

Mechanical efficiency depends on throttle position, engine design, and engine speed. Typical values for car engines at WOT are 90% @2000 RPM and 75% @ max speed.

Power and Torque versus Engine Speed



There is a maximum in the brake power versus engine speed called the **rated brake power**.

At higher speeds brake power decreases as friction power becomes significant compared to the indicated power

There is a maximum in the torque versus speed called **maximum brake torque** (MBT).

Brake torque drops off:

- at lower speeds do to heat losses
- at higher speeds it becomes more difficult to ingest a full charge of air.

Indicated Mean Effective Pressure (IMEP)

imep is a fictitious *constant* pressure that would produce the same work per cycle if it acted on the piston during the power stroke.

$$imep = W_i / V_d = (P_i\, n_R) / (V_d\, N)$$

so $P_i = imep\ V_d\ N / n_R = imep\ A_p\ U_p / (2\ n_R)$

imep does not depend on engine speed, just like torque.

imep is a better parameter than torque to compare engines for design and output because it is independent of engine speed, N, and engine size, V_d.

Brake mean effective pressure (bmep) is defined as:

$$bmep = \frac{W_b}{V_d} = \frac{2\pi \cdot T \cdot n_R}{V_d} \quad \rightarrow \quad T = \frac{bmep \cdot V_d}{2\pi \cdot n_R}$$

Maximum BMEP

$$bmep = \frac{W_b}{V_d} = \frac{2\pi \cdot T \cdot n_R}{V_d}$$

- The maximum bmep is obtained at WOT at a particular engine speed

- Closing the throttle decreases the bmep

- For a given displacement, a higher maximum bmep means more torque

- For a given torque, a higher maximum bmep means smaller engine

- Higher maximum bmep means higher stresses and temperatures in the engine hence shorter engine life, or bulkier engine.

- For the same bmep 2-strokes have almost twice the power of 4-stroke

Specific Fuel Consumption

- For transportation vehicles fuel economy is generally given as mpg, or liters/100 km.

- In engine testing the fuel consumption is measured in terms of the fuel mass flow rate.

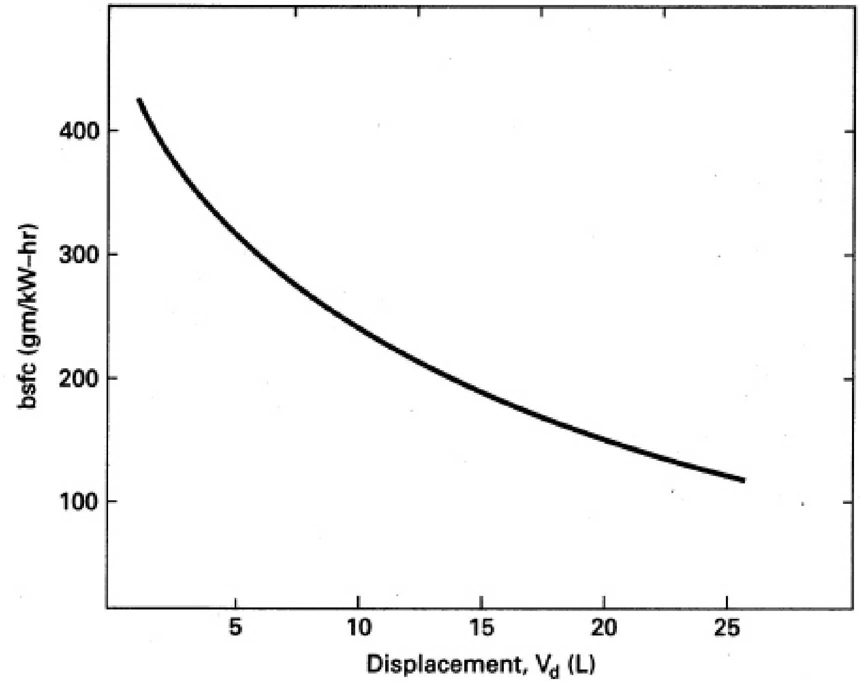
- The **specific fuel consumption,** sfc, is a measure of how efficiently the fuel supplied to the engine is used to produce power,

$$bsfc = \dot{m}_f / P_b \qquad isfc = \dot{m}_f / P_i \quad \text{(w/units: g/kW-hr)}$$

- Clearly a low value for sfc is desirable since at a given power level less fuel will be consumed

Brake Specific Fuel Consumption vs Size

•BSFC decreases with engine size due to reduced heat losses from gas to cylinder wall.

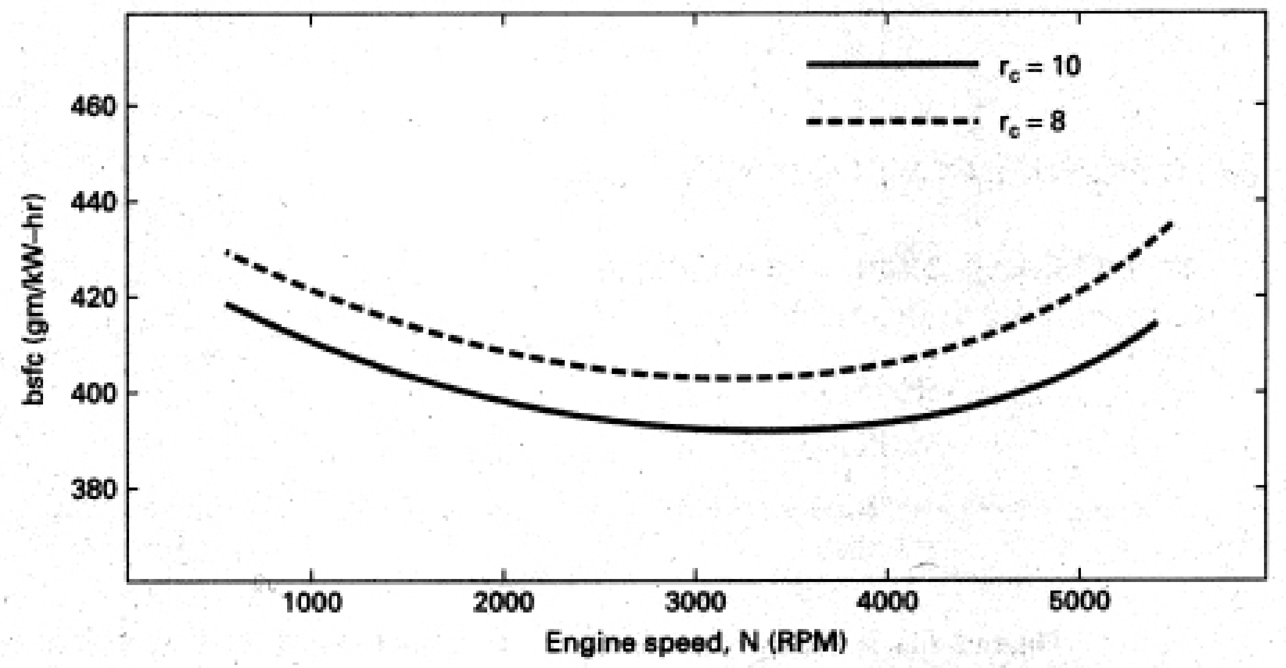


•Note: cylinder surface to volume ratio increases with bore diameter.

$$\frac{cylinder\ surface\ area}{cylinder\ volume} = \frac{2\pi r L}{\pi r^2 L} \propto \frac{1}{r}$$

Brake Specific Fuel Consumption vs Speed

- There is a minimum in the bsfc versus engine speed curve

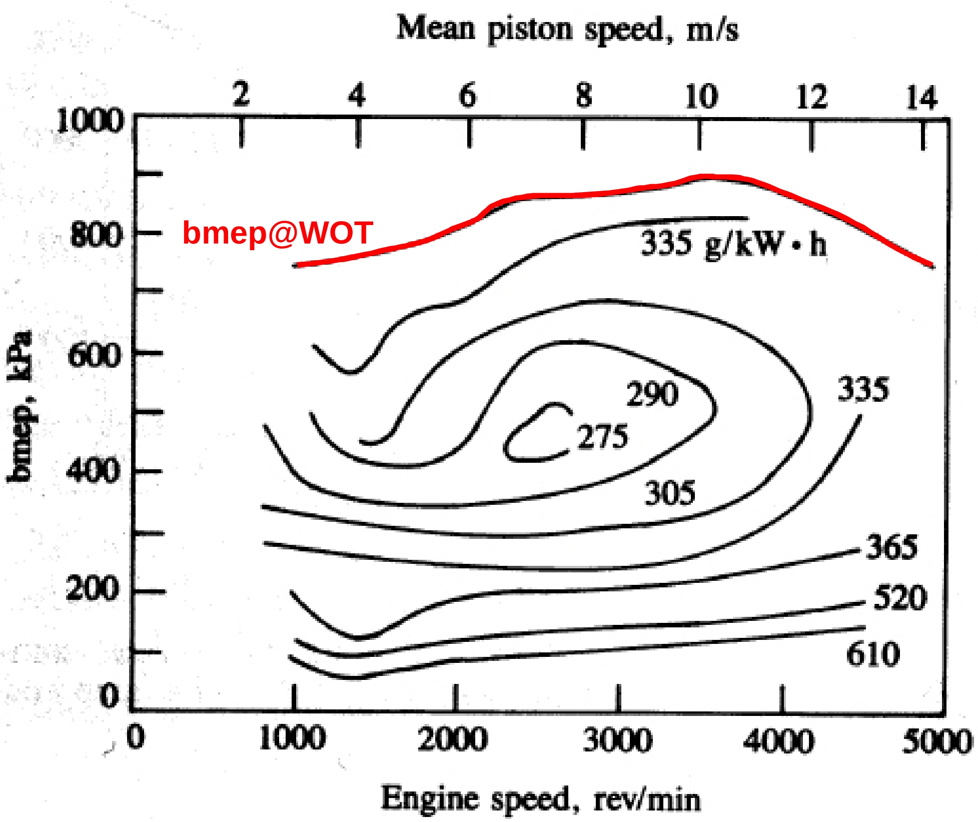


- At high speeds the bsfc increases due to increased friction
- At lower speeds the bsfc increases due to increased time for heat losses from the gas to the cylinder and piston wall
- Bsfc increases with compression ratio due to higher thermal efficiency

Performance Maps

Performance map is used to display the bsfc over the engines full load and speed range. Using a dynamometer to measure the torque and fuel mass flow rate you can calculate:

$$bmep = 2\pi\ T\ n_R\ /\ V_d \qquad P_b = 2\pi\ N\ T$$



$$bsfc = \dot{m}_f\ /\ P_b$$

Constant bsfc contours from a two-liter four cylinder SI engine

Combustion Efficiency

- The time for combustion in the cylinder is very short so not all the fuel may be consumed or local temperatures may not support combustion

- A small fraction of the fuel may not react and exits with the exhaust gas. The **combustion efficiency** is defined as actual heat input divided by theoretical heat input:

$$\eta_c = Q_{in} / (m_f\ Q_{HV}) = \dot{Q}_{in} / (\dot{m}_f\ Q_{HV})$$

Where Q_{in} = heat added by combustion per cycle
m_f = mass of fuel added to cylinder per cycle
Q_{HV} = heating value of the fuel (chemical energy per unit mass)

Thermal Efficiency

η_t = work per cycle / heat input per cycle

$\eta_t = W / Q_{in} = W / (\eta_c\, m_f\, Q_{HV})$

or in terms of <u>rates</u>…

η_t = power out/rate of heat input

$\eta_t = P/\dot{Q}_{in} = P/(\eta_c\, \dot{m}_f\, Q_{HV})$

- Thermal efficiencies can be given in terms of brake or indicated values
- Indicated thermal efficiencies are typically 50% to 60% and brake thermal efficiencies are usually about 30%

Arbitrary Efficiency
(aka fuel conversion efficiency)

$$\eta_f = W_b / (m_f\, Q_{HV}) = P_b / (\dot{m}_f\, Q_{HV})$$

Note: η_f is very similar to η_t, the difference is that η_t takes into account only the actual fuel combusted in the engine.

Recall that $sfc = \dot{m}_f / P_b$

Thus $\eta_f = 1 / (sfc\ Q_{HV})$

Volumetric Efficiency

- Due to the short cycle time and flow restrictions less than ideal amount of air enters the cylinder.

- The effectiveness of an engine to induct air into the cylinders is measured by the volumetric efficiency which is the ratio of actual air inducted divided by the theoretical air inducted:

$$\eta_v = m_a / (\rho_a V_d) = n_R \dot{m}_a / (\rho_a V_d N)$$

where ρ_a is the density of air at atmospheric conditions P_o, T_o for an ideal gas $\rho_a = P_o / R_a T_o$ and $R_a = 0.287\ kJ/kg\text{-}K$ (at standard conditions $\rho_a = 1.181\ kg/m^3$)

- Typical values for WOT are in the range 75%-90%, and lower when the throttle is closed

Air-Fuel Ratio

- For combustion to take place, the proper ratio of air and fuel must be present in the cylinder.

- The **air-fuel ratio** is defined as

$$AF = m_a / m_f = \dot{m}_a / \dot{m}_f$$

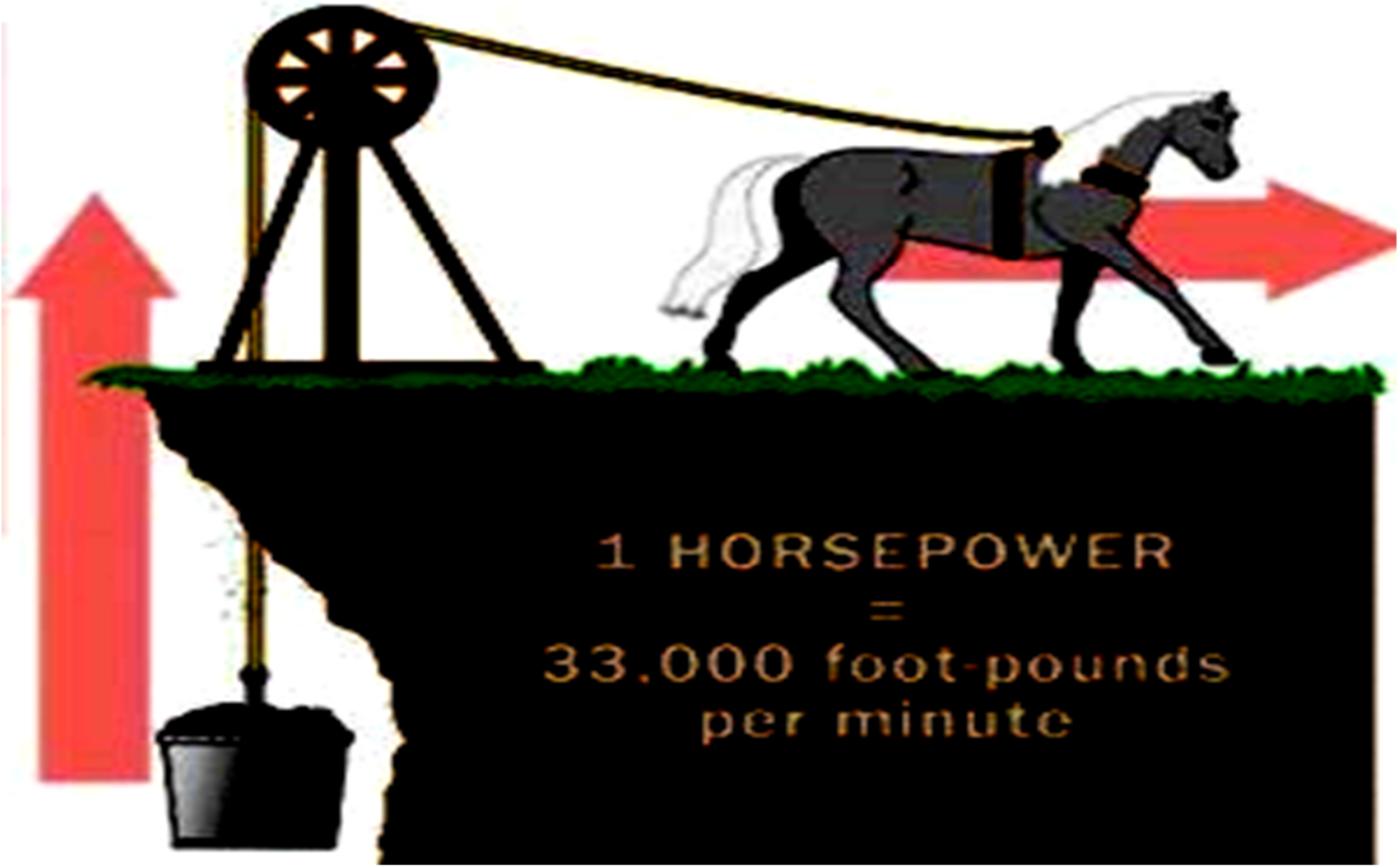
- The ideal AF is about 15:1, with homogenous combustion possible in the range of 6 to 19.

- For a SI engine the AF is in the range of 12 to 18 depending on the operating conditions.

- For a CI engine, where the mixture is highly non-homogeneous and the AF is in the range of 18 to 70.

James Watts Solution

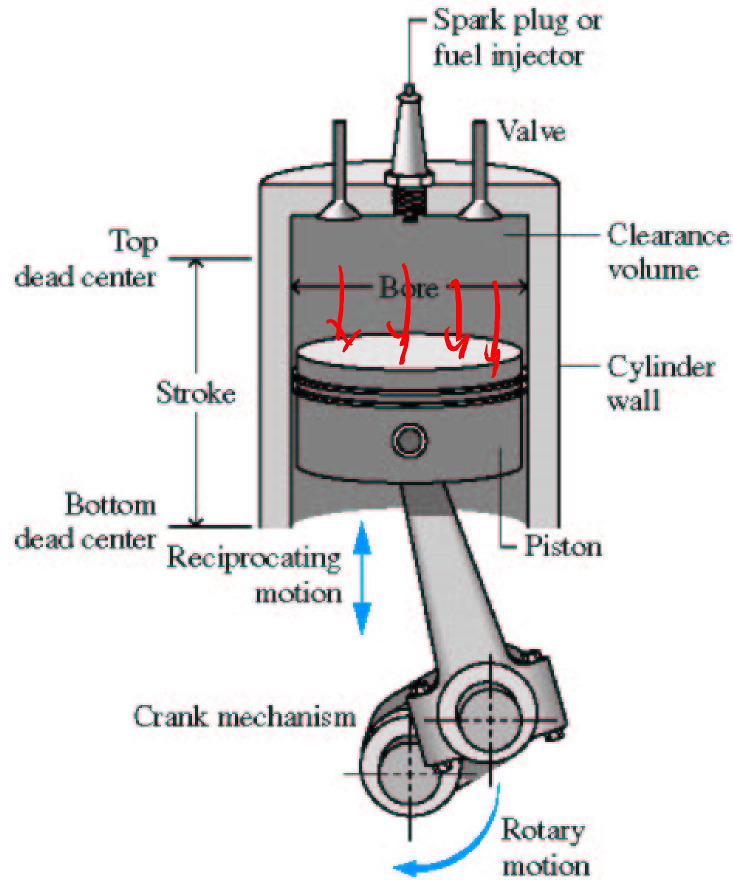
Walt



Cycle Performance Parameters

Net Work Transfer :

$$W_{net} = \oint pd(mv)$$

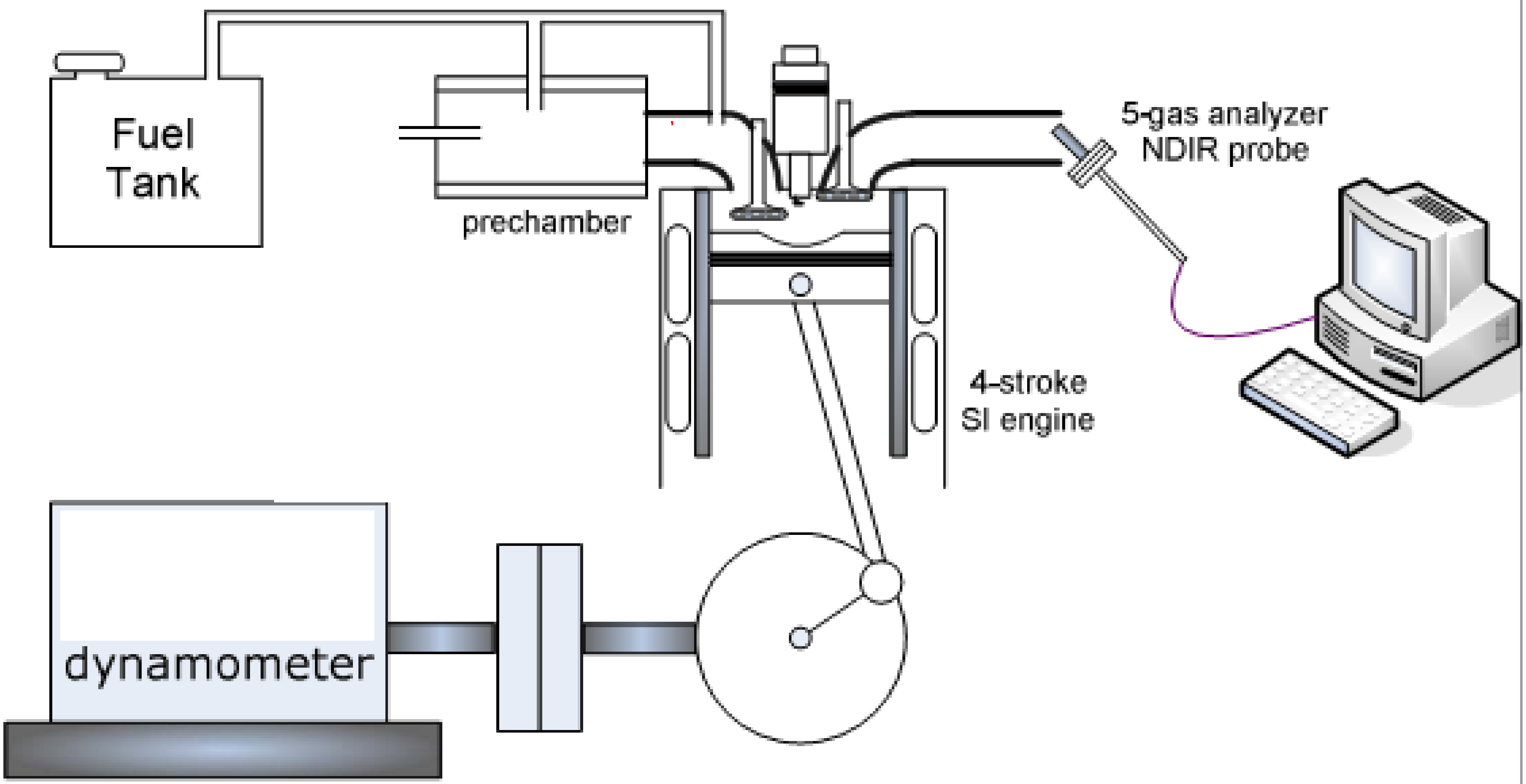


This is work done by working fluid on the piston, also called as Indicated Work.

Indicative Performance:

$$\eta_{ind} = \frac{\oint pd(mv)}{m_f \times CV}$$

I.C. Engine Test Rig



Specific Fuel Consumption

Fuel consumption of an engine reported in L/h or kg/h because these values ignore engine power. A better measure of fuel consumption is,

$$XSFC = \frac{m_f}{P_X}$$

- XSFC – specific fuel consumption (kg/kWh).
- X must always be specified when reporting these values (i.e., I for indicated)

Specific Fuel Consumption Variations

- ISFC – indicated specific fuel consumption
- BSFC - brake specific fuel consumption
- PSFC – PTO specific fuel consumption
- DSFC – drawbar specific fuel consumption

Parameters for Performance Diagnosis

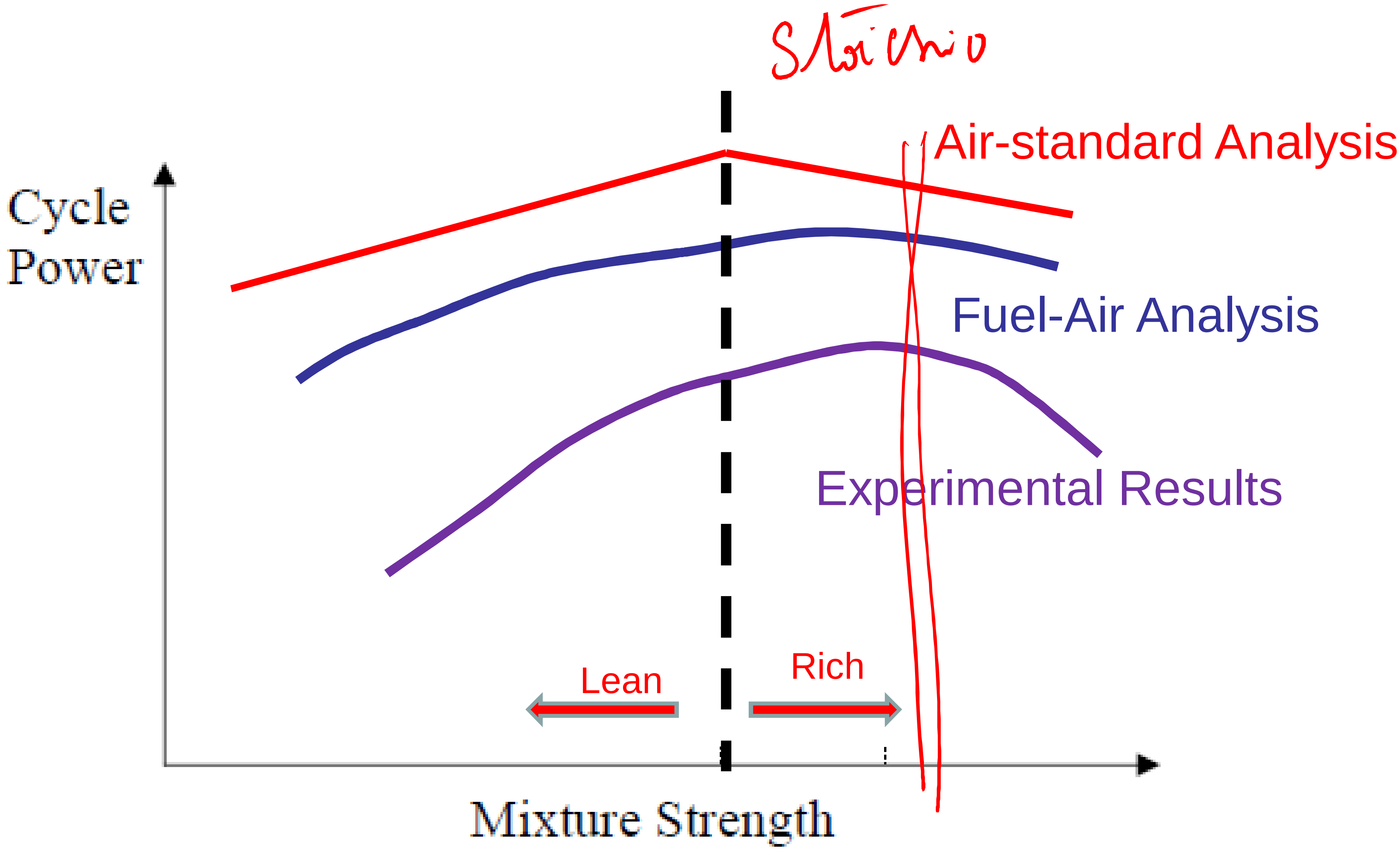
Indicative Mean Effective Pressure: $$IMEP = \frac{\oint pdv}{v_{\max} - v_{\min}}$$

Actual Fuel- Air Ratio : $$\left(\frac{F}{A}\right)_{act} = \frac{m_{fuel}}{m_{air,act}}$$

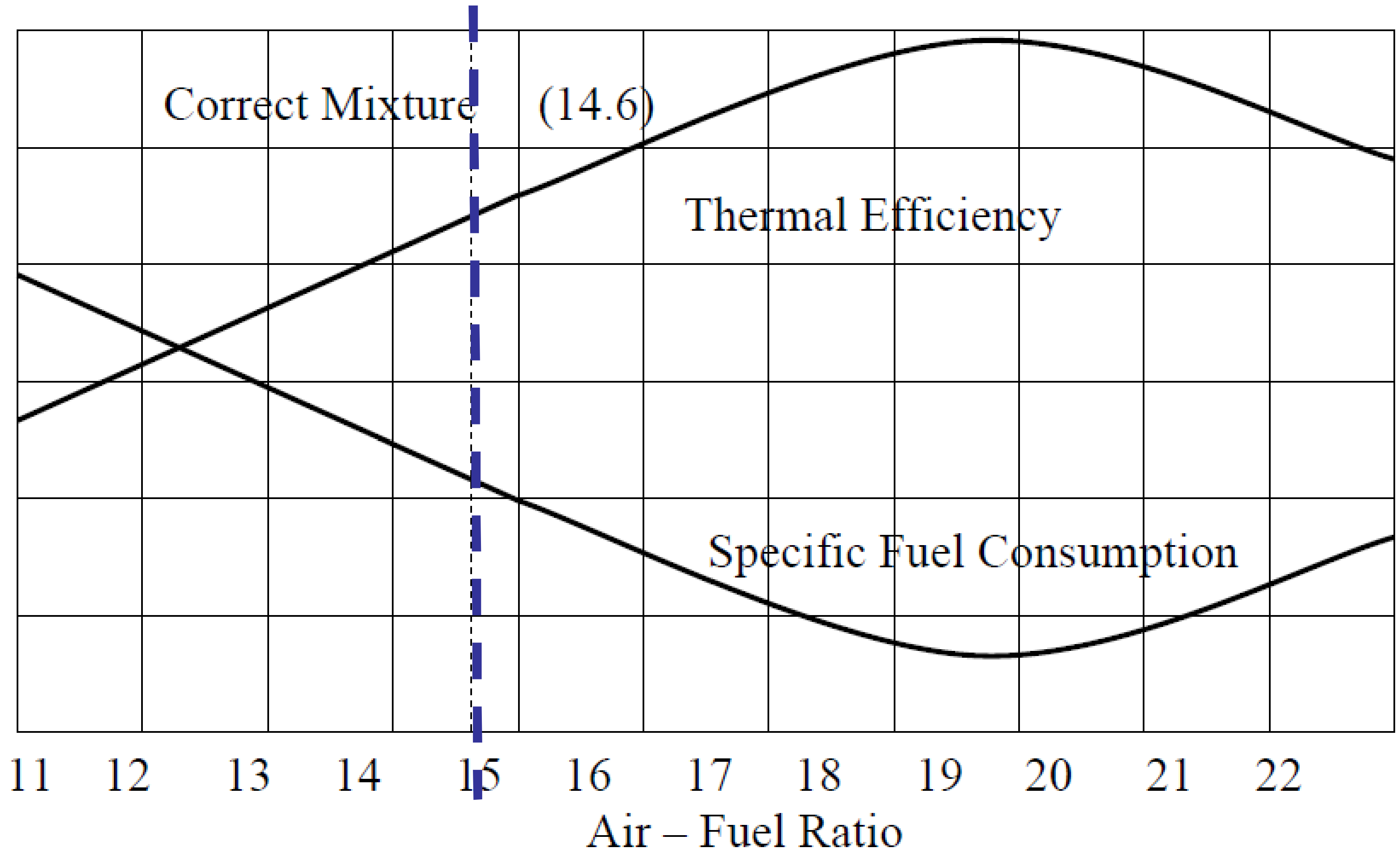
Stoichiometric Fuel- Air Ratio : $$\left(\frac{F}{A}\right)_{sto} = \frac{m_{fuel}}{m_{air,sto}}$$

Fuel Air Equivalence Ratio: $$\phi = \left(\frac{F}{A}\right)_{act} \Big/ \left(\frac{F}{A}\right)_{sto}$$

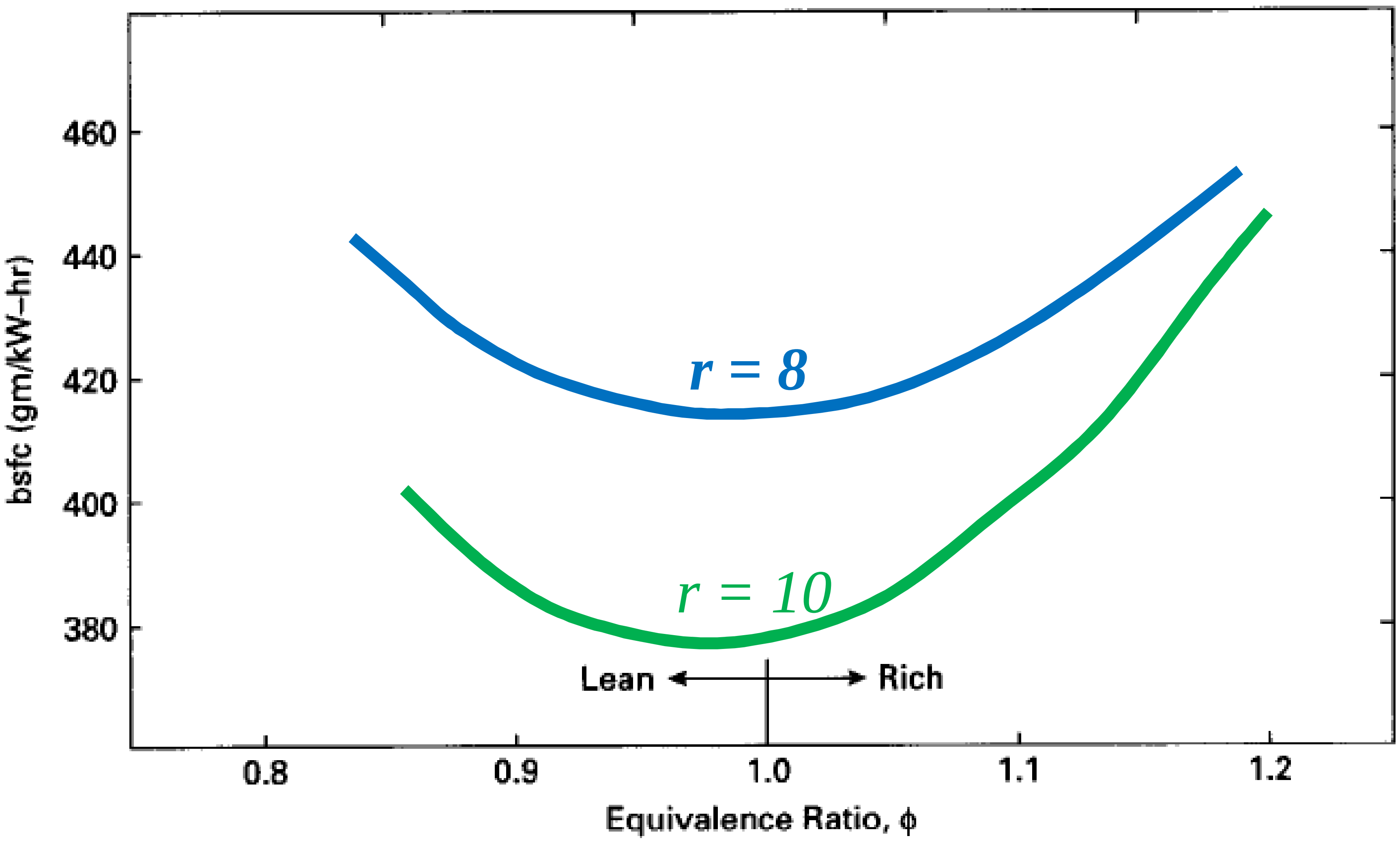
Selection of Mixture Strength for Better Reaction -1



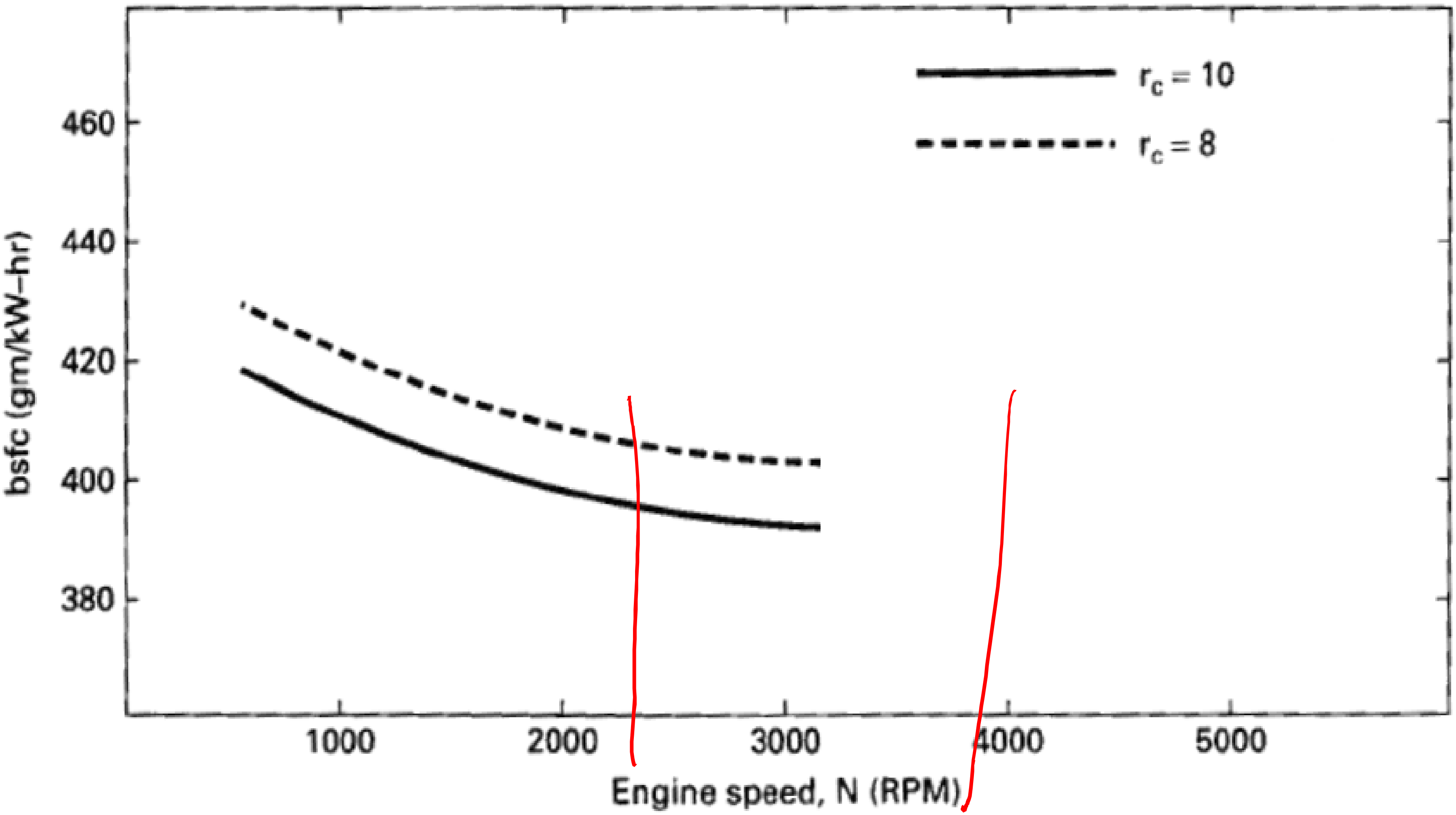
Selection of Sufficient Air : Optimization of Total Cost



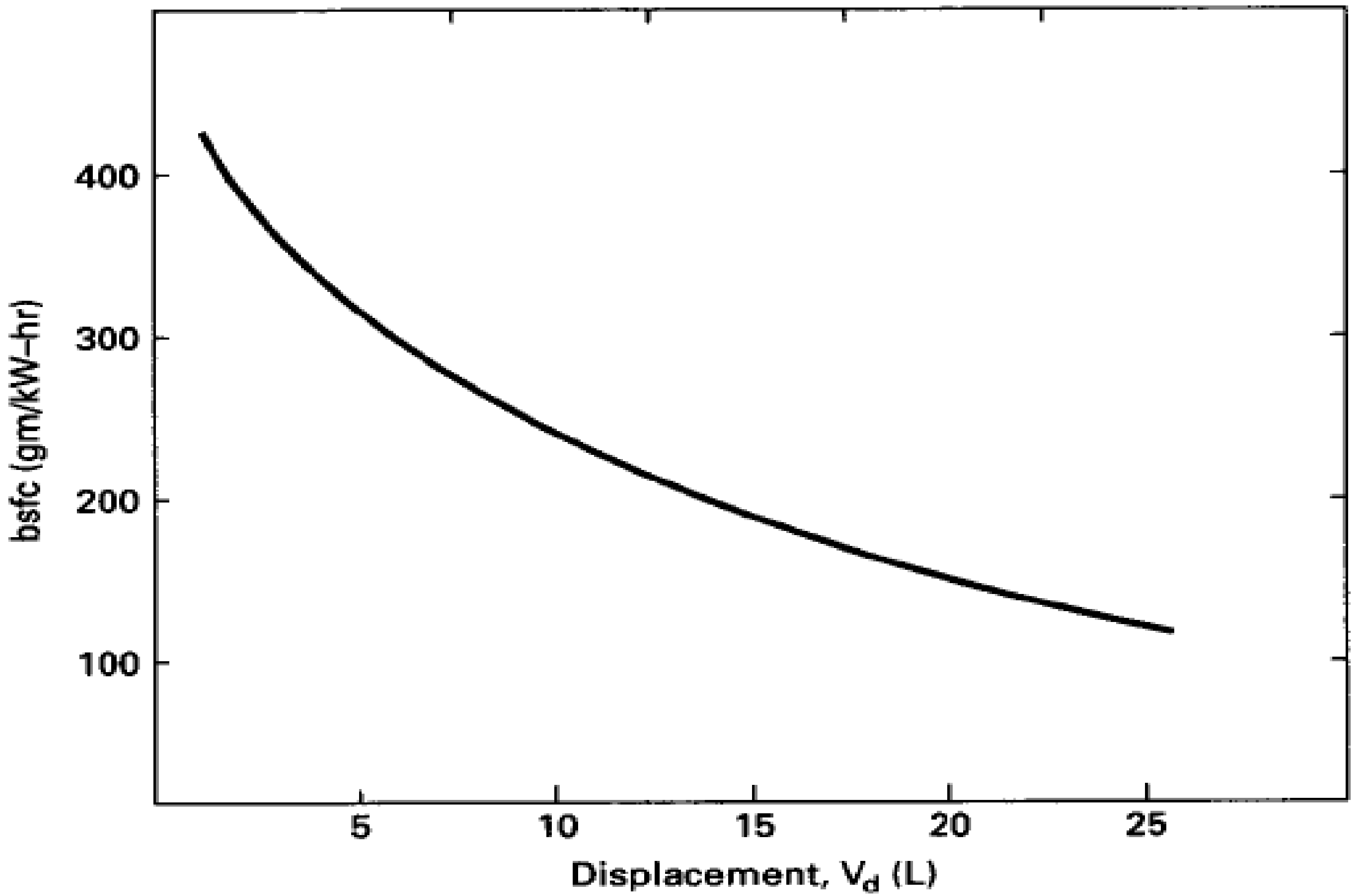
Combined Thermodynamic & Chemical Optimization for Better Reaction -2



Combined Thermodynamic & Kinetic Optimization for Better Reaction -2



Engine Capacity Vs Performance



Optimizing Engine Performance

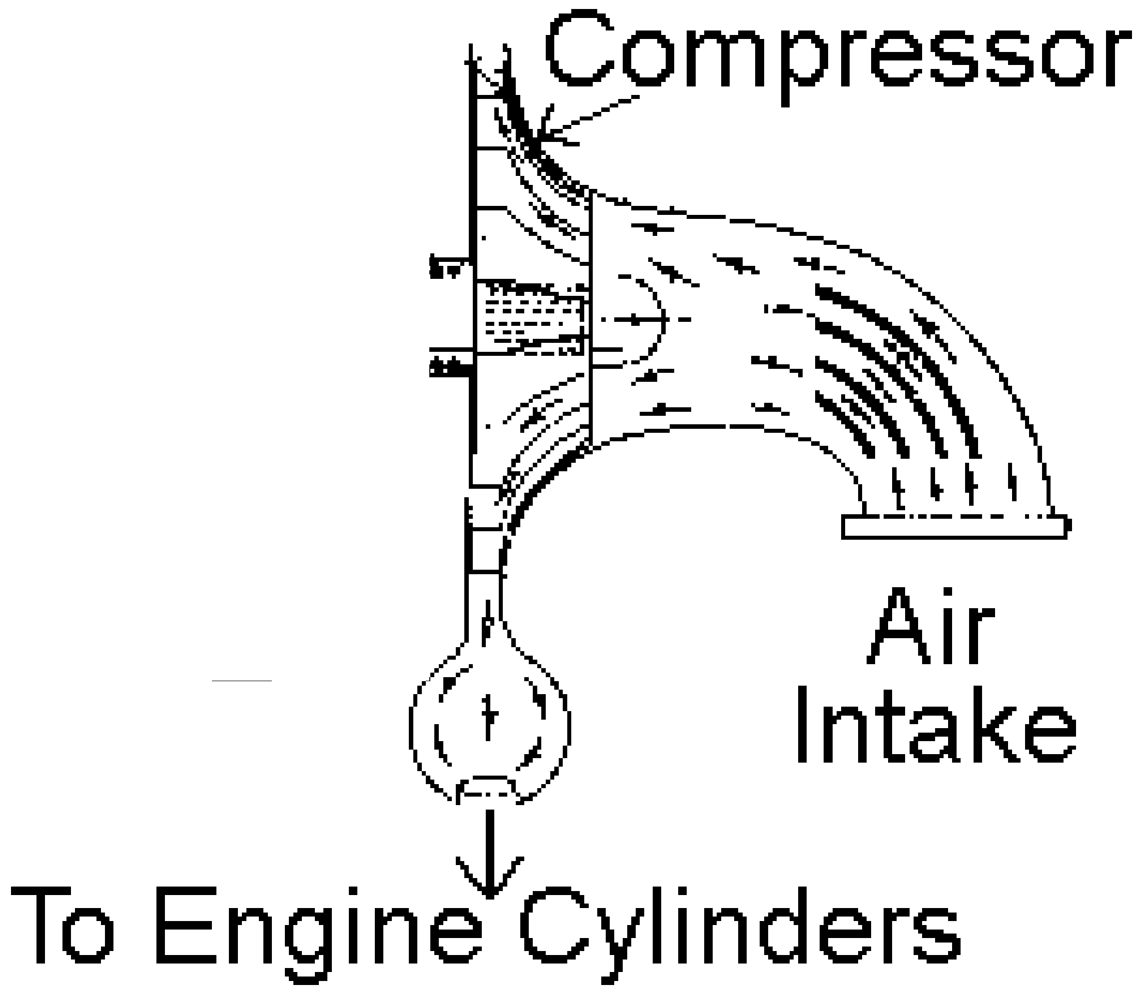
- Engines are most efficient at or near peak load.
- Efficiency drops with a reduction in torque load.
- At zero brake torque, all fuel energy is expended in engine friction.
- Lower rated engine speeds provide lower BSFC, and at the same time reduce torque reserve – design compromise.
- Heavy engines for a given capacity….. More inertial losses…
- Compromise – Necessary Evil …..
- Any alternate to overcome this fact…..
- Develop an idea to Change natural behaviour…..

Artificial Breathing Attachments to Engines

Preferred Artificial Breathing ….

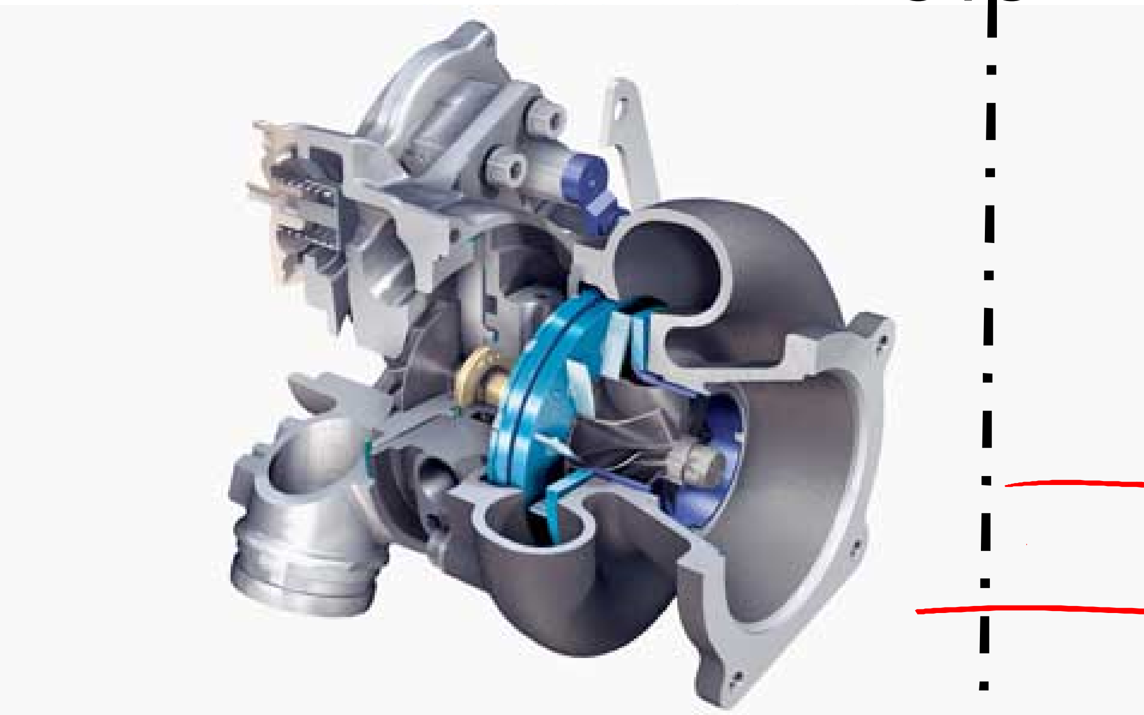
Engine Artificial Respiratory System

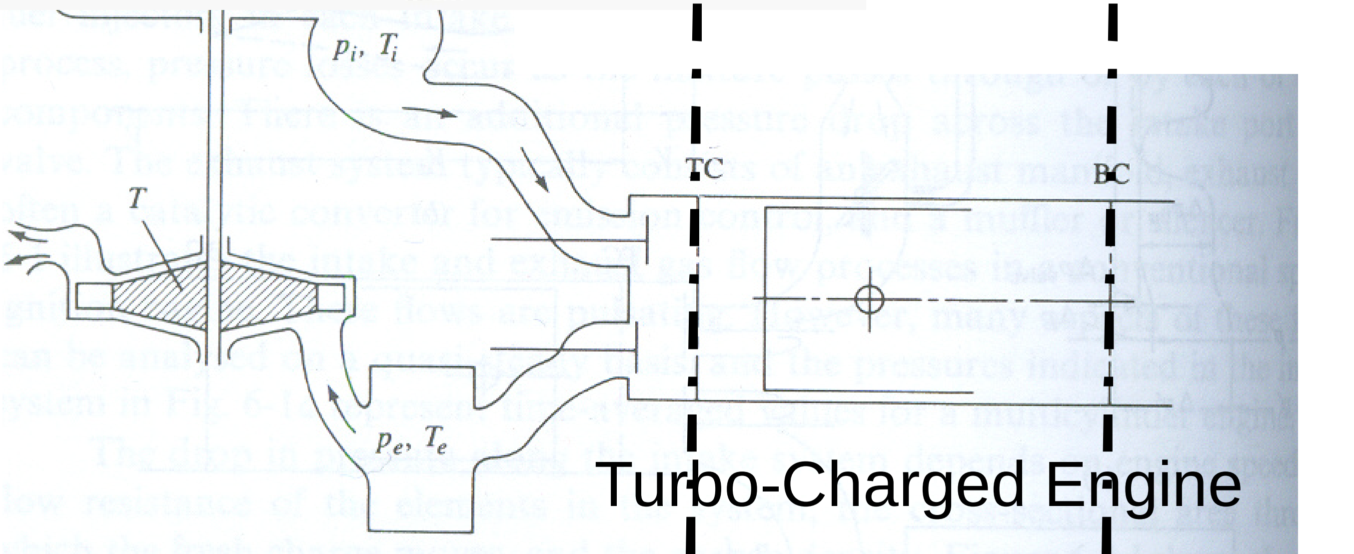
An Inclusion of A Pure CV



Super Charged Engine

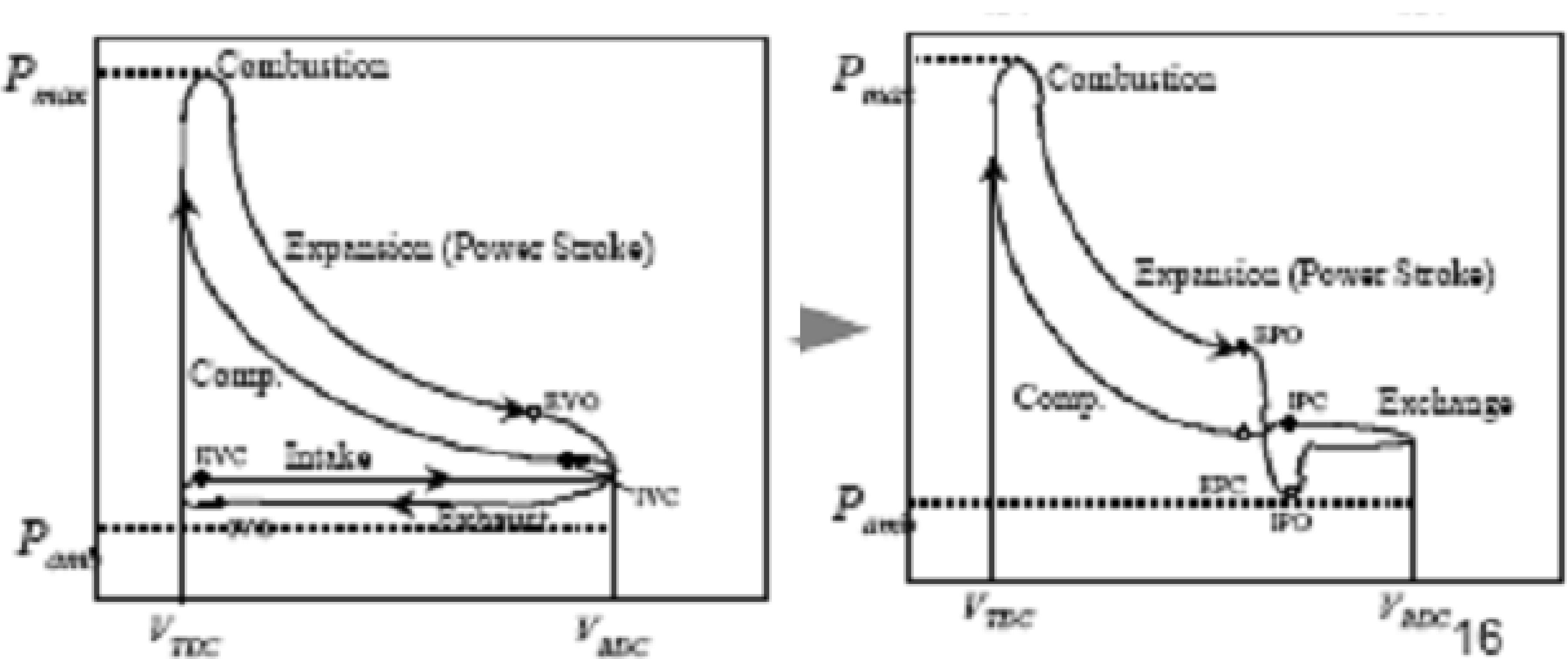
Turbo Charging of Engine : An Inclusion of two PURE CVs



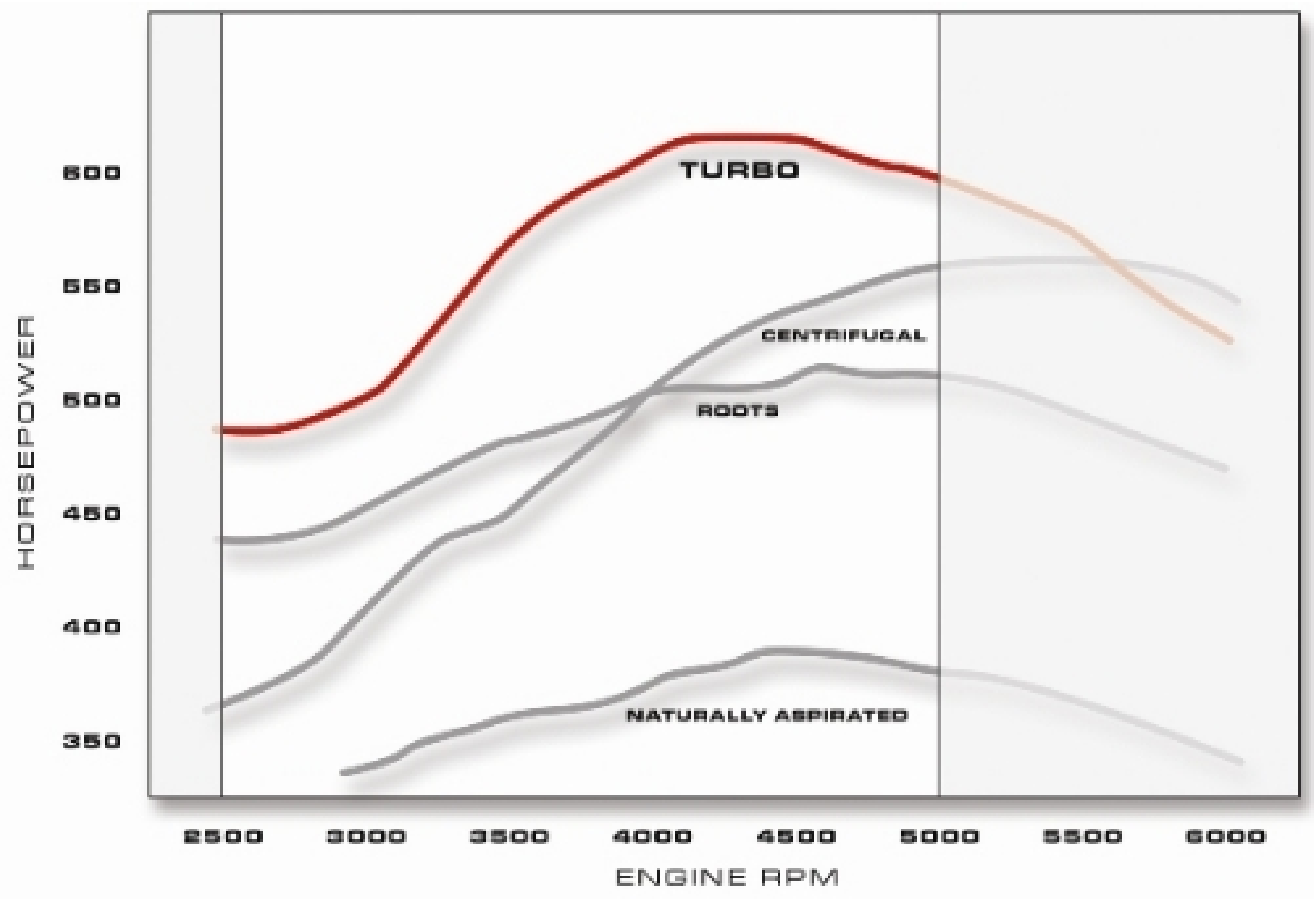


Turbo-Charged Engine

Artificially Aspirated Engines



Characteristics of Artificially Charged Engines



- Heat input to engine due to combustion of air-fuel mixture
- Combustion and heat release depends on Air-Fuel Ratio:

$$AF = m_a / m_f = \dot{m}_a / \dot{m}_f$$

$$FA = m_f / m_a = \dot{m}_f / \dot{m}_a$$

$m_a = mass\ of\ air$

$\dot{m}_a = mass\ flow\ rate\ of\ air$

$m_f = mass\ of\ fuel$

$\dot{m}_f = mass\ flow\ rate\ of\ fuel$

Equivalence Ratio

- Actual FA ratio to Stoichiometric FA ratio

$$\phi = (FA)_{act} / (FA)_{stoich} = (AF)_{stoich} / (AF)_{act}$$

- Specific fuel Consumption is:
  - Fuel Flow per unit Power Output.

$$sfc = \dot{m}_f / \dot{W}$$

- Brake Specific Fuel Consumption:

$$bsfc = \dot{m}_f / \dot{W}_b$$

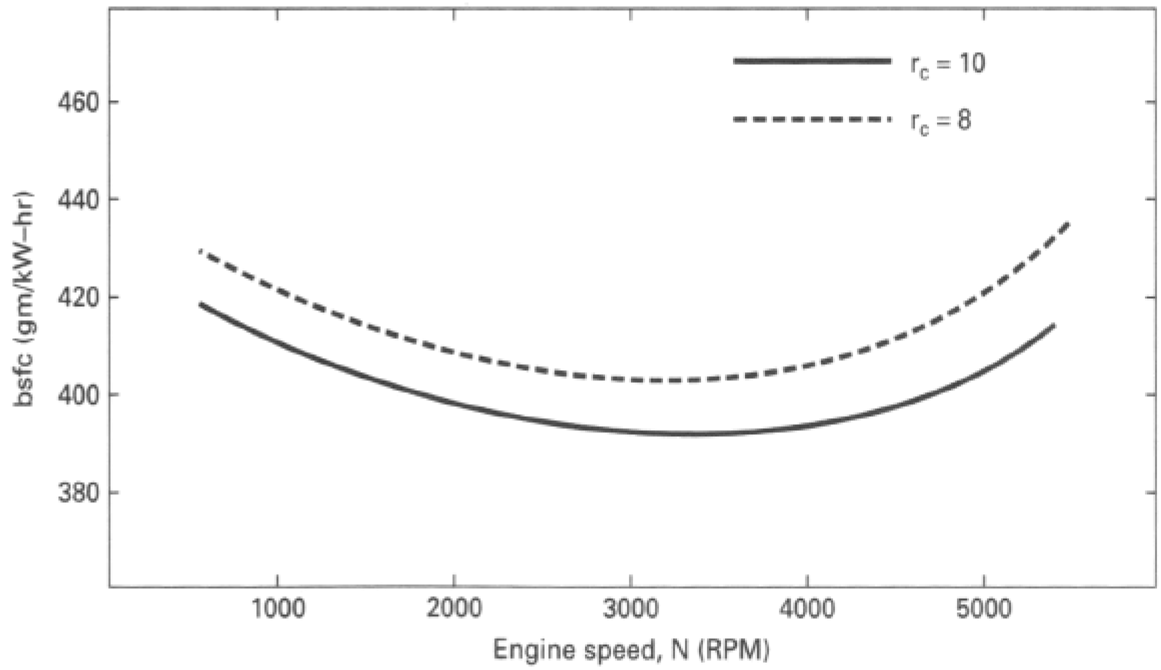
- Indicated Specific Fuel Consumption:

$$isfc = \dot{m}_f / \dot{W}_i$$

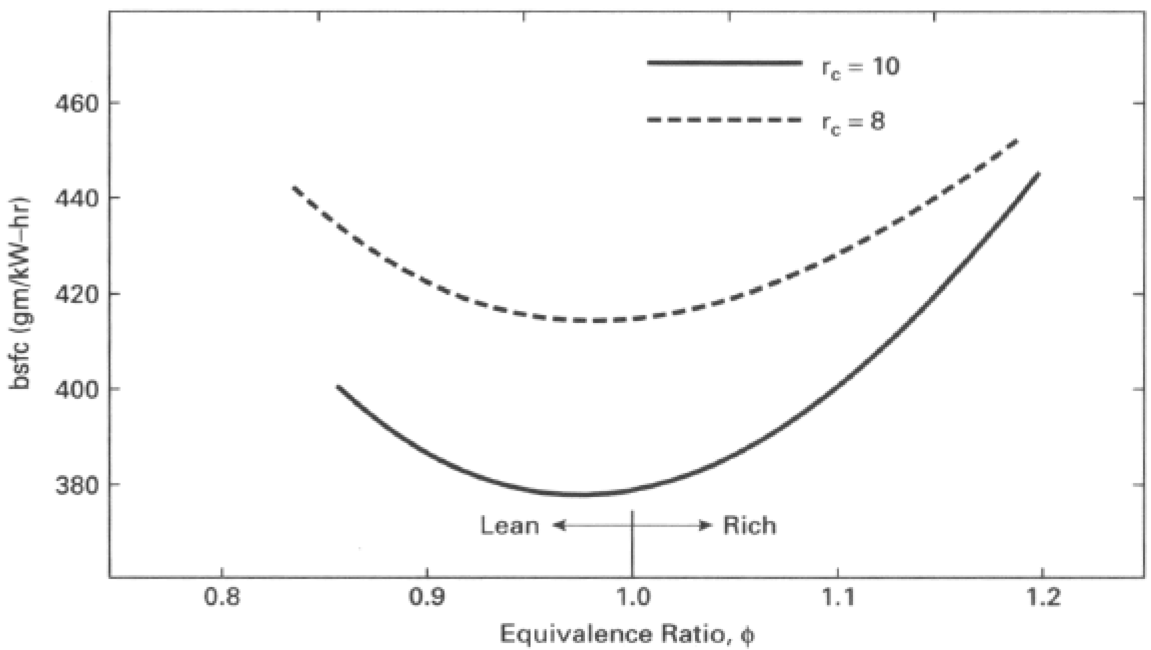
$$\begin{aligned}\eta_m &= \dot{W}_b / \dot{W}_i \\ &= (\dot{m}_f / \dot{W}_i)/(\dot{m}_f / \dot{W}_b) \\ &= (isfc)/(bsfc)\end{aligned}$$

$\eta_m = \text{engine mechanical efficiency}$

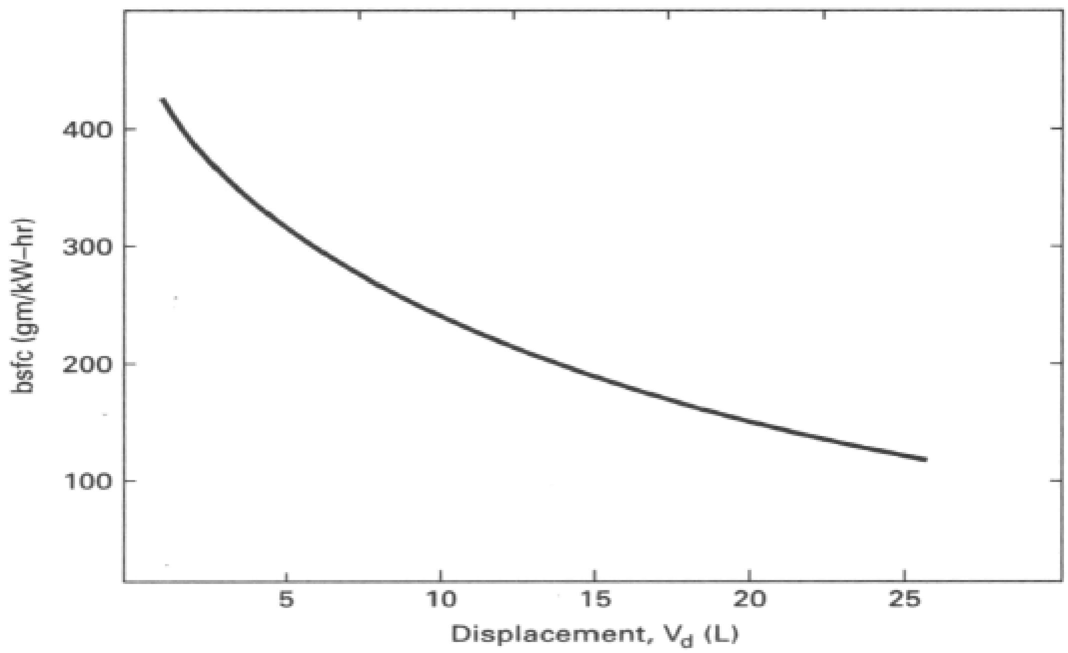
bsfc vs Engine Speed



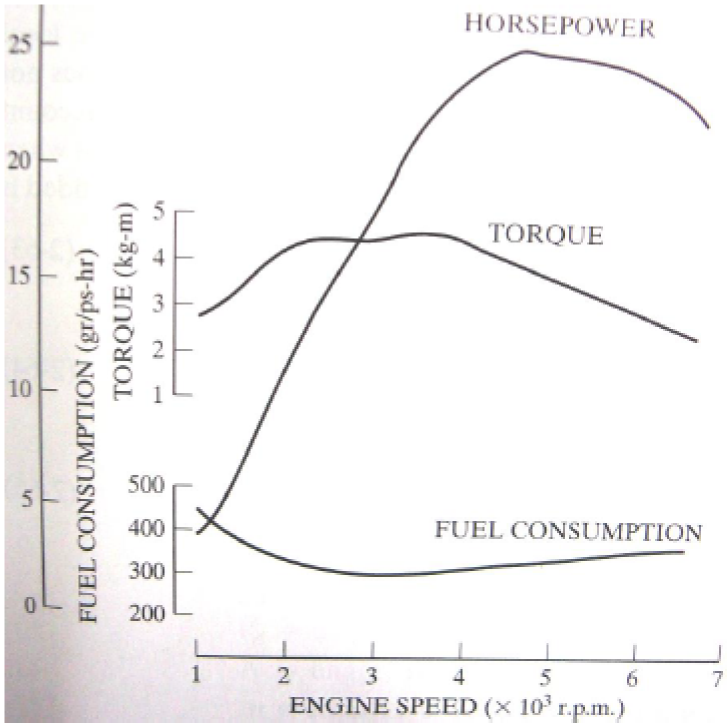
bsfc vs Equivalence Ratio



bsfc vs Displacement



Performance Curves



Volumetric Efficiency

- *Measure of air mass entering cylinder during intake stroke.*
- *Typical values at WOT → 75% to 90%.*

$$\eta_v = m_a / \rho_a V_d$$
$$\eta_v = n\,\dot{m}_a / \rho_a V_d N$$

m_a = mass of air into the cylinder for one cycle
$\dot{m}_a$ = steady state flow of air into the engine
ρ_a = air density
V_d = displacement volume
N = engine speed
n = number of revolutions per cycle

- Heat added for 1 cycle in 1 cylinder is:

$$Q_{in} = m_f Q_{HV} \eta_c$$

$\eta_c \rightarrow$

- *Combustion Efficiency.*
- *Typically → 0.95 ~ 0.98*

- *Heat added for steady state is:*

$$\dot{Q}_{in} = \dot{m}_f Q_{HV} \eta_c$$

W = work of one cycle

$\dot{W}$ = power

m_f = mass of fuel for one cycle

$\dot{m}_f$ = mass flow rate of fuel

Q_{HV} = heating value of fuel

η_f = fuel conversion efficiency

- *Thermal Efficiency,*

$$\eta_t = W/Q_{in} = \dot{W}/\dot{Q}_{in} = \dot{W}/\dot{m}_f Q_{HV} \eta_c = \eta_f/\eta_c$$

Engine Efficiencies Relationship

$$\eta_m = (\eta_t)_b / (\eta_t)_i$$

$$\eta_f = W / m_f Q_{HV} = \dot{W} / \dot{m}_f Q_{HV}$$

$$\eta_f = 1/(sfc) Q_{HV}$$

$(\eta_t)_i \sim 40\%$ to 50%

$(\eta_t)_b \sim 30\%$

$\sim$ greater than 50% for some large CI engines.

Example 1

John's automobile has a 3 liter SI V6 engine that operates on a 4-stroke cycle at 3600 rpm. The compression ratio is 9.5, the length of the connecting rods is 16.6 cm, and the engine is square (B=S). At this speed, combustion ends at 20^0 ATDC. Calculate:

1. Cylinder bore and stroke length
2. Average piston speed
3. Clearance volume of one cylinder
4. Piston speed at the end of combustion
5. Distance the piston has traveled from TDC at the end of combustion
6. Volume in the combustion chamber at the end of combustion

Example 2

The engine in Example 1 is connected to a dynamometer which gives a brake output torque reading of 205 Nm at 3600 rpm. At this speed air enters the cylinders at 85 kPa and 60^0C, and mechanical efficiency of the engine is 85%. Calculate:

1. Brake power
2. Indicated power
3. bmep
4. imep
5. fmep
6. Power lost to friction
7. Brake power per unit mass of gas in the cylinder
8. Brake specific power
9. Brake output per displacement
10. Engine specific volume

- 4 main engine exhaust emissions → NOx, CO, HC, PM
- Two common methods of measuring the amounts of pollutants →
  - Specific Emissions (SE)
  - Emissions Index (EI)

$$(SE)_{NOx} = \dot{m}_{NOx} / \dot{W}_b$$
$$(SE)_{CO} = \dot{m}_{CO} / \dot{W}_b$$
$$(SE)_{HC} = \dot{m}_{HC} / \dot{W}_b$$
$$(SE)_{PM} = \dot{m}_{PM} / \dot{W}_b$$

$$(EI)_{NOx} = \dot{m}_{NOx}\,[\mathrm{g/sec}] / \dot{m}_f\,[\mathrm{kg/sec}]$$
$$(EI)_{CO} = \dot{m}_{CO}\,[\mathrm{g/sec}] / \dot{m}_f\,[\mathrm{kg/sec}]$$
$$(EI)_{HC} = \dot{m}_{HC}\,[\mathrm{g/sec}] / \dot{m}_f\,[\mathrm{kg/sec}]$$
$$(EI)_{PM} = \dot{m}_{PM}\,[\mathrm{g/sec}] / \dot{m}_f\,[\mathrm{kg/sec}]$$

$\dot{m}$ = *flow rate of emissions in g/hr*

$\dot{W}_b$ = *brake power in kW*

Example 4

A 12 cylinder, 4-stroke cycle CI engine produces 2440 kW of brake power at 550 rpm using stoichiometric light diesel fuel. The engine has bore of 24 cm, stroke of 32 cm, volumetric efficiency of 97%, mechanical efficiency of 88% and combustion efficiency of 98%. Calculate:

1. Mass flow rate of fuel into engine
2. Brake specific fuel consumption
3. Indicated specific fuel consumption
4. Specific emissions of hydrocarbons due to unburned fuel
5. Emissions index of hydrocarbons due to unburned fuel

Example 01

A four-cylinder, 2.5-liter, SI automobile engine operates at WOT on a four-stroke air-standard Otto cycle at 3000 RPM. The engine has a compression ratio of 8.6:1, a mechanical efficiency of 86%, and a stroke-to-bore ratio $S/B = 1.025$. Fuel is isooctane with AF = 15, a heating value of 44,300 kJ/kg, and combustion efficiency $\eta_c = 100\%$. At the start of the compression stroke, conditions in the cylinder combustion chamber are 100 kPa and 60°C. It can be assumed that there is a 4% exhaust residual left over from the previous cycle.

Do a complete thermodynamic analysis of this engine.

Answer 01

For one cylinder, the displacement volume is

$$V_d = 2.5 \text{ liter}/4 = \underline{0.625 \text{ L} = 0.000625 \text{ m}^3}$$

Use Eq. (2-12) to find clearance volume:

$$r_c = V_1/V_2 = (V_c + V_d)/V_c = 8.6 = (V_c + 0.000625)/V_c$$

$$\underline{V_c = 0.0000822 \text{ m}^3 = 0.0822 \text{ L} = 82.2 \text{ cm}^3}$$

Use Eq. (2-8) to find bore and stroke:

$$V_d = (\pi/4)B^2S = (\pi/4)B^2(1.025B) = 0.000625 \text{ m}^3$$

$$\underline{B = 0.0919 \text{ m} = 9.19 \text{ cm}}$$

$$S = 1.025B \underline{= 0.0942 \text{ m} = 9.42 \text{ cm}}$$

State 1:

$$\underline{T_1 = 60°\text{C} = 333\text{ K}} \quad \text{given in problem statement}$$
$$\underline{P_1 = 100\text{ kPa}} \qquad \text{given}$$
$$V_1 = Vd + V_c = 0.000625 + 0.0000822 \ \underline{= 0.000707\text{ m}^3}$$

Mass of gas mixture in the cylinder can be calculated at State 1. The mass within the cylinder will then remain the same for the entire cycle.

$$\begin{aligned} m_m &= P_1V_1/RT_1 = (100\text{ kPa})(0.000707\text{ m}^3)/(0.287\text{ kJ/kg-K})(333\text{ K}) \\ &= 0.000740\text{ kg} \end{aligned}$$

State 2: The compression stroke 1-2 is isentropic. Use Eqs. (3-4) and (3-5) to find the pressure and temperature:

$$\begin{aligned} P_2 &= P_1(r_c)^k = (100\text{ kPa})(8.6)^{1.35} \ \underline{= 1826\text{ kPa}} \\ T_2 &= T_1(r_c)^{k-1} = (333\text{ K})(8.6)^{0.35} \ \underline{= 707\text{ K} = 434°\text{C}} \\ V_2 &= mRT_2/P_2 = (0.000740\text{ kg})(0.287\text{ kJ/kg-K})(707\text{ K})/(1826\text{ kPa}) \\ &\underline{= 0.0000822\text{ m}^3 = V_c} \end{aligned}$$

This is the clearance volume of one cylinder, which agrees with the preceding. Another way of getting this value is to use Eq. (2-12):

$$V_2 = V_1/r_c = 0.000707\ \mathrm{m}^3/8.6 = 0.0000822\ \mathrm{m}^3$$

The mass of gas mixture m_m in the cylinder is made up of air m_a, fuel m_f, and exhaust residual m_{ex}:

mass of air	$m_a = (15/16)(0.96)(0.000740)$	$= 0.000666$ kg
mass of fuel	$m_f = (1/16)(0.96)(0.000740)$	$= 0.000044$ kg
mass of exhaust	$m_{ex} = (0.04)(0.000740)$	$= 0.000030$ kg
Total		$m_m = 0.000740$ kg

State 3: Use Eq. (3-10) to calculate the heat added during one cycle:

$$\begin{aligned} Q_{\text{in}} &= m_f Q_{\text{HV}} \eta_c = m_m c_v (T_3 - T_2) \\ &= (0.000044 \text{ kg})(44{,}300 \text{ kJ/kg})(1.00) \\ &= (0.000740 \text{ kg})(0.821 \text{ kJ/kg-K})(T_3 - 707 \text{ K}) \end{aligned}$$

Solving this for T_3

$$\underline{T_3 = 3915 \text{ K} = 3642°\text{C} = T_{\text{max}}}$$
$$V_3 = V_2 = \underline{0.0000822 \text{ m}^3}$$

For constant volume

$$P_3 = P_2(T_3/T_2) = (1826 \text{ kPa})(3915/707) \underline{= 10{,}111 \text{ kPa} = P_{\text{max}}}$$

State 4: Power stroke 3-4 is isentropic. Use Eq. (3-16) and (3-17) to find temperature and pressure:

$$\begin{aligned} T_4 &= T_3(1/r_c)^{k-1} = (3915 \text{ K})(1/8.6)^{0.35} \underline{= 1844 \text{ K} = 1571°\text{C}} \\ P_4 &= P_3(1/r_c)^k = (10{,}111 \text{ kPa})(1/8.6)^{1.35} \underline{= 554 \text{ kPa}} \\ V_4 &= mRT_4/P_4 = (0.000740 \text{ kg})(0.287 \text{ kJ/kg-K})(1844 \text{ K})/(554 \text{ kPa}) \\ &\underline{= 0.000707 \text{ m}^3 = V_1} \end{aligned}$$

This agrees with the value of V_1 found earlier.

Work produced in the isentropic power stroke for one cylinder during one cycle is

$$\begin{aligned} W_{3-4} &= mR(T_4 - T_3)/(1 - k) \\ &= (0.000740\text{ kg})(0.287\text{ kJ/kg-K})(1844 - 3915)\text{K}/(1 - 1.35) \\ &= 1.257\text{ kJ} \end{aligned}$$

Work absorbed during the isentropic compression stroke for one cylinder during one cycle is

$$\begin{aligned} W_{1-2} &= mR(T_2 - T_1)/(1 - k) \\ &= (0.000740\text{ kg})(0.287\text{ kJ/kg-K})(707 - 333)\text{K}/(1 - 1.35) \\ &= -0.227\text{ kJ} \end{aligned}$$

Work of the intake stroke is canceled by work of the exhaust stroke.

Net indicated work for one cylinder during one cycle is

$$W_{\text{net}} = W_{1-2} + W_{3-4} = (-0.227) + (+1.257) \underline{= +1.030\text{ kJ}}$$

Use Eq. (3-10) to find heat added for one cylinder during one cycle:

$$Q_{\text{in}} = m_f Q_{\text{HV}} \eta_c = (0.000044\text{ kg})(44{,}300\text{ kJ/kg})(1.00) \underline{= 1.949\text{ kJ}}$$

Indicated thermal efficiency is

$$\eta_t = W_{\text{net}}/Q_{\text{in}} = 1.030/1.949 \underline{= 0.529 = 52.9\%}$$

or, using Eqs. (3-29) and (3-31):

$$\begin{aligned} \eta_t &= 1 - (T_1/T_2) = 1 - (1/r_c)^{k-1} \\ &= 1 - (333/707) = 1 - (1/8.6)^{0.35} = 0.529 \end{aligned}$$

Equation (2-29) is used to find indicated mean effective pressure:

$$\text{imep} = W_{\text{net}}/(V_1 - V_2) = (1.030\text{ kJ})/(0.000707 - 0.0000822)\text{m}^3 \underline{= 1649\text{ kPa}}$$

indicated power at 3000 RPM is obtained using Eq. (2-42):

$$\dot{W}_i = WN/n$$
$$= [(1.030\ \text{kJ/cyl-cycle})(3000/60\ \text{rev/sec})/(2\ \text{rev/cycle})](4\ \text{cyl})$$
$$\underline{= 103\ \text{kW} = 138\ \text{hp}}$$

Equation (2-2) is used to find mean piston speed:

$$\overline{U}_p = 2SN = (2\ \text{strokes/rev})(0.0942\ \text{m/stroke})(3000/60\ \text{rev/sec})$$
$$\underline{= 9.42\ \text{m/sec}}$$

Equation (2-27) gives net brake work for one cylinder during one cycle:

$$W_b = \eta_m W_i = (0.86)(1.030\ \text{kJ})\ \underline{= 0.886\ \text{kJ}}$$

Brake power at 3000 RPM is

$$\dot{W}_b = (3000/60\ \text{rev/sec})(0.5\ \text{cycle/rev})(0.886\ \text{kJ/cyl-cycle})(4\ \text{cyl})$$
$$\underline{= 88.6\ \text{kW} = 119\ \text{hp}}$$

or

$$\dot{W}_b = \eta_m \dot{W}_i = (0.86)(103\ \text{kW}) = 88.6\ \text{kW}$$

Torque is calculated using Eq. (2-43):

$$\tau = \dot{W}_b/2\pi N = (88.6\ \text{kJ/sec})/(2\pi\text{radians/rev})(3000/60\ \text{rev/sec})$$
$$\underline{= 0.282\ \text{kN-m} = 282\ \text{N-m}}$$

Friction power lost is calculated using Eq. (2-49):

$$\dot{W}_f = \dot{W}_i - \dot{W}_b = 103 - 88.6 \underline{= 14.4\ \text{kW} = 19.3\ \text{hp}}$$

Equation (2-37c) is used to find brake mean effective pressure:

$$\text{bmep} = \eta_m(\text{imep}) = (0.86)(1649\ \text{kPa}) \underline{= 1418\ \text{kPa}}$$

This allows another way of finding torque using Eq. (2-41), which gives consistent results:

$$\tau = (\text{bmep})V_d/4\pi = (1418\ \text{kPa})(0.0025\ \text{m}^3)/4\pi = 0.282\ \text{kN-m}$$

Brake specific power is calculated using Eq. (2-51):

$$\text{BSP} = \dot{W}_b/A_p = (88.6\ \text{kW})/\{[(\pi/4)(9.19\ \text{cm})^2](4\ \text{cyl})\} = \underline{0.334\ \text{kW/cm}^2}$$

Output per displacement is found using Eq. (2-52):

$$\text{OPD} = \dot{W}_b/V_d = (88.6\ \text{kW})/(2.5\ \text{L}) \underline{= 35.4\ \text{kW/L}}$$

Equation (2-60) is used to find brake specific fuel consumption:

$$\text{bsfc} = \dot{m}_f/\dot{W}_b$$
$$= (0.000044\ \text{kg/cyl-cycle})\ (50\ \text{rev/sec})\ (0.5\ \text{cycle/rev})\ (4\ \text{cyl})/(88.6\ \text{kW})$$
$$\underline{= 0.000050\ \text{kg/sec/kW} = 180\ \text{gm/kW-hr}}$$

Equation (2-70) is used to find volumetric efficiency using one cylinder and standard air density:

$$\eta_v = m_a/\rho_a V_d = (0.000666\ \text{kg})/(1.181\ \text{kg/m}^3)(0.000625\ \text{m}^3)$$
$$= \underline{0.902 = 90.2\%}$$