Engine Performance Parameters

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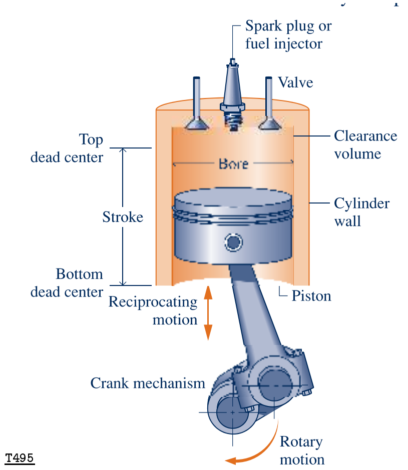
ME 417: Internal Combustion Engines

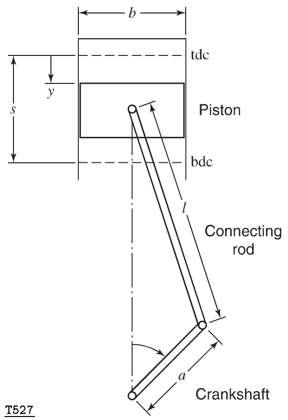
`http://zahurul.buet.ac.bd/ME417/`

ICE Nomenclature



T495

- $V_d \equiv$ displacement volume
  $V_d \equiv V_{BDC} - V_{TDC} = \frac{\pi}{4} b^2 s$
- $b \equiv$ bore
- $s \equiv$ stroke
- compression ratio, $r_c \equiv \frac{V_{BDC}}{V_{TDC}}$



T527

V_c	:	clearance volume [m^3]
y	:	instantaneous stroke [m]
V_θ	:	instantaneous volume [m^3]
$\bar{V}_p$	:	average piston speed [m/s]
N	:	engine rotational speed [rev/s]
$\dot{m}_f$	:	fuel consumption rate [kg/s]
$\dot{W}_b$	:	brake power [kW]
$bsfc$	:	brake sp. fuel consumption [kg/kW-hr]
Q_{LHV}	:	lower heating value of fuel [kJ/kg]
η_{th}	:	thermal efficiency [-]

- $y = l + a - \left[\sqrt{l^2 - a^2 \sin^2\theta} + a\cos\theta\right]$
- $V_\theta = V_c + \frac{\pi}{4} b^2 y$

$\Rightarrow$ $\boxed{\bar{V}_p == \frac{s}{\frac{1}{2}N} = 2 \cdot N \cdot s}$

$\Rightarrow$ $\boxed{bsfc = 3600 \left[\frac{\dot{m}_f}{\dot{W}_b}\right] \quad : \quad \eta_{th} = \frac{100}{bsfc \cdot Q_{LHV}} \%}$

Air-Standard Efficiency, η_{as}

$$\eta_{as,\ otto} = 1 - \frac{1}{r_c^{k-1}}$$

$$\eta_{as,\ diesel} = 1 - \frac{1}{r_c^{k-1}} \left[\frac{\beta^k - 1}{k(\beta - 1)}\right]$$

- $r_c \equiv$ compression ratio
- $\beta \equiv$ cut-off ratio

§ η_{as} is not same as actual efficiency, because:

1. working fluid is not air:
   - intake & compression stroke: air (+ fuel)
   - Power & exhaust stroke: mix of N_2, O_2, CO_2, CO, H_2O, NO ...
   - $k = 1.3 \pm 0.05$, rather than 1.4 for air.
2. significant heat loss from engine cylinder
3. combustion is not instantaneous

Real Gas Efficiency, η_o

§ Real gas efficiency, η_o takes into account of the effects of the properties of real gases, but ignores heat loss and combustion delay effects. Mixture composition during power and exhaust stroke is a function of temperature and is varying.

η_o/η_{as}
- ∽ 0.69, for CFR (Cooperative Fuel Research) engine, $\phi = 1.0$,
- ∽ primarily a function of ϕ, as it changes k,
- ∽ very weak function of compression ratio,
- ∽ essentially independent of inlet & exhaust conditions.

Indicated Efficiency, η_i

$$\eta_i \equiv \frac{\text{Indicated Power}}{\text{Heat Input Rate}} = \frac{\dot{W}_i}{\dot{m}_f Q_{LHV}} \times 100\%$$

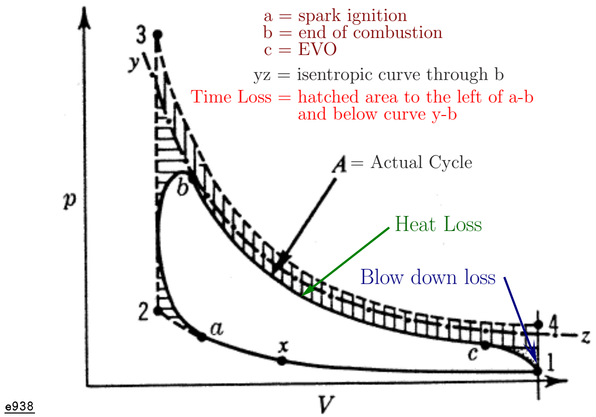
- actual, real, but measured at the cylinder, not at the flywheel,
- does not contain mechanical efficiency, η_m which accounts:
  - energy expended in pumping the gases into & out of cylinder,
  - driving some accessories,
  - turning the cam-shaft & crank-shaft,
  - rubbing the piston rings up & down against cylinder walls.
- $\eta_i/\eta_o = 0.86$: for CFR engine, $\phi = 1.13$

⇒ $\eta_i/\eta_{as} = 0.69 \cdot 0.86 = 0.59$

► η_i/η_o: primarily a function of ϕ, weak function of combustion chamber, surface/volume ratio influencing heat loss.

A More Realistic Cycle

$$\dot{W}_i = \oint P dV$$



Effects of real intake & exhaust strokes are not considered; these are normally included in the mechanical efficiency.

- **Time Loss**: Combustion is not instantaneous. Combustion is started before TDC and continues substantially past TDC. Usually, greatest efficiency is obtained when the point of ignition and the point at which combustion is completed are roughly symmetric.

$$\text{Time Loss} \sim 30\% \text{ of } (\eta_i - \eta_o)$$

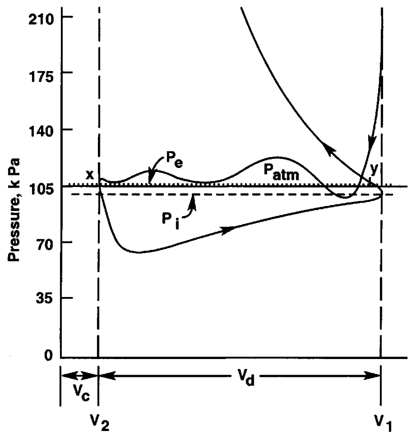
- **Heat Loss**: loss during compression is negligible.

$$\text{Heat Loss} \sim 60\% \text{ of } (\eta_i - \eta_o)$$

- **Blow-down Loss**: the difference between the ideal and actual cycle at EVO represents unavailable work.

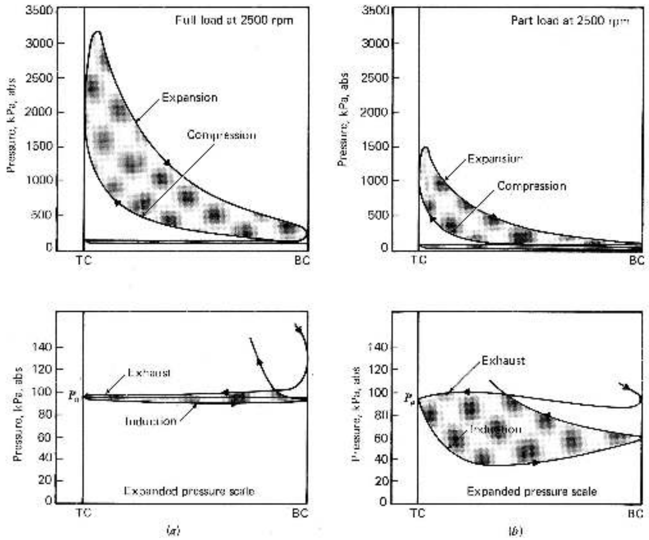
$$\text{Blow by Loss} \sim 10\% \text{ of } (\eta_i - \eta_o)$$

Other Losses



T530

Inlet/exhaust part of the indicator diagram at partial throttle



T522

Indicator diagram of a 4s SI engine a) WOT, b) part-load

1. Pumping Losses

- negligible at WOT (wide open throttle) condition.
- become a substantial fraction of the total at partial throttle when the manifold vacuum is high (idle condition).
- at idle, the engine produce no net work, the work produced by the cylinder must balance the mechanical losses (pumping, accessories, friction etc.)

2. Losses Due to Unburned Fuel

- because of poor mixing control, either within cylinder or from cylinder to cylinder, some of the mixture is not burned.
- some of the charge next to the cylinder wall, crevices around the spark plug and the valves are chilled, and will not burn.

⇒ loss of energy and HC pollution.

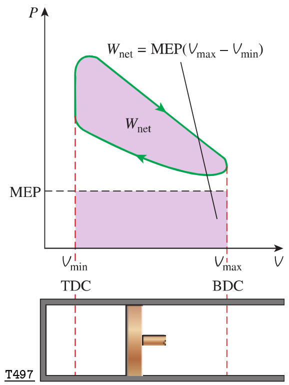
$$\eta_c \approx \begin{cases} 0.98 & \text{if } \phi \leq 1 \\ \frac{1}{\phi} & \text{if } \phi > 1 \end{cases}$$

3. Leakage Losses

- Piston rings and sometimes the valves, do not seal properly and consequently the cylinder pressures do not rise as high as they should, represents a loss.
- For an engine in good condition: leakage loss is negligible.
- Leakage losses are less serious at higher engine speeds, as it takes finite time for fluid to leak through an orifice or obstruction.
- Many racing engines use only one piston ring in order to reduce the friction at higher piston speeds, the resulting leakage not being a problem at operating speeds.

Mean Effective Pressures

$$\boxed{\dot{W} = mep \cdot V_d \cdot \frac{N}{X} = mep \cdot (A_p \cdot s \cdot N_{cyl}) \cdot \frac{N}{X}}$$



T497

$$X = \begin{cases} 1 & \text{for 2s engine} \\ 2 & \text{for 4s engine} \end{cases}$$

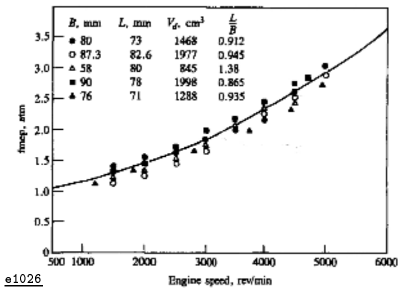
- $\dot{W}_b$: power is measured at flywheel.
- $\dot{W}_i$: power developed within cylinder.

$\Rightarrow$ $\boxed{\dot{W}_b = \dot{W}_i - \dot{W}_f \Rightarrow bmep = imep - fmep}$

$\Rightarrow$ $\boxed{\eta_m = \frac{\dot{W}_b}{\dot{W}_i} \times 100\%}$

$$bmep = \begin{cases} 0.9 - 1.1 \text{ MPa} & \text{for SI engine, WOT} \\ 0.7 - 0.9 \text{ MPa} & \text{for CI engine} \end{cases}$$

Friction Mean Effective Pressure (fmep)



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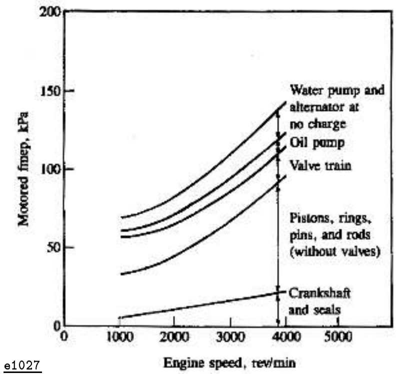


e1027

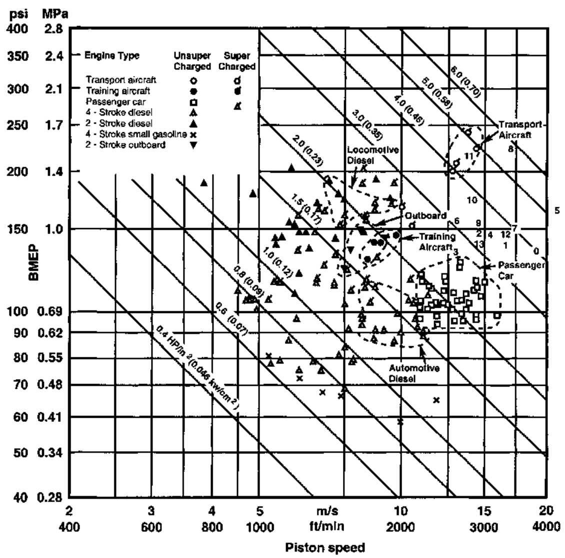
$$fmep(bar) = 0.97 + 0.15\left(\frac{N}{1000}\right) + 0.05\left(\frac{N}{1000}\right)^2$$

Engine Brake Power, $\dot{W}_b$

- Piston Speed, $\overline{V}_P = 2 \cdot N \cdot s$: comparable among engines.
- Maximum gas flow in engine is limited by the sonic velocity in the valve aperture ~ related to $\overline{V}_P$.
- specific power $\equiv \frac{\dot{W}_b}{A_p N_{cyl}} = bmep \frac{\overline{V}_P}{2X}$

$$\boxed{\dot{W}_b = bmep \cdot A_P \cdot N_{cyl} \cdot \frac{\overline{V}_P}{2X} \qquad \frac{\dot{W}_b}{V_d} = \frac{bmep}{s} \cdot \frac{\overline{V}_P}{2X}}$$

- for same *bmep* and piston speed: $s \downarrow \Rightarrow \dot{W}_b/V_d \uparrow$, an engine with shorter stroke will have a higher power per unit displacement. So, recent trend is reduction in stroke:bore ratio.
- Now-a-days:
  - $bmep = 1.1$ MPa, $\overline{V}_P = 15$ m/s, $s/b \sim 0.92$
  - specific power ~ 0.38 kW/cm^2; $\dot{W}_b/V_{disp} \sim 48$ kW/L.
  - For similar *specific power* & *piston speed*, the *bmep* for a four-stroke engine will be twice the *bmep* for a two-stroke engine.



T1352

Engine Power Equation

$$\eta_i \equiv \frac{\dot{W}_i}{\dot{Q}_in} = \frac{\dot{W}_i}{\eta_c \cdot \dot{m}_f \cdot Q_{LHV}}$$
$$\eta_m \equiv \frac{\dot{W}_b}{\dot{W}_i}$$
$$\eta_v \equiv \frac{m_a}{\rho_{a,i} V_d} = \frac{\dot{m}_a}{\rho_{a,i} V_d}\frac{X}{N}$$
$$F/A \equiv \dot{m}_f / \dot{m}_a$$

$$\begin{aligned}\dot{W}_b &= \eta_m \cdot \dot{W}_i \\ &= \eta_m \cdot \eta_i \cdot \eta_c \cdot \dot{m}_f \cdot Q_{LHV} \\ &= \eta_m \cdot \eta_i \cdot \eta_c \cdot \dot{m}_a \cdot (F/A) \cdot Q_{LHV} \\ &= \eta_m \cdot \eta_i \cdot \eta_c \cdot \eta_v \cdot \frac{N}{X} \cdot V_d \cdot \rho_{a,i} \cdot (F/A) \cdot Q_{LHV}\end{aligned}$$

$$\boxed{bmep = \eta_i \cdot \eta_c \cdot \eta_m \cdot \eta_v \cdot \{\rho_{a,i} \cdot (F/A) \cdot Q_{LHV}\}}$$

- $\rho_{a,i} \cdot (F/A) \cdot Q_{LHV} = bmep$ for $\eta_i = \eta_c = \eta_m = \eta_v = 1.0$: a value of *bmep*, far above anything practically attainable, as it would correspond to efficiencies of unity.
- For gasoline: $\rho_{a,i} = 1.2\ \mathrm{kg/m^3}, Q_{LHV} = 42.7\ \mathrm{MJ/kg}, (F/A) = 1/14.7 \Longrightarrow \rho_i \cdot (F/A) \cdot Q_{LHV} = 3.44\ \mathrm{MPa}$ ◀
- For methane: $\rho_{a,i} = 1.2\ \mathrm{kg/m^3}, Q_{LHV} = 50.01\ \mathrm{MJ/kg}, (F/A) = 1/17.12 \Longrightarrow \rho_i \cdot (F/A) \cdot Q_{LHV} = 3.51\ \mathrm{MPa}$ ◀.

  ▶ $\eta_{v,methane} < \eta_{v,gasoline}$, hence less output power is expected from methane fuelled engines.

Effect of ϕ

- At full power, engines operate rich, upto perhaps $\phi = 1.2$: η_i falls as properties of the gases are not the same, η_c falls as not all of the fuel is burnt.

$$\frac{\eta_i \eta_c}{\eta_{as}} = 0.921 - 0.327\phi \quad : \text{CFR engine}$$

- $\frac{\eta_i \eta_c}{\eta_{as}} = \begin{cases} 0.594 & \text{for } \phi = 1.0 \\ 0.528 & \text{for } \phi = 1.2 \end{cases}$
- CFR engine, $r_c = 8,\ \eta_v = 0.83,\ \eta_m = 0.85$

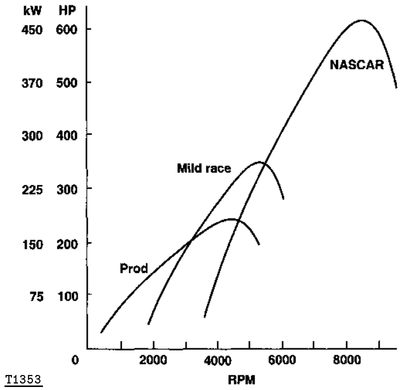
$$\Longrightarrow bmep = \begin{cases} 0.815\ \mathrm{MPa} & \text{for } \phi = 1.0 \\ 0.872\ \mathrm{MPa} & \text{for } \phi = 1.2 \end{cases}$$

  ▶ 20% more fuel $\leadsto$ 7% more power.
- If inlet air is warm and heating in the manifold: density $\sim$ 0.86 of the ambient value $\Rightarrow bmep = 0.75$ MPa.

Racing Engine Design

- Exotic fuel: 5% nitrobenzene + 80% methanol + 15% acetone.
- Two important factors:
  1. High latent heat of evaporation, 1.1 MJ/kg $\Rightarrow$ chilling of charge.
  2. Very high resistance to knock.
  - $r_c = 16 \rightarrow \eta_{as} = 0.67, \phi = 1.2 \rightarrow \eta_i \eta_c / \eta_{as} = 0.528 \Rightarrow \eta_i \eta_c = 0.354$.
  - $Q_{LHV} = 20.9\ \mathrm{MJ/kg}, (F/A) = 0.15 \Rightarrow \rho_i \cdot (F/A) \cdot Q_{LHV} = 4.51\ \mathrm{MPa}$

$$\Longrightarrow bmep = \begin{cases} 1.1\ \mathrm{MPa} & \\ 1.4\ \mathrm{MPa} & \text{with tuning} \end{cases}$$

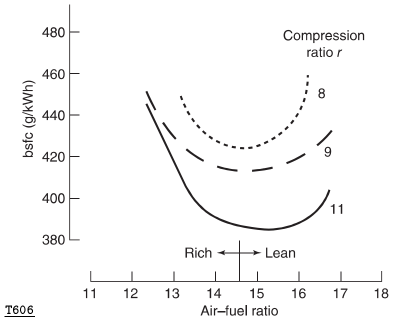


T1353

	Production Engine	Mild Race	NASCAR
IO CAD BTC	3	20	60
IC CAD ABC	47	50	70
EO CAD BBC	47	60	90
EC CAD ATC	3	10	40
P_b kW (HP)	179 (240)	261 (350)	447 (600)
N rpm	4500	5500	8500

T1354

Effects of ϕ & r_c (SIE)



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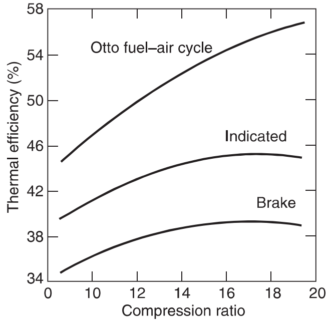
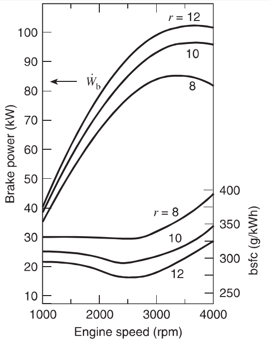
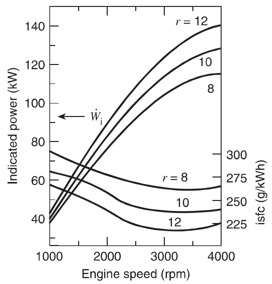


T605

- Indicated specific fuel consumption (*isfc*) improves at a faster rate with increasing compression ratio than the *bsfc*, because both friction and heat losses are increasing with compression ratio.
- Hence, there is an optimum compression ratio due to these effects. The spark advance is set for best efficiency at $\phi = 0.91$.

Effects of Engine Speed

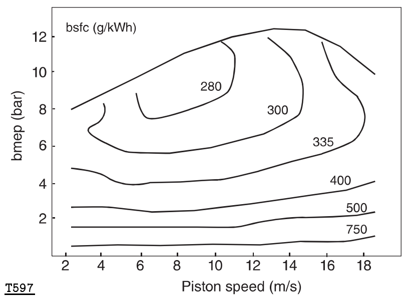




T596

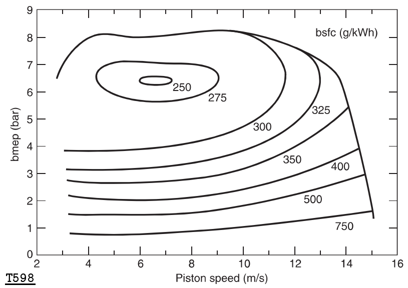
- *isfc* decreases with increasing N_{rpm} and then levels out because % of heat loss to the coolant decreases with increasing N_{rpm}.
- *bsfc* is fairly flat or decreasing at low speeds and is increasing at higher speeds. At higher speeds, friction & pumping losses become more significant, as they increase quadratically with engine speed.

Comparison of SIE & CIE Performances



T597

SI Engine



T598

CI Engine

- bsfc's of SIE are higher.
- bmep's of SIE are higher.

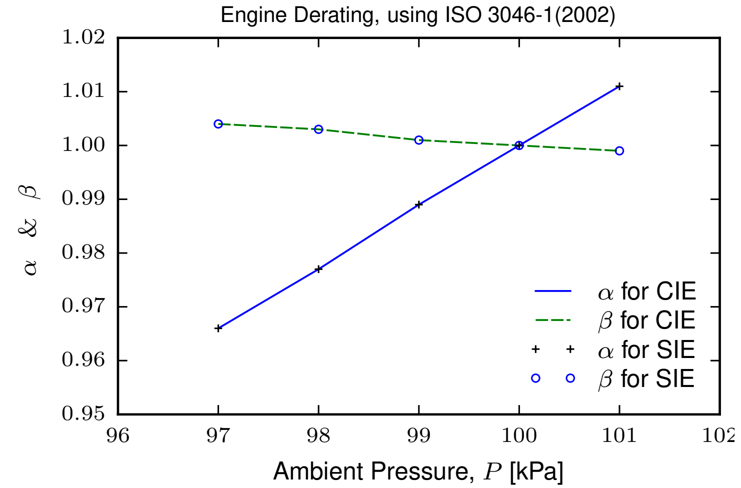
Engine De-rating: ISO 3046 [2002]

⇒ $P_x = \alpha \ P_r$

⇒ $bsfc_x = \beta \ bsfc_r$

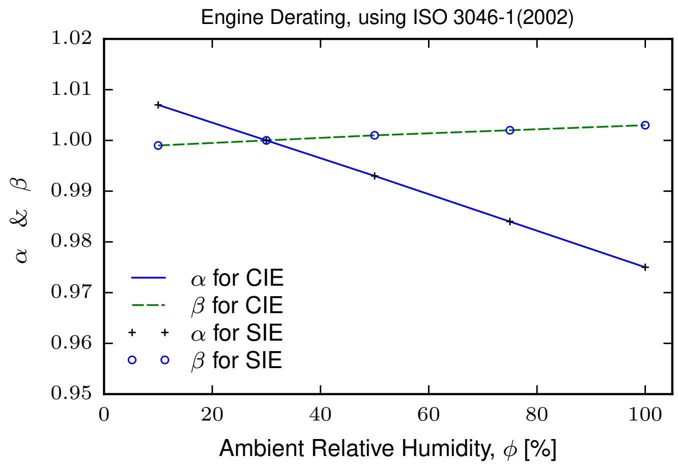
- **x:** lab condition, **r:** ref. condition
- P_x: power at lab condition
- $bsfc_x$: bsfc at lab condition
- $P_r = 100 \ kPa, \ \ T_r = 298 \ K, \ \ \phi_r = 30\%$
- $\alpha \equiv$ power adjustment factor
- $\beta \equiv$ fuel consumption adjustment factor

Effect of Ambient Pressure



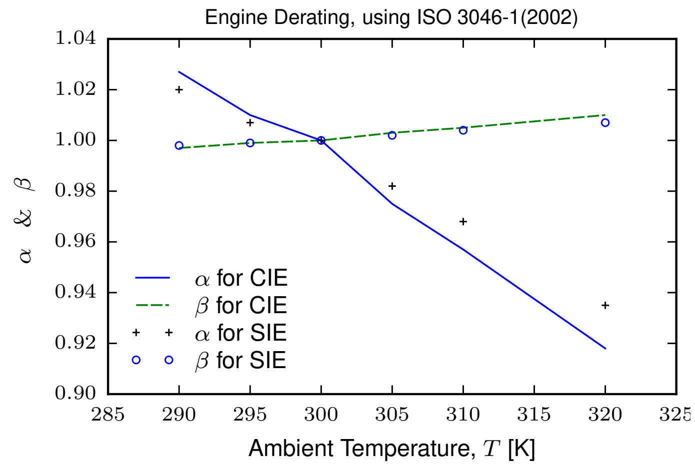
T610

Effect of Relative Humidity



T611

Effect of Ambient Temperature



T612

Comparison of CI and SI Engine Performances

Fuel-injection system	Rated speed n_{rated} [rpm]	Compression ratio ε	Mean pressure [1] p_e [bar]	Specific power output $p_{e,\,spec.}$ [kW/l]	Power-to-weight ratio $m_{spec.}$ [kg/kW]	Specific fuel consumption [2] b_e [g/kWh]
Diesel engines						
IDI[3] conventionally aspirated car engines	3,500...5,000	20...24:1	7...9	20...35	1:5...3	320...240
IDI[3] turbocharged car engines	3,500...4,500	20...24:1	9...12	30...45	1:4...2	290...240
DI[4] conventionally aspirated car engines	3,500...4,200	19...21:1	7...9	20...35	1:5...3	240...220
DI[4] turbocharged car engines with i/clr[5]	3,600...4,400	16...20	8...22	30...60	4...2	210...195
DI[4] convent. aspirated comm. veh. engines	2,000...3,500	16...18:1	7...10	10...18	1:9...4	260...210
DI[4] turbocharged comm. veh. engines	2,000...3,200	15...18:1	15...20	15...25	1:8...3	230...205
DI[4] turboch. comm. veh. engines with i/clr[5]	1,800...2,600	16...18	15...25	25...35	5...2	225...190
Construct. and agricultural machine engines	1,000...3,600	16...20:1	7...23	6...28	1:10...1	280...190
Locomotive engines	750...1,000	12...15:1	17...23	20...23	1:10...5	210...200
Marine engines (4-stroke)	400...1,500	13...17:1	18...26	10...26	1:16...13	210...190
Marine engines (2-stroke)	50...250	6...8:1	14...18	3...8	1:32...16	180...160
Gasoline engines						
Conventionally aspirated car engines	4,500...7,500	10...11:1	12...15	50...75	1:2...1	350...250
Turbocharged car engines	5,000...7,000	7...9:1	11...15	85...105	1:2...1	380...250
Comm. veh. engines	2,500...5,000	7...9:1	8...10	20...30	1:6...3	380...270

T613