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School of Engineering

EEN-E2002, Internal Combustion Air Management

Basshuysen Chapter 11 Supercharging of Internal Combustion Engines
Heywood Chapter 6 Gas exchange process

January 2019, Martti Larmi

Gas Exchange in 4-Stroke Engines

- Exhaust and intake strokes = gas exchange

- NON SUPERCHARGED = NATURALLY ASPIRATED ENGINES
- Gas exchange starts at exhaust valve opening. A high pressure pulse (blow down pulse) enters the exhaust channel and exhaust manifold and cylinder pressure starts rapidly to fall down.
- There is an over pressure in the cylinder over the whole exhaust stroke while piston pushes residual gases away from the cylinder.

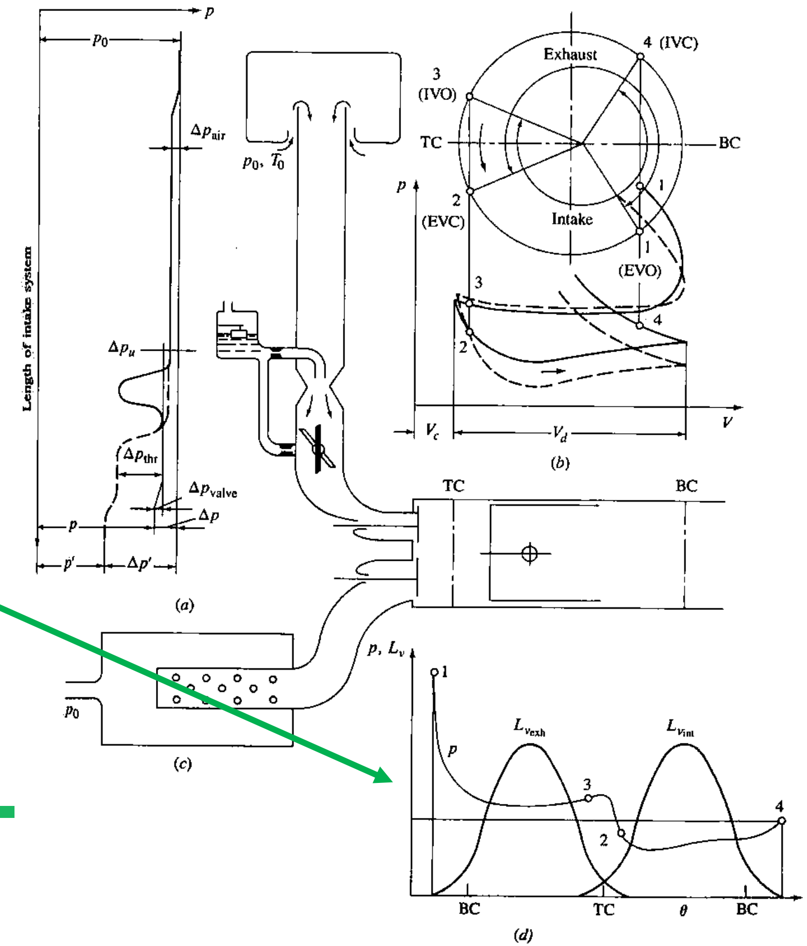
Gas Exchange in 4-Stroke Engines

- The exhaust valve is opened before the bottom dead center in order to have the valves open enough when piston starts to move up. The exhaust valve is closed after the top dead centerin order not to choke the flow too early. Heywood Fig. 6-1 (d).
- The piston needs to make work on the gases during gas exchange. Especially in SI engines with stoichiometric combustion the gas exchange work is remarkable at part load = pumping losses. The intake air flow is throttled and there is a remarkable underpressure in the intake manifold.

Gas Exchange in 4-Stroke Engines

Note the pressure graph with valve lift curves

Pressure differences are the driving force in gas exchange



Valve Overlap

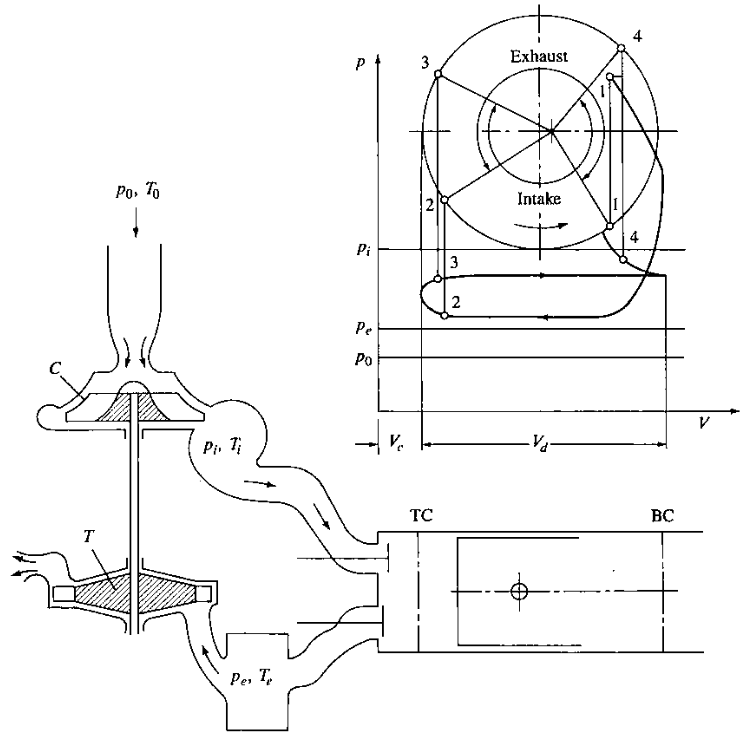
During valve overlap period one must take care of valve and piston clearance. Valves may not collide with the piston. Some pistons have so called valve pockets if compression ratio is high and large overlap perod. Turbocahrged engine typically habe larger overlap than naturally aspirated engine to ease the operation of compressor at low engine speed.

Negative overlap tells about special valve timing, where exhaust valves are closed before top dead center and intake valves are opened after top dead center. This is made deliberately to leave some of the residual gases in the cylinder. This is also called Internal Exhaust Gas Recirculation, Internal EGR). This is made to reduce combustion temperatures to reduce NOx formation.

Turbocharged Engines

During intake stroke there is over pressure in the intake manifold or scavenging air receiver. At high load and high charge air pressure the gas work towards piston might even be positive.

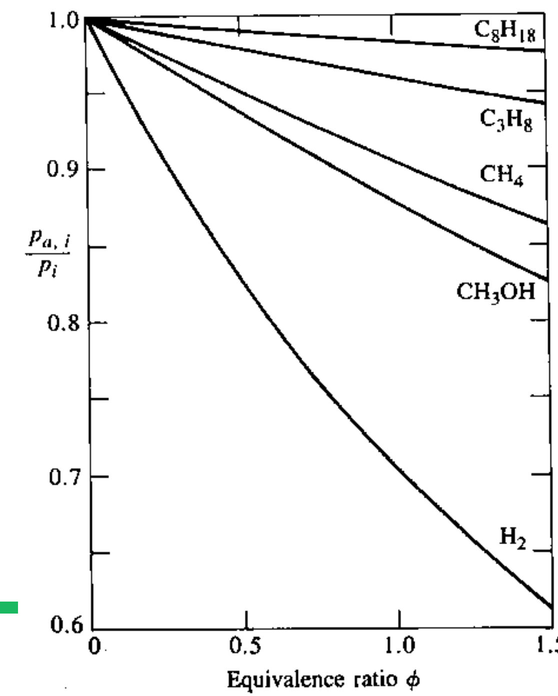
At part load however, there may be some backflow during gas exchange.



Volumetric Efficiency, Heywood 6-9

- If the fuel air mixture i.e. charge is made in the intake manifold or in the intake channel, the volume fraction of the fuel (on molar basis) takes part of the cylinder volume. That reduced volumetric efficiency. (Kvasi-static effect) Gaseous fuels may lead up to 30% reduction of the volumetric efficiency.
- On the contrary, the liquid fuel evaporation may reduce the charge temperature which then increases the volumetric efficiency.
- Residual gases left in the cylinder also reduce the volumetric efficiency

$$\eta_v = \frac{m_a}{\rho V_d}$$

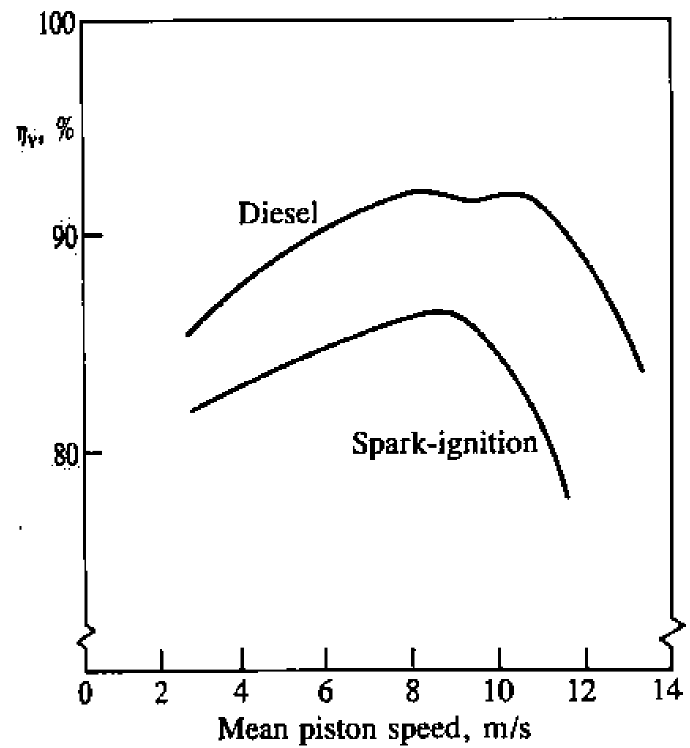


Volumetric Efficiency

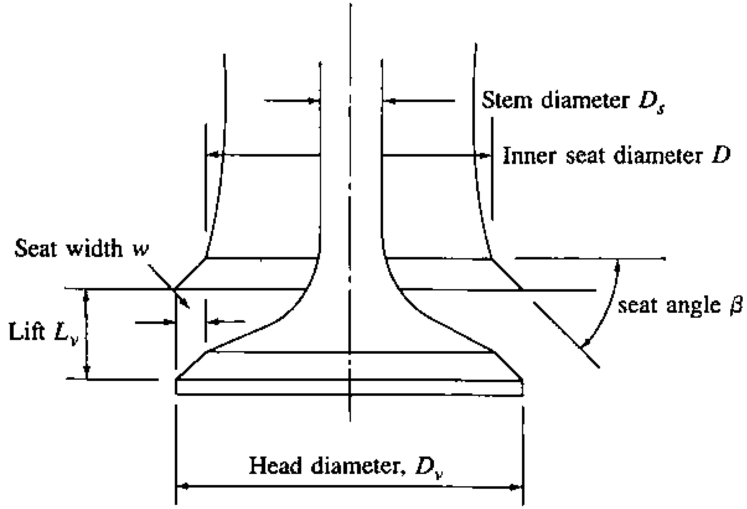
- The restricted (small) valve opening periods increase the low speed area volumetric efficiency and thus increase the low speed torque, as well.
- Large opening periods do increase the high speed volumetric efficiency and torque, but too large period make the low speed torque remarkably low.
- The valve lift increase makes the volumetric efficiency higher to some extent. Typically there is no increase of the flow cross section area over the lift 0.25*D, where D is the inner seat diameter.
- Flow phenomena and pressure pulses affect very much on volumetric efficincy

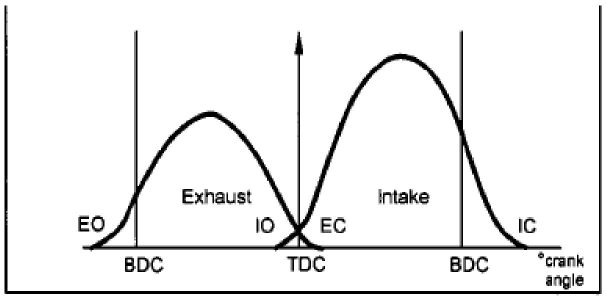
Volumetric efficiency CI vs. SI

- CI engine (diesel) volumetric efficiency at Wide Open Throttle (WOT). The double pulse is due to adjustable intake tuning.
- The SI engine volumetric efficiency is less due to the throttling of air flow, charge heat up, fuel evaporation and the large amount of residual gases in cylinder.



Poppet Valve Geometry and flow cross section

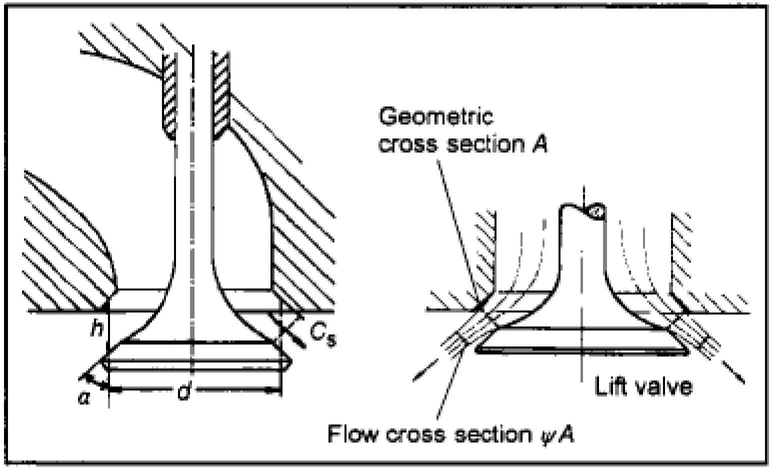




$$A_m = \pi L_v \cos\beta \left(D_v - 2w + \frac{L_v}{2}\sin 2\beta\right)$$

$$A_m = \pi D_m \left[(L_v - w\tan\beta)^2 + w^2\right]^{1/2}$$

$$A_m = \frac{\pi}{4}(D_p^2 - D_s^2)$$



Valve flow reference area

1. Inner Seat Diameter bases area, pi*D**2/4
2. Valve Curtain area pi*D*L
3. Real, physical cross section

Case one is the most suitable. The discharge coefficient Cd (or alpha or myy-sigma) starts from zero and strongly depends on L/D ratio. Reference area is constant in case 1. In Europe Cd is often replaced by the symbol $\mu\sigma$ (myy-sigma). Discharge coefficients should be measured anyway.

Flow equations, Basshuysen

$$\dot{m} = \dot{V} \cdot \rho = A_S \cdot c_S \cdot \rho = \psi \cdot A \cdot \varphi \cdot c_{iS} \cdot \rho \quad (10.20)$$

with

ρ = Density in the flow cross section
ψ = Jet contraction (constriction number)
φ = Friction coefficient

$$c_{iS} = \sqrt{\frac{2 \cdot \kappa}{\kappa - 1} \cdot R_L \cdot T_1 \cdot \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa - 1}{\kappa}}\right]} \quad (10.23)$$

and

$$\rho_{iS} = \rho_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}} \quad (10.24)$$

$\kappa = 1.4$ for air

$$\dot{m} = A_{\text{eff}} \cdot p_{01} \cdot \sqrt{\frac{2}{R \cdot T_{01}}} \cdot \psi \quad (10.28)$$

where

$$A_{\text{eff}} = \alpha \cdot \frac{d_{vi}^2 \cdot \pi}{4} \quad (10.29)$$

and the flow function ψ in the subsonic range is

$$\psi = \sqrt{\frac{\chi}{\chi - 1} \cdot \left[\left(\frac{p_2}{p_{01}}\right)^{\frac{2}{\chi}} - \left(\frac{p_2}{p_{01}}\right)^{\frac{\chi + 1}{\chi}}\right]} \quad (10.30)$$

and in the transonic range

$$\psi = \psi_{\max} = \left(\frac{2}{\chi + 1}\right)^{\frac{1}{\chi + 1}} \cdot \sqrt{\frac{\chi}{\chi + 1}} \quad (10.31)$$

Detailed equations can be found in Heywood Appendix C

Intake valve
flow coefficients

Myy-sigma
= alpha
= Cd

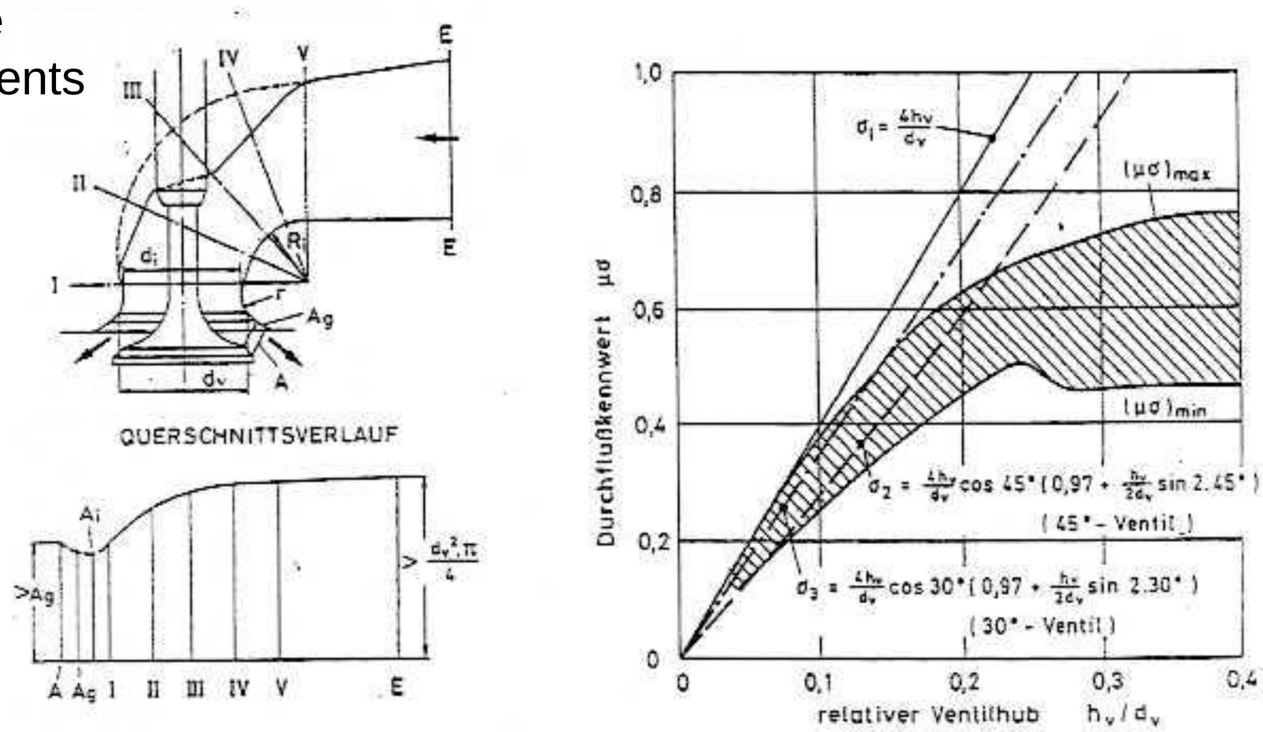


Abb. 3.27. Einlaßkanal; Formgebung (links) und Versperrungsziffern verschiedener Ventilsitze sowie Streuband der Durchflußkennwerte von 18 drallfreien Einlaßkanälen

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Pischinger: Thermodynamic der
Verbrennungskraftsmaschinen

Exhaust valve flow coefficients

Myy-sigma
= alpha
= Cd

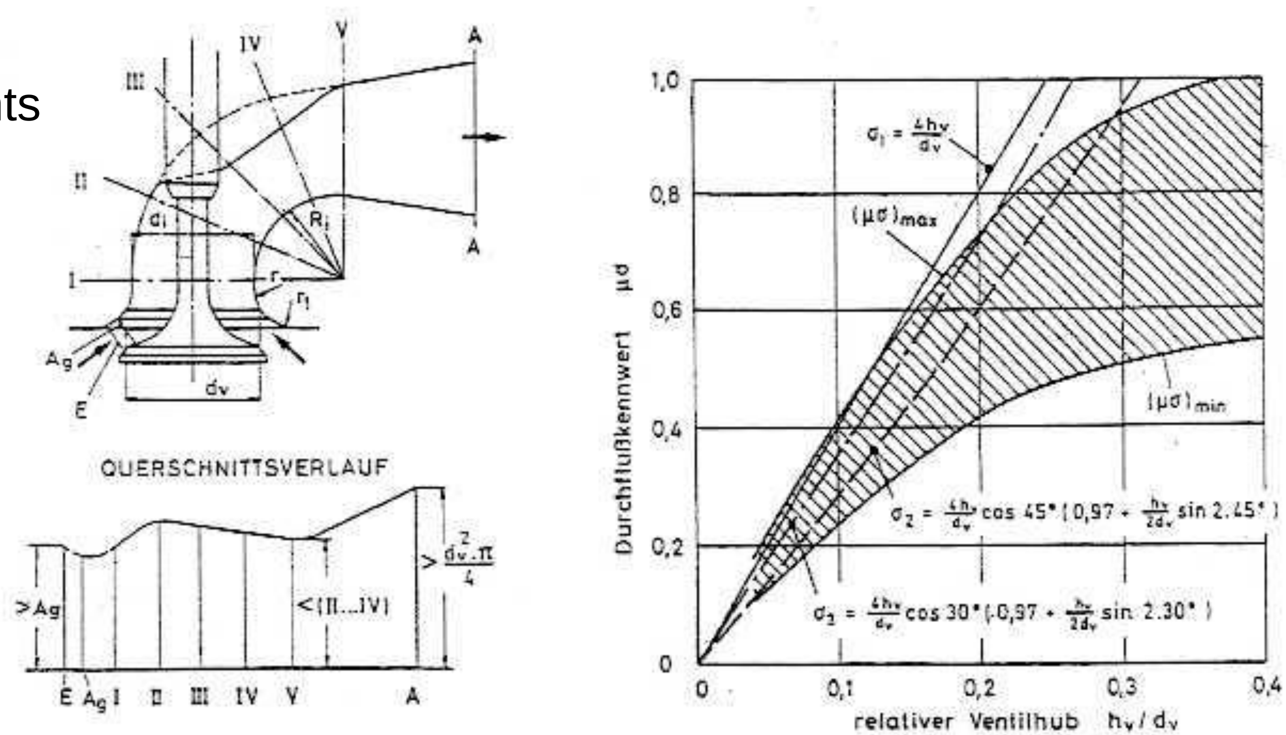

Abb. 3.28. Auslaßkanal; Formgebung (links) und Versperrungsziffern verschiedener Ventilsitze sowie Streuband der Durchflußkennwerte von 40 Einlaßkanälen

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Pischinger: Thermodynamic der Verbrennungskraftsmaschinen

Two-Stroke

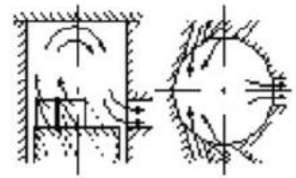
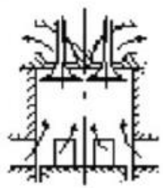
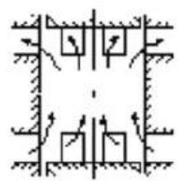
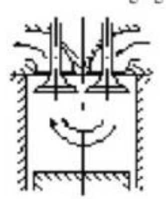
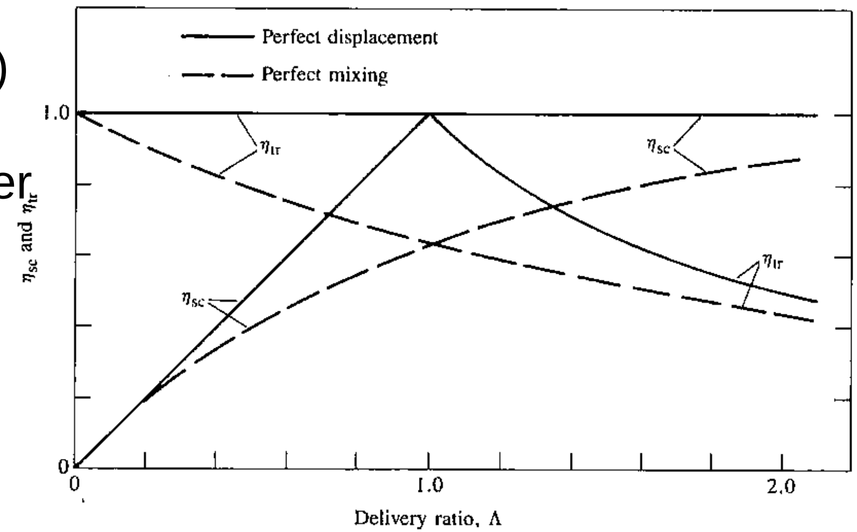
Scavenging approach	Advantages	Disadvantages
1. Loop scavenging 	• Compact design • High speeds are possible • The combustion chamber recess can be located in the cylinder head where it is well cooled • Simple design without a piston valve	• Asymmetrical timing diagram is possible only with additional components (piston valve) • Asymmetrical thermal load on the piston • The piston rings are especially endangered by the scavenging and exhaust ports • Comparatively difficult to generate charge turbulence
2. Uniflow scavenging with exhaust valves 	• Effective scavenging/low air expenditure • Easy to generate and influence the combustion chamber turbulence • The combustion procedure can largely be transferred to four-stroke engines • Asymmetrical timing diagram is possible without additional components	• Larger overall height in comparison to 1 • A more involved and optimized valve gear is required for large effective cylinder strokes and low consumption
3. Uniflow scavenging with opposed pistons 	• Minimization of the combustion chamber surfaces heated in the high-pressure phase • Asymmetrical timing diagram can be achieved only by controlling the piston edges • Effective scavenging/low air expenditure	• More involved construction • Larger overall height (overall width) • Extreme thermal load on the piston controlling the exhaust ports • A conventional combustion method cannot be used due to the arrangement of the nozzle holder/spark plug
4. Reversed head scavenging 	• Engine-transmission unit very similar to that of four-stroke engines • The piston rings are not endangered from scavenging and exhaust ports	• Low scavenging effect/large air expenditure • Because of the restricted opening time cross section, strong rise in the charge cycle work and consumption at higher speeds

Fig. 10-56 Comparison of different scavenging approaches.

Two stroke scavenging

Delivery ratio Λ (Heywood 6.20) is fresh air mass delivered divided by a reference mass (displacement * density at intake conditions)
Scavenging efficiency ηsc is the new fresh charge in the cylinder divided by the cylinder charge.
Trapping efficiency ηtr is the ratio of the new fresh air trapped in the cylinder to the air delivered.
Purity (does not include residual gas air)
Charging efficiency ηch is the ratio of the new fresh air trapped in the cylinder to the displacement * density at intake conditions



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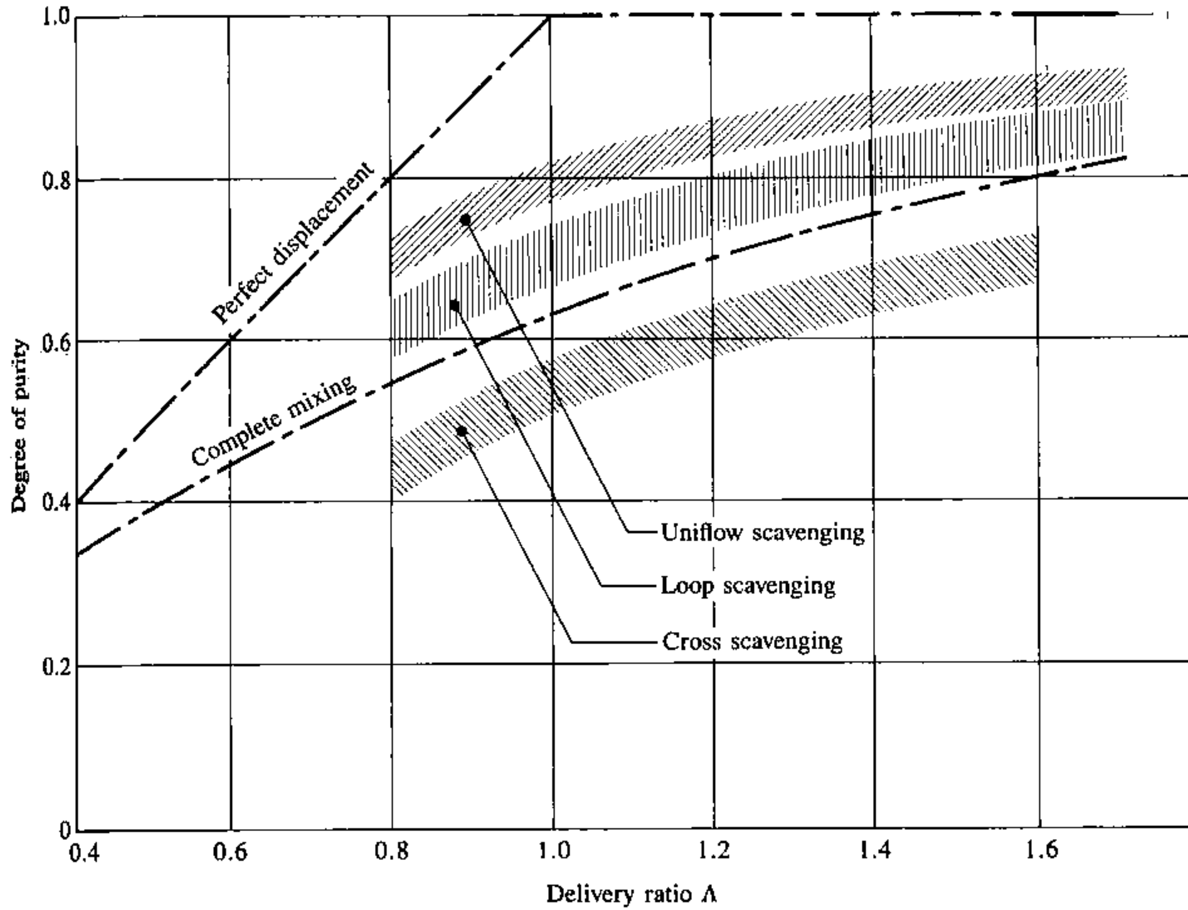
Real scavenging



Cam operates valves mechanically

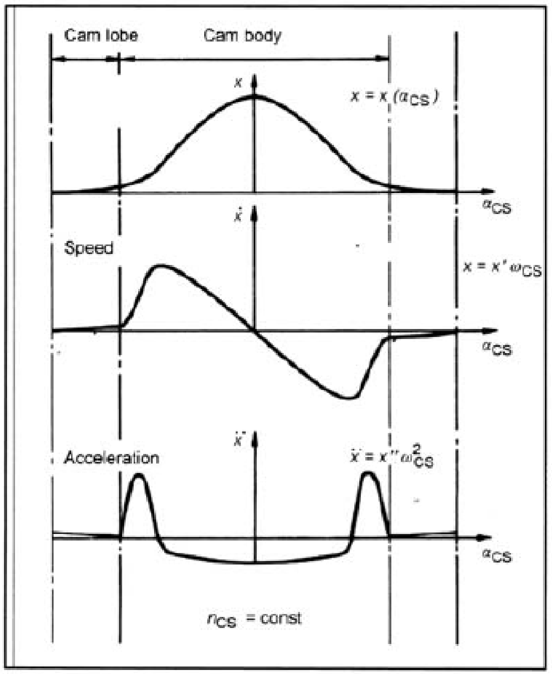
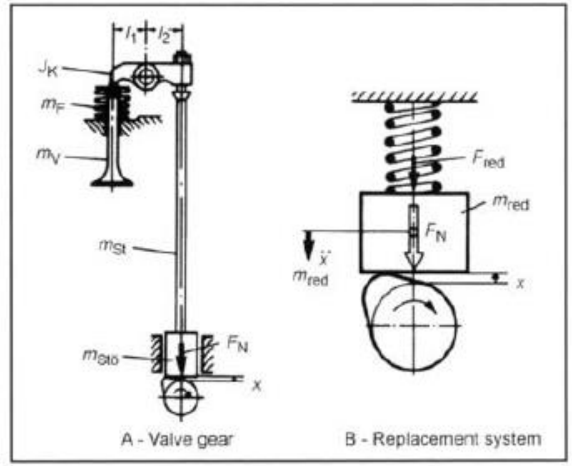


Fig. 10-22 Kinematics of the cams.[1]



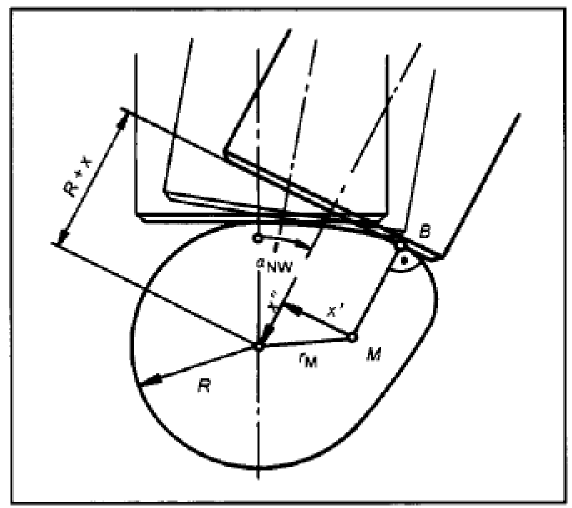


Fig. 10-23 Kinematics of the tappet stroke.

Variable lenght intake runner

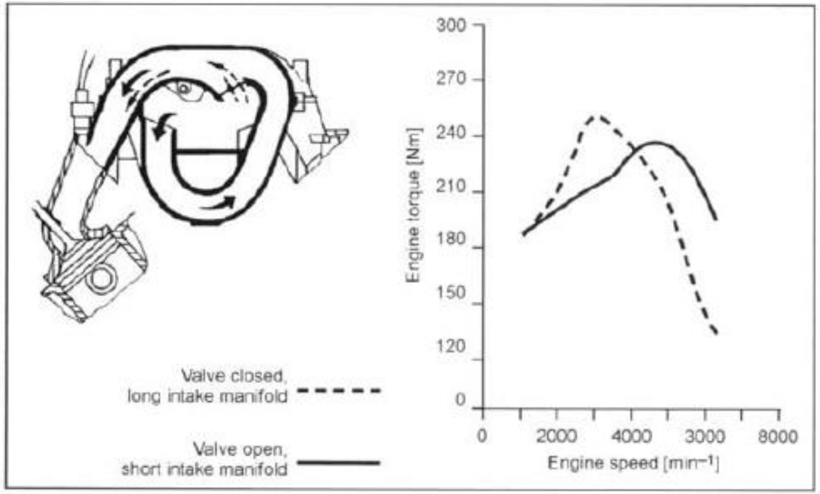


Fig. 10-32 Intake system with two-stage manifold; diagram (Audi V6).

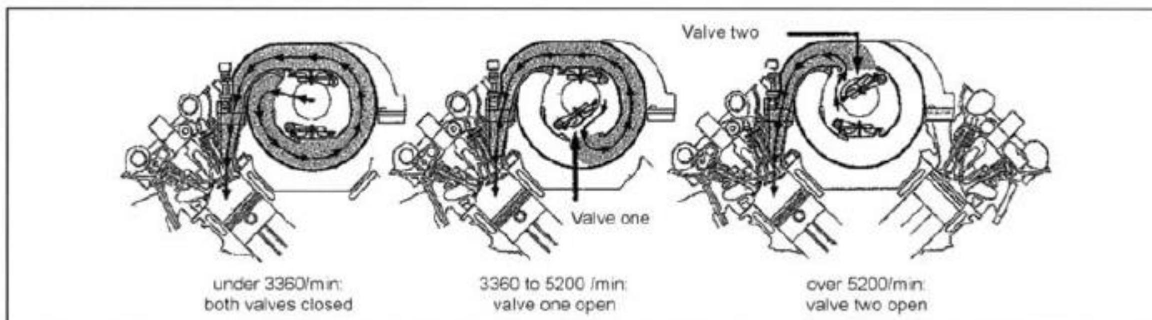


Fig. 10-33 Intake system with three-stage manifold.

Miller cycle

- Miller cycle, early intake valve closing (FES)
- Atkinsson cycle, late intake valve closing (SES)
- Increased thermal efficiency of the engine process, lower compression temperatures and lower peak temperature during combustion => reduced NOx formation
- To keep the same in-cylinder air amount (trapped mass) and BMEP, we do have to increase charge air pressure substantially.

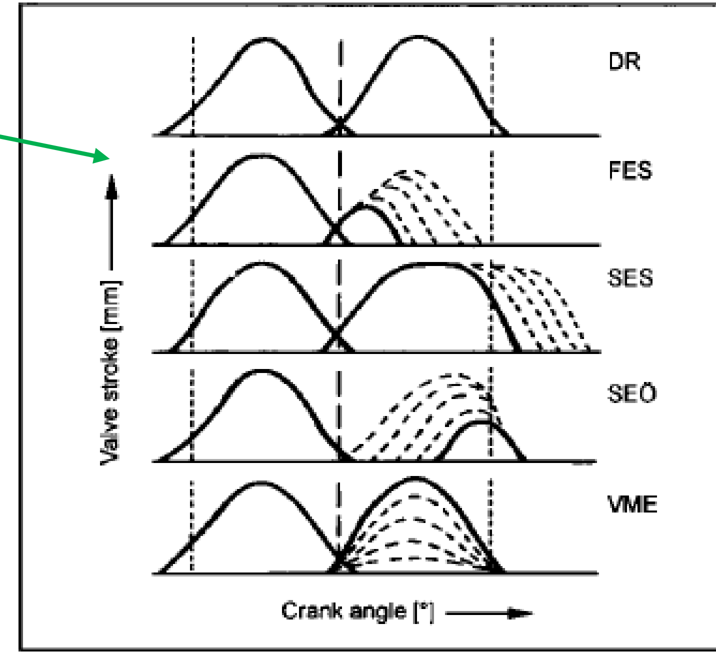


Fig. 10-62 Possibilities of adjusting the valve lifting curves with variable valve actuation.