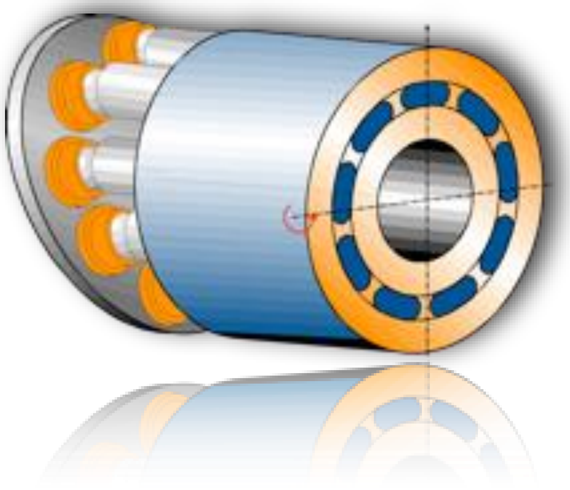
Αρχές λειτουργίας και τεχνολογία μηχανών θετικής μετατόπισης

Σ. Τσολάκης

ΕΜΠ
24/02/2020

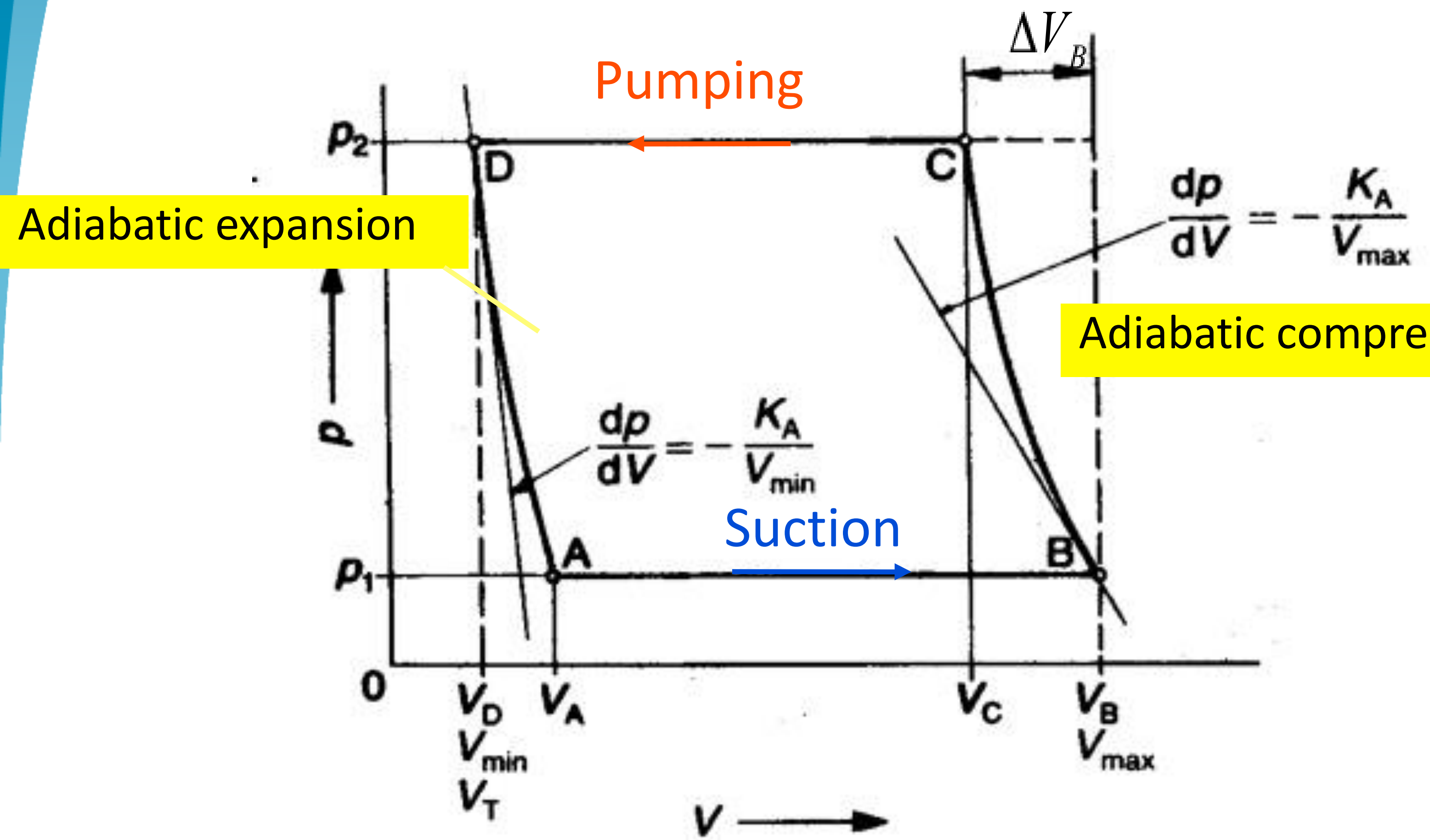
Περιεχόμενα

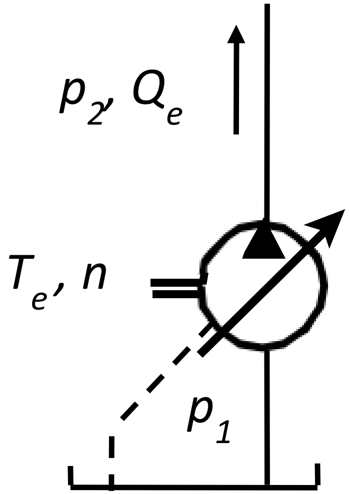
- Displacement machines – design principles & scaling laws
- Power density comparison between hydrostatic and electric machines
- Volumetric losses, effective flow, flow ripple, flow pulsation
- Steady state characteristics of an ideal and real displacement machine
- Torque losses, torque efficiency



Μηχανή θετικής μετατόπισης

due to compressibility of a real fluid





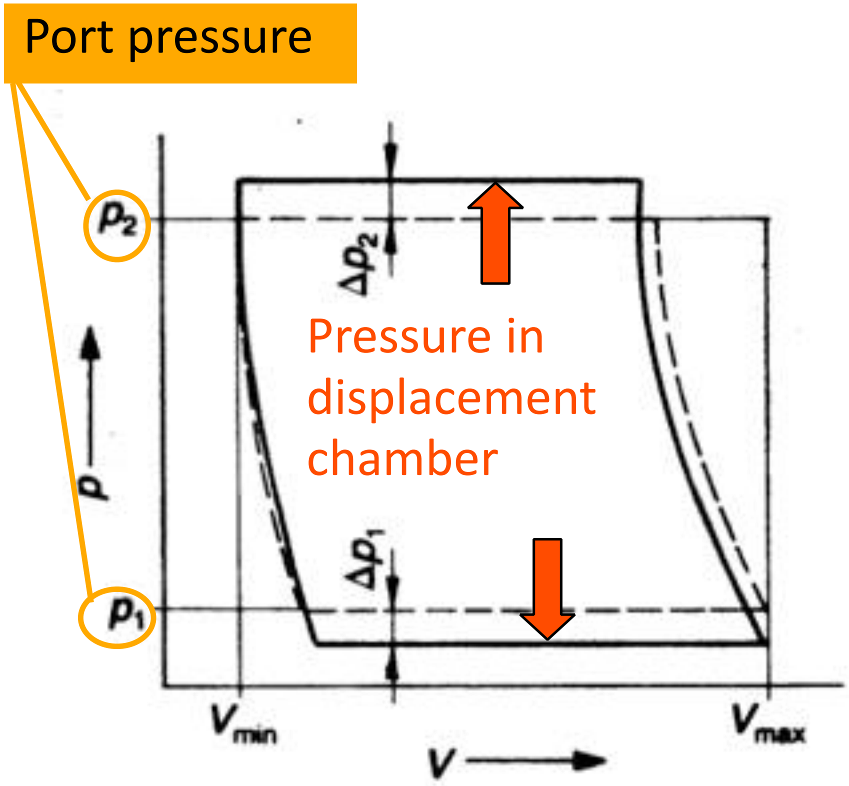
$V_{min}=V_T$ with V_T.. dead volume

K_A.. adiabatic bulk modulus

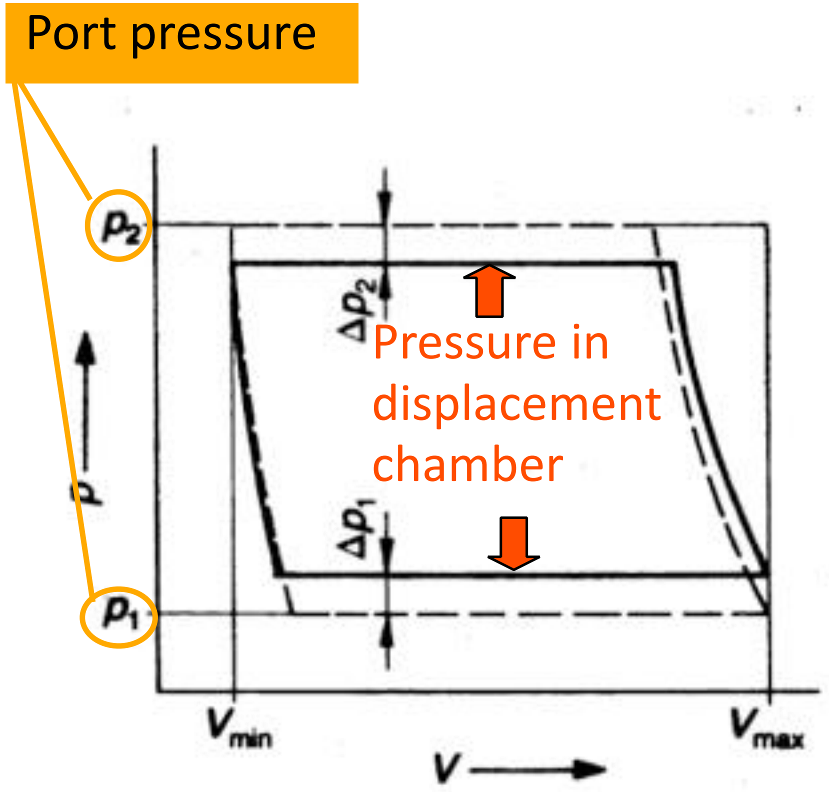
Μηχανή θετικής μετατόπισης

due to viscosity & compressibility of a real fluid

Pressure drop between displacement chamber and port



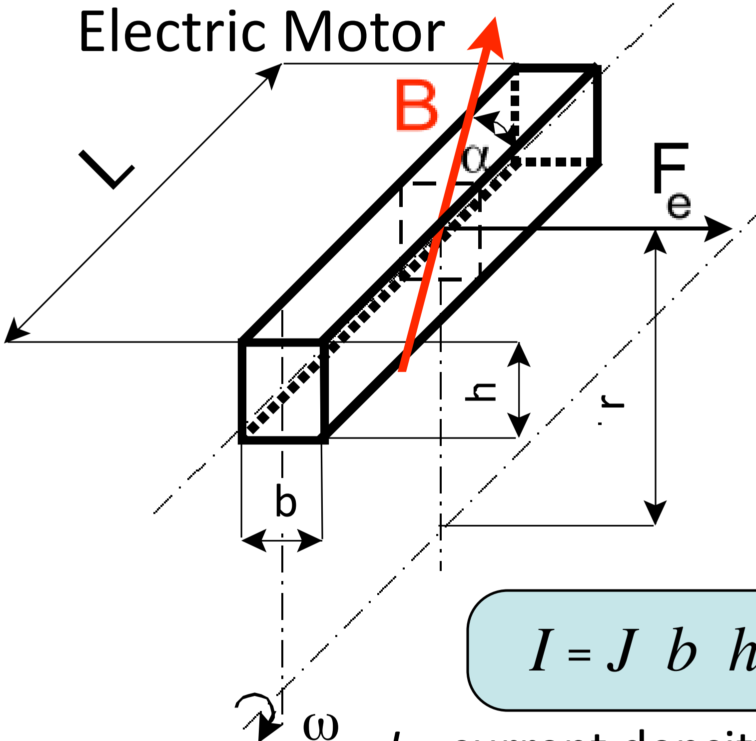
Pump



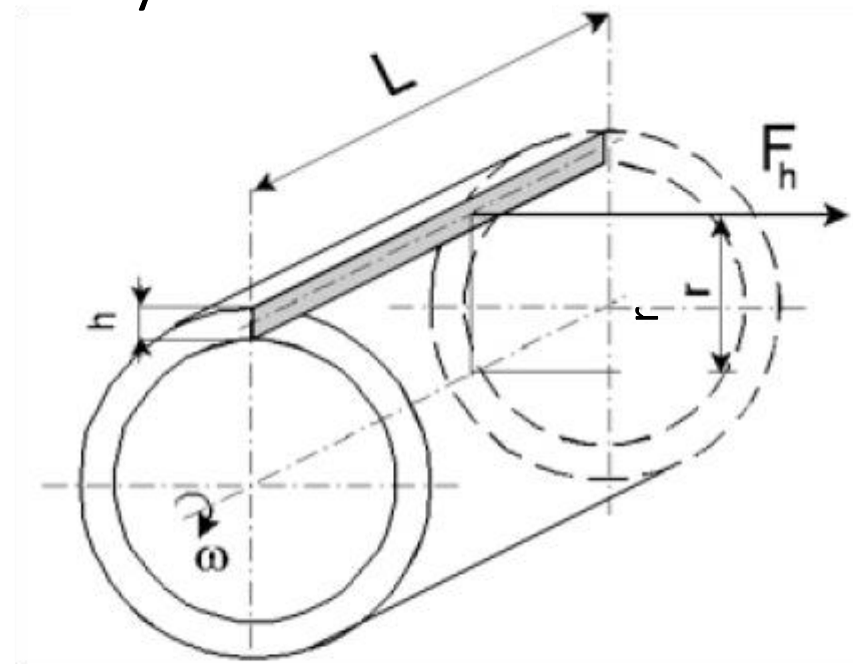
Motor

Συγκέντρωση ισχύος

Electric Motor



Hydraulic Motor



$$I = J \; b \; h$$

J… current density [A/m²]

$$F_e = I \cdot B \cdot L \cdot \sin\alpha$$

with *I* current [A]

$$F_h = p \cdot L \cdot h$$

B … magnetic flux density [T] or [Vs/m²]

Torque: $T = I \cdot B \cdot L \cdot r \cdot \sin\alpha$

$$T = p \cdot L \cdot h \cdot r$$

Παράδειγμα

Power: $P = T\,\omega = T\,2\pi\,n$

For electric motor follows: $P = I\,B\,L\,r\,2\pi\,n$ assuming α=90°

For hydraulic motor follows: $\mathrm{P} = \mathrm{p}\cdot\mathrm{L}\cdot\mathrm{h}\cdot\mathrm{r}\cdot 2\pi\cdot\mathrm{n}$

Force density:

Electric Motor

$$\frac{F_e}{L\cdot h} = \frac{J\cdot b\cdot h\cdot B\cdot L}{L\cdot h} = J\cdot b\cdot B$$

$$7.6\cdot 10^{6}\,\mathrm{A}\cdot\mathrm{m}^{-2}\cdot 1.8\,\mathrm{Vs}\cdot\mathrm{m}^{-2}\cdot 3\cdot 10^{-3}\,\mathrm{m} = 4.1\cdot 10^{4}\,\mathrm{Pa}$$

with a cross section area of conductor: $9\ 10^{-6}\ \mathrm{m}^2$

Hydraulic Motor

$$\frac{F_h}{L\cdot h} = p$$

up to $5\ 10^{7}$ Pa

Λόγος μάζας-ισχύος

	Electric Machine	Positive displacement machine
$\dfrac{\text{mass}}{\text{power}} =$	1 …. 15 kg/kW	0.1 … 1 kg/kW

→ Positive displacement machines (pumps & motors) are:

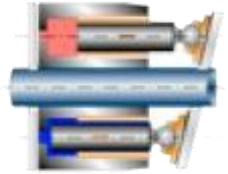
- 10 times lighter
- min. 10 times smaller
- much smaller mass moment of inertia (approx. 70 times)

↳ much better dynamic behavior of displacement machines

Μηχανές θετικής μετατόπισης

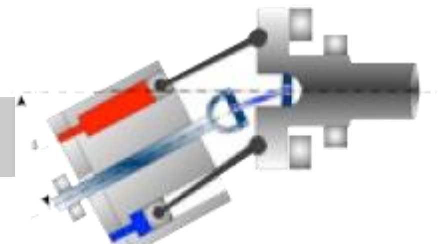
Piston Machines

Axial Piston Machines



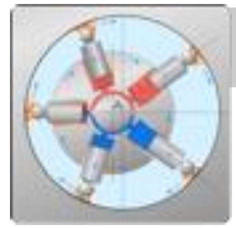
Swash Plate Machines

Bent Axis machines



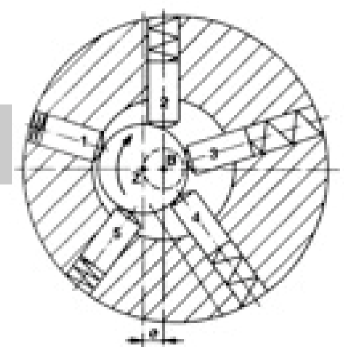
In-line Piston Machines

Radial Piston Machines

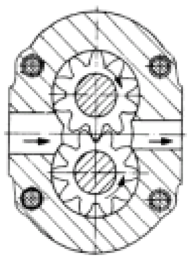


with external piston support

with internal piston support

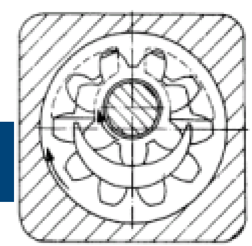


Gear Machines

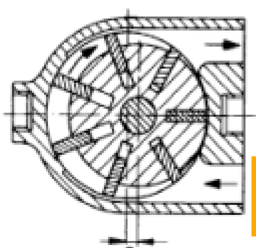


External Gear

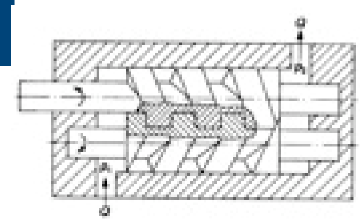
Internal Gear



Annual Gear



Vane Machines

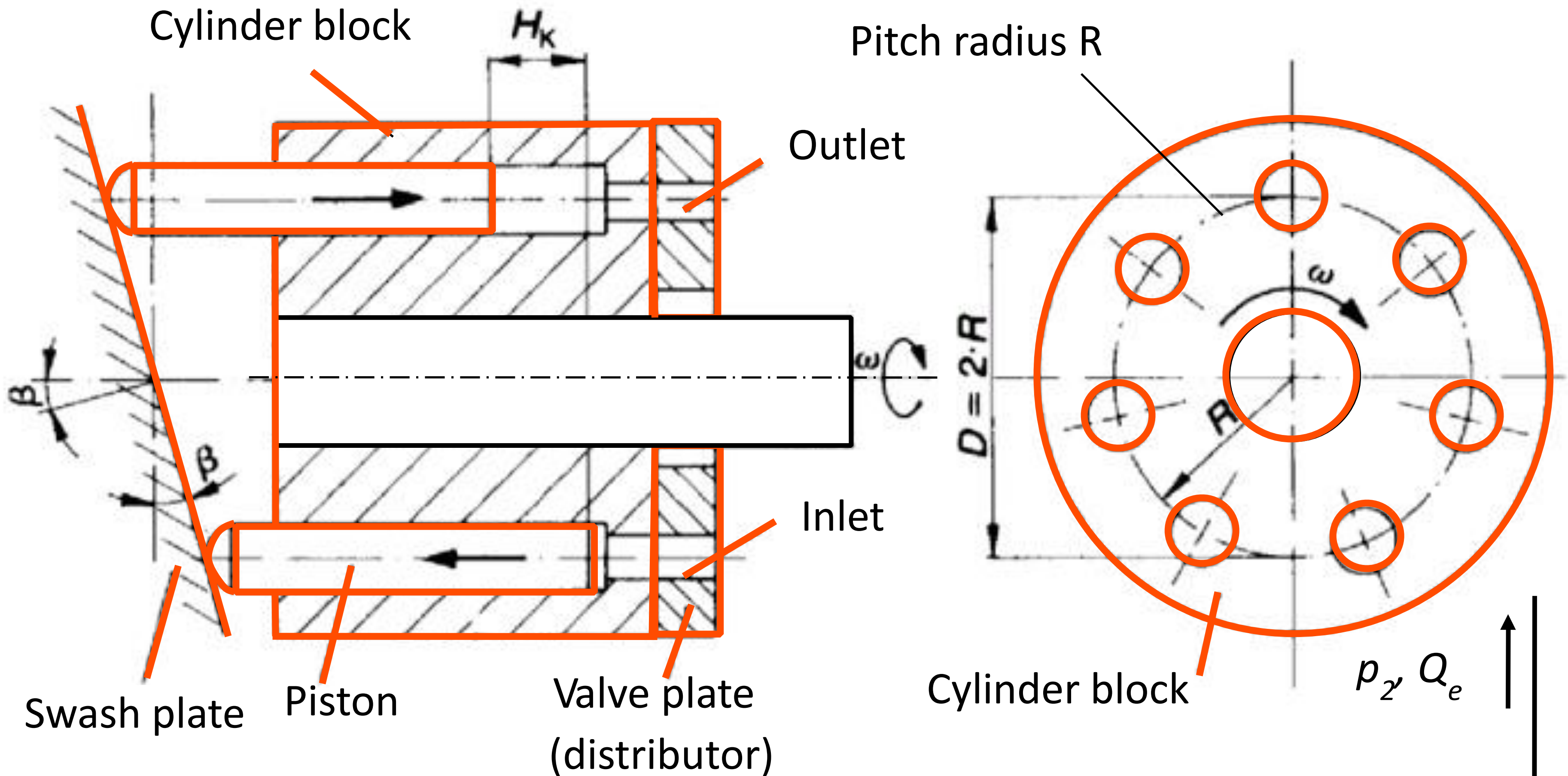


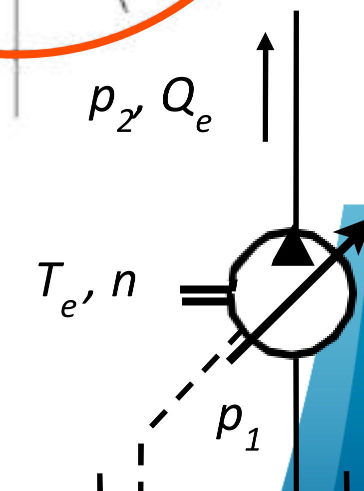
Screw Machines

others

Fixed displacement machines ⟺ Variable displacement machines

Αντλίες αξονικών εμβόλων





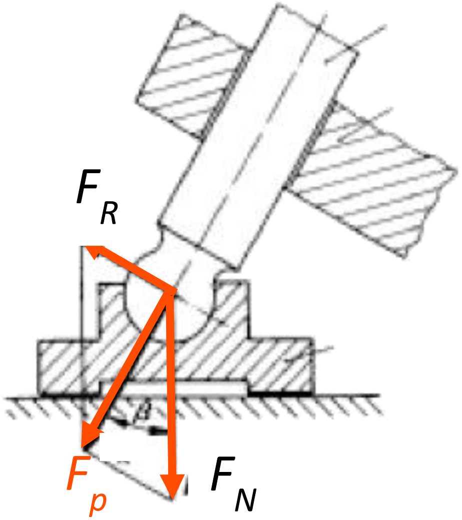
Piston stroke = f (ß,R)

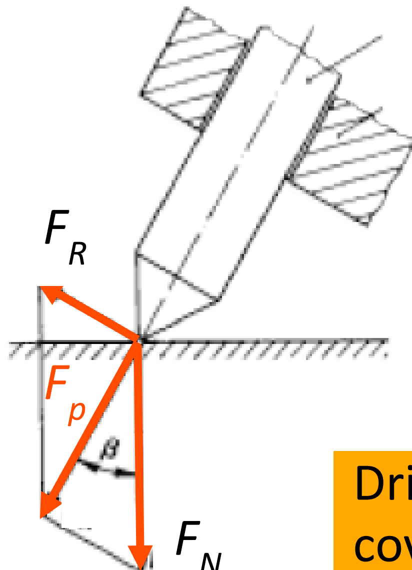
Variable displacement pump
Requires continous change of ß

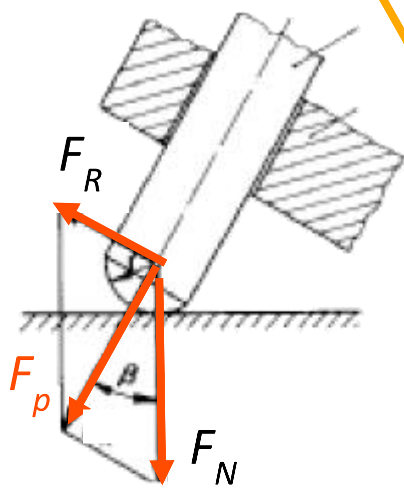
Μηχανές Bent Axis & Swash Plate

Torque generation on cylinder block

Swash plate design

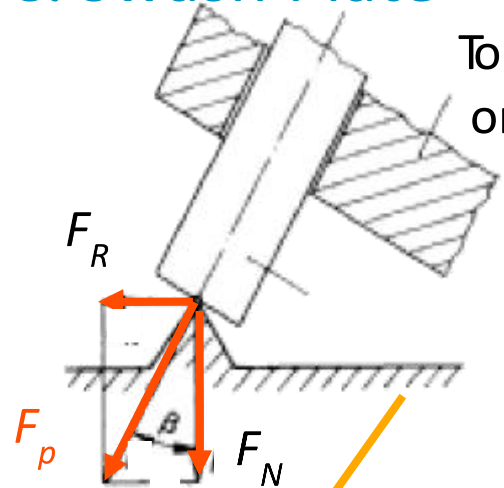




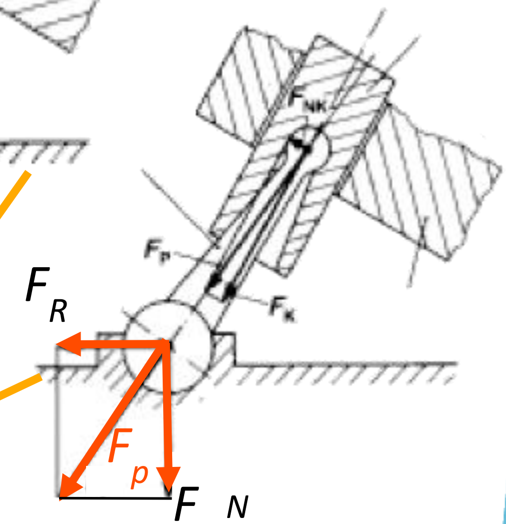


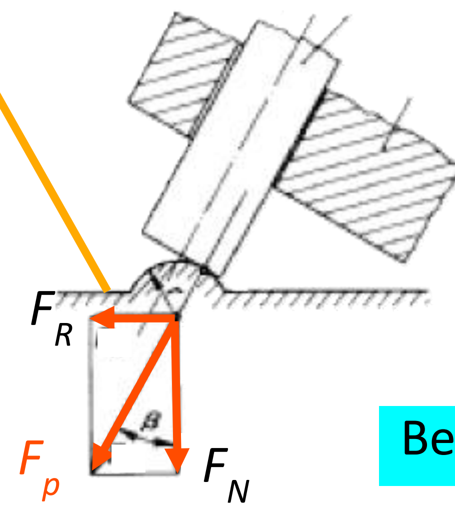
Radial force F_R exerted on piston!

Torque generation on swash plate



Driving flange must cover radial force

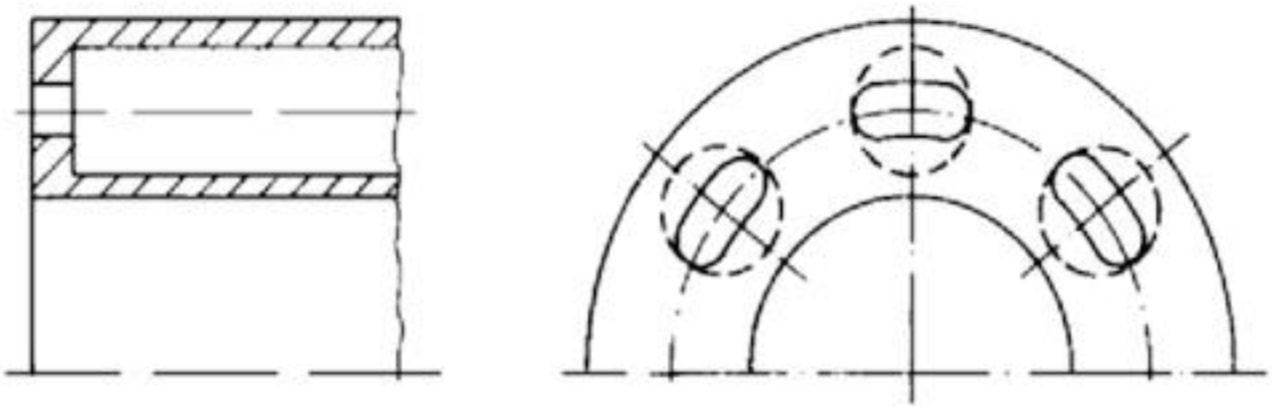




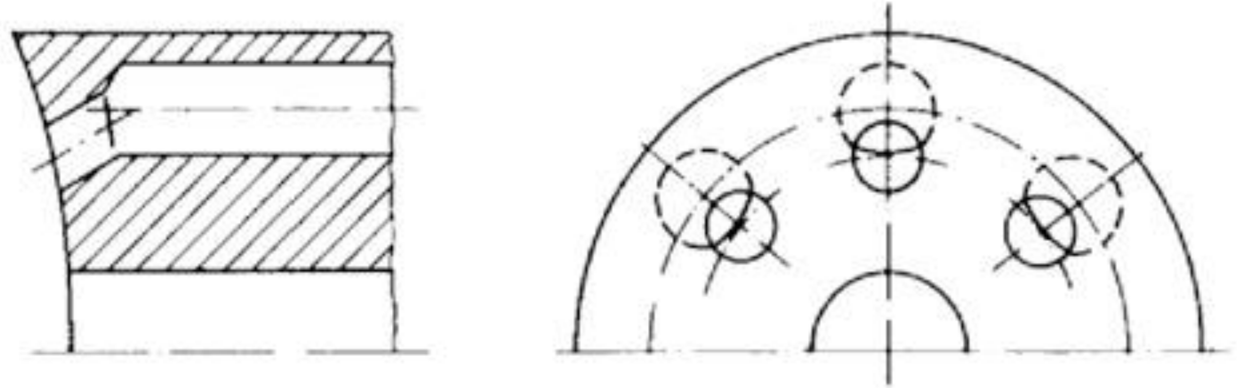
Bent axis machines

Αντλίες αξονικών εμβόλων

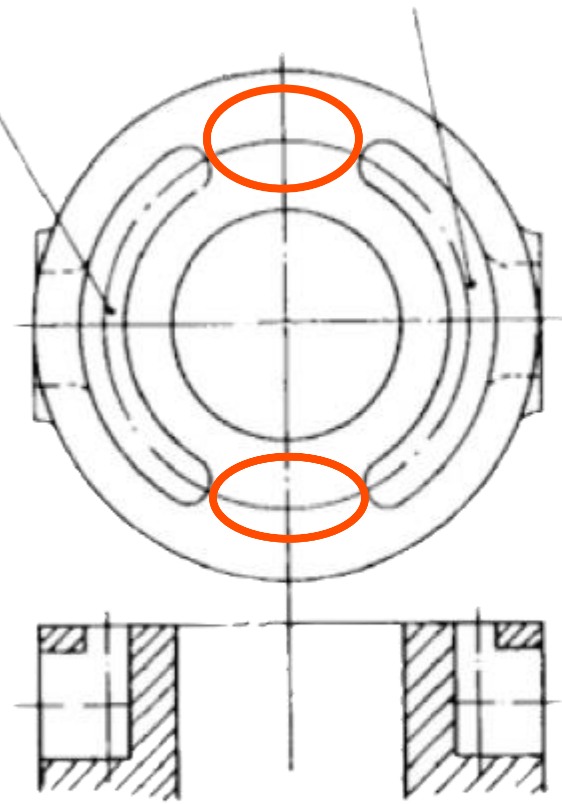
Openings in cylinder bottom In case of plane valve plate

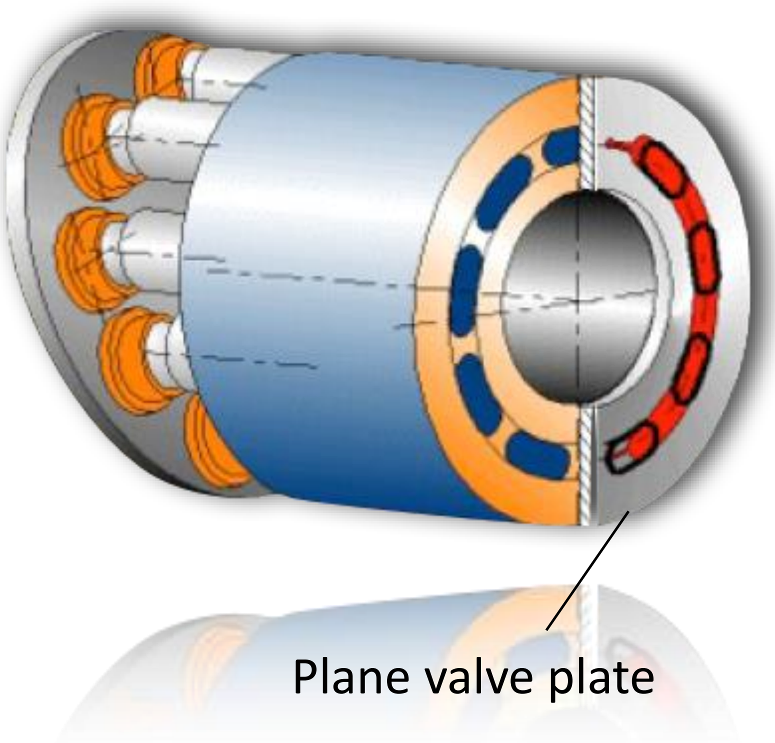


In case of spherical valve plate



Αντλίες αξονικών εμβόλων

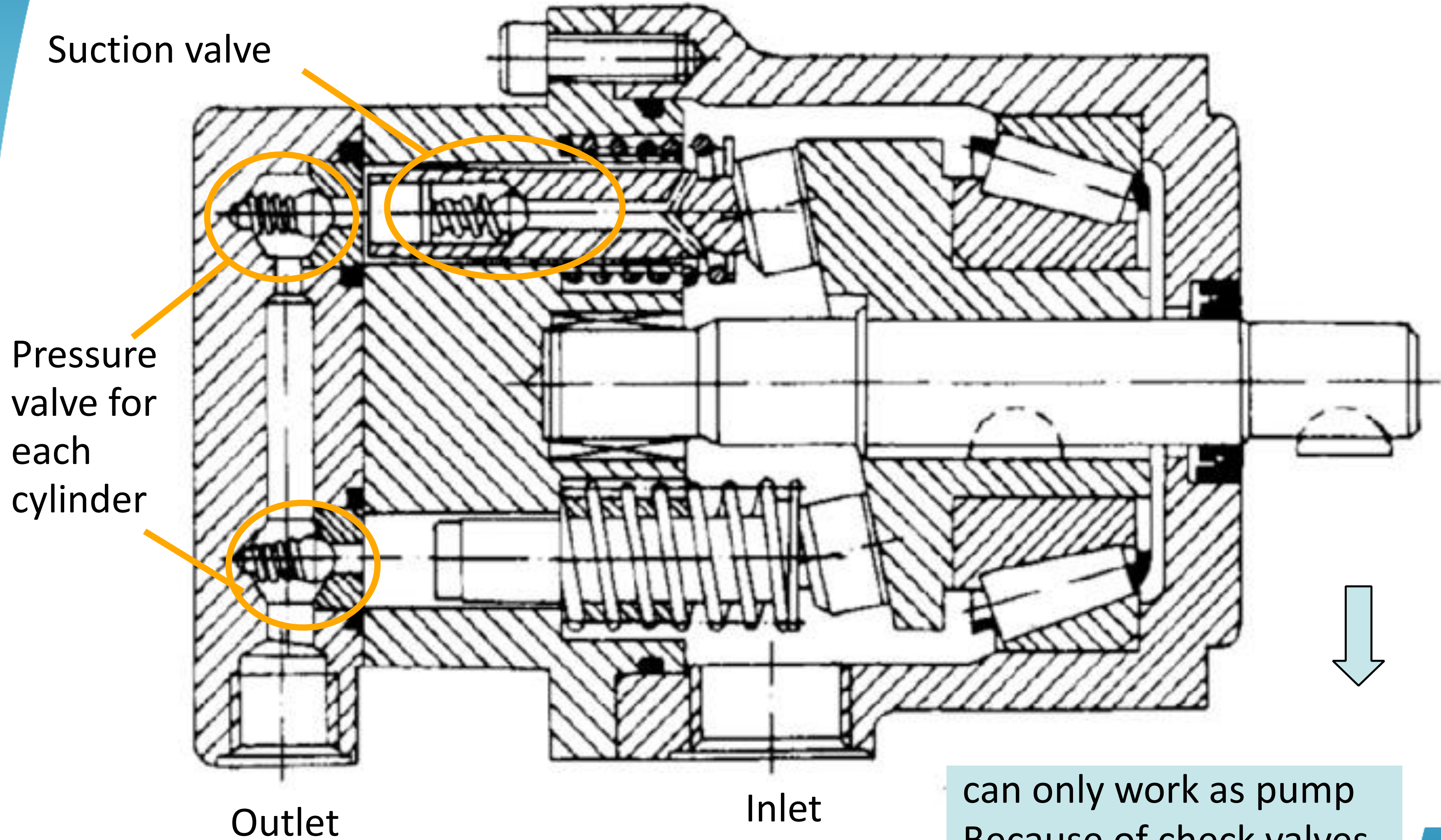




Connection of displacement chambers with suction and pressure port

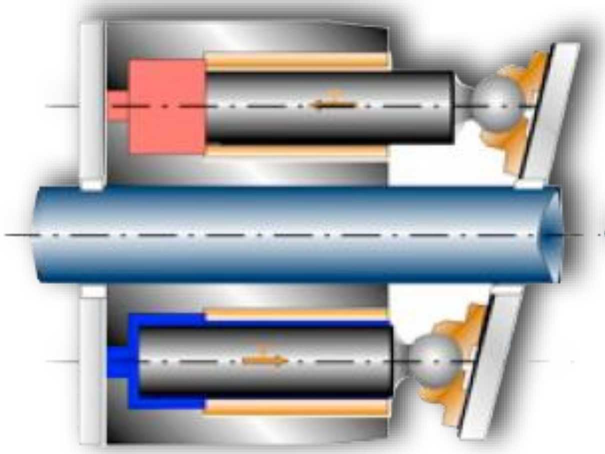
Αντλίες αξονικών εμβόλων

Kinematic reversal: pump with rotating swash plate



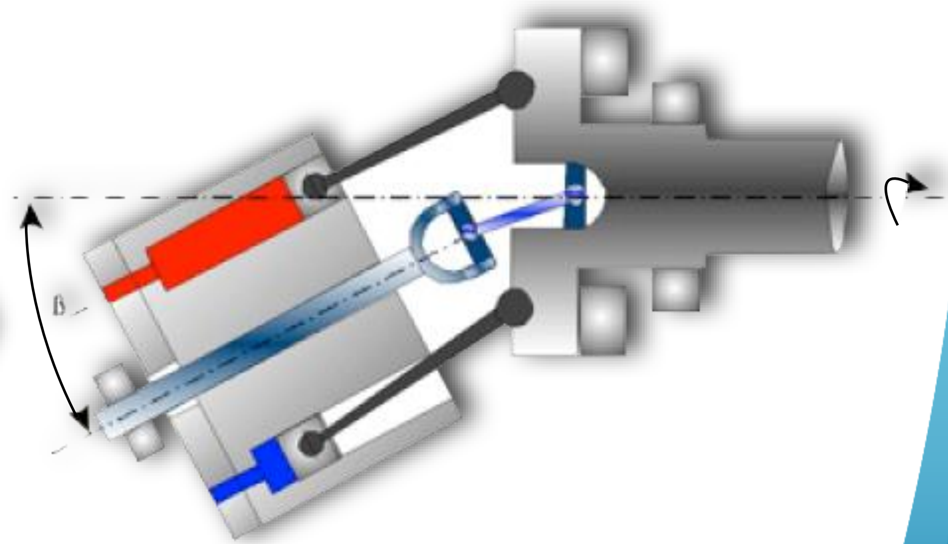
can only work as pump
Because of check valves

Αντλίες αξονικών εμβόλων-Σύγκριση



- Simple and compact design
- Short response time, high bandwidth
- Through going shaft
- Long service life, low loaded bearings
- Limited swash plate angle $ß_{max}$ ca. 21°
- High radial piston forces

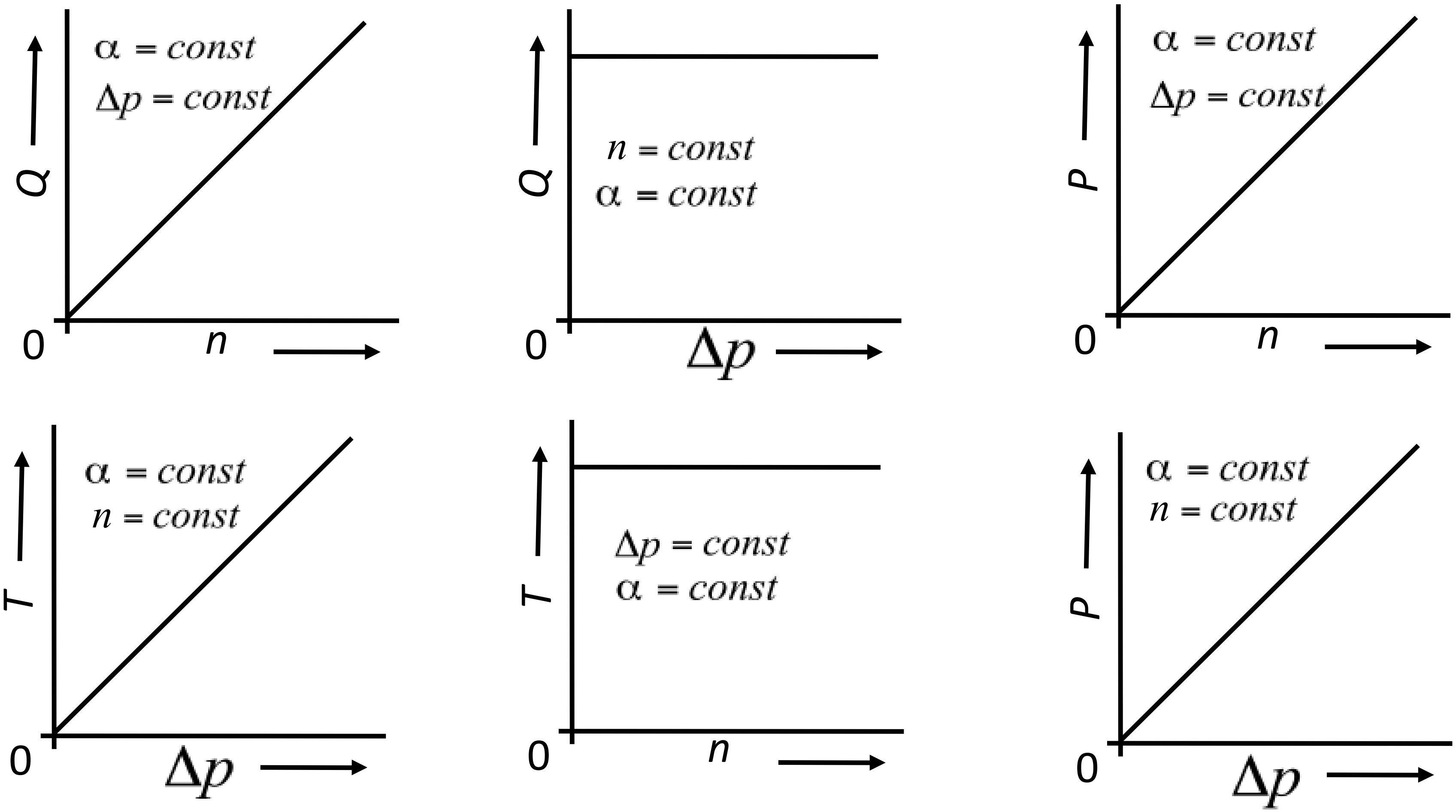
- higher max. speed
- Angle ß up to 45°
- Less losses
- High loaded bearings
- Expensive design
- Synchronisation required



Συμπεριφορά steady state

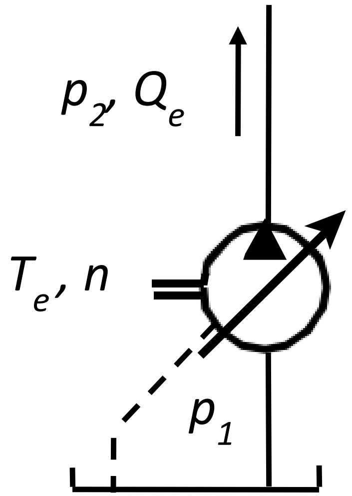
Displacement volume of a variable displacement machine:

$$V = \alpha \; V_{max}$$



Ομοιώτητα

The pump size is determined by the displacement volume V [cm^3/rev]. Usually a proportional scaling law, conserving geometric similarity, is applied, resulting in stresses remaining constant for all sizes of units.



$$T = \frac{\Delta p \cdot V}{2 \cdot \pi} \qquad Q = V \; n \qquad \Delta p = p_2 - p_1$$

First Order Scaling Laws : λ … linear scaling factor

$L = \lambda \, . \, L_0$	$T = \lambda^3 . \, T_0$	$P = \lambda^2 . \, P_0$
$V_i = \lambda^3 . \, V_{i0}$	$m = \lambda^3 . \, m_0$	$n = \lambda^{-1} . \, n_0$

Assuming same maximal operating pressures for all unit sizes and a constant maximal sliding velocity !

Παράδειγμα

The maximal shaft speed of a given pump is 5000 rpm. The displacement volume of this pump is V= 40cm³/rev. The maximal working pressure is given with 40 MPa. Using first order scaling laws, determine:

- the maximal shaft speed of a pump with 90 cm³/rev
- the torque of this larger pump
- the maximal volume flow rate of this larger pump
- the power of this larger pump

For the linear scaling factor follows:

$$\lambda = \sqrt[3]{\frac{V}{V_0}} = \sqrt[3]{\frac{90}{40}} = 1.31$$

Maximal shaft speed of the larger pump:

$$n = \lambda^{-1} \cdot n_0 = 1.31^{-1} \cdot 5000\,\text{rpm} = 3816.8\,\text{rpm}$$

Torque of the larger pump:

$$T = \frac{\Delta p \cdot V}{2 \cdot \pi} = \frac{40 \cdot 10^6\,\text{Pa} \cdot 90 \cdot 10^{-6}\,\text{m}^3}{2 \cdot \pi} = 573.25\,\text{Nm}$$

Maximal volume flow rate:

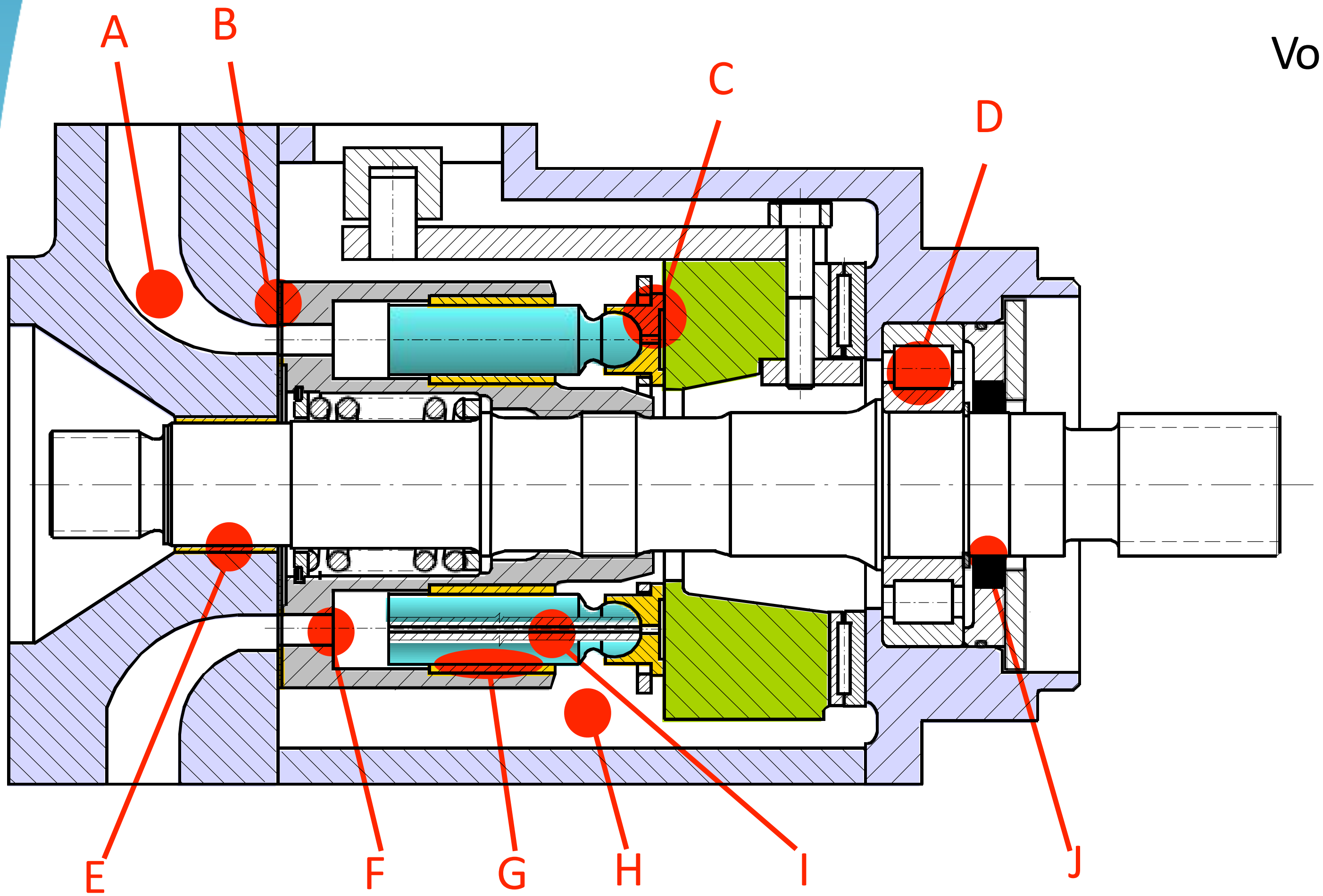
$$Q_{max} = V \cdot n_{max} = 90 \cdot 10^{-6}\,\text{m}^3/\text{rev} \cdot 3816.8\,\text{rpm} = 0.3435\,\text{m}^3/\text{min} = 343.5\,\text{l/min}$$

Power of the larger pump:

$$P = \Delta p \cdot Q = 40 \cdot 10^6\,\text{Pa} \cdot 0.3435\,\text{m}^3 \cdot \frac{1}{60}\,\text{s}^{-1} = 229\text{kW}$$

Απώλειες στις αντλίες θετικής μετατόπισης

Axial piston machine – swash plate design



Volumetric Losses

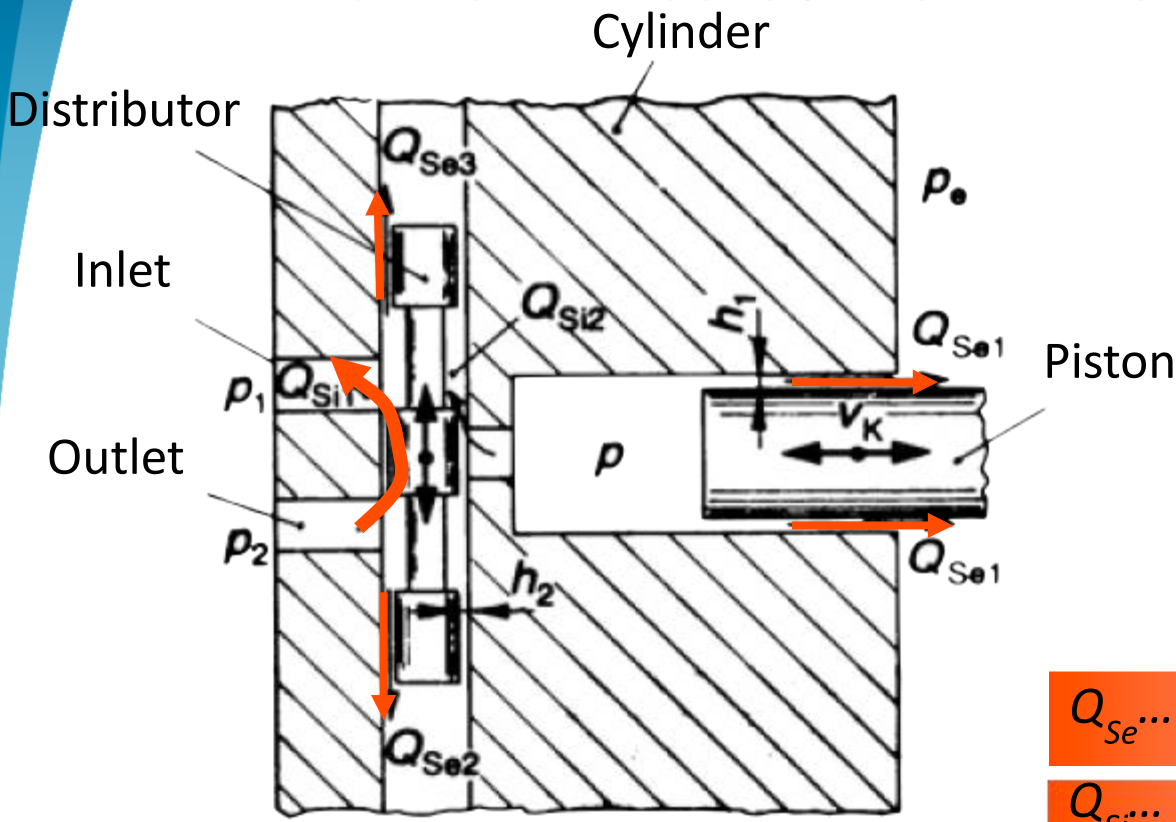
B C F G I

Torque Losses

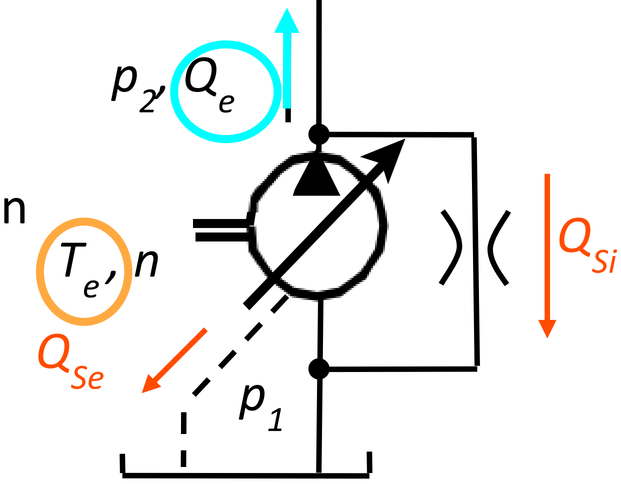
A B C D E

G H J

Πραγματική μηχανή θετικής μετατόπισης



$$\Delta p = p_2 - p_1$$



Q_{Se}... external volumetric losses

Q_{Si}... internal volumetric losses

Effective Flow rate: $Q_e = \alpha V_{max}\, n - Q_s$

Q_S … *volumetric losses*

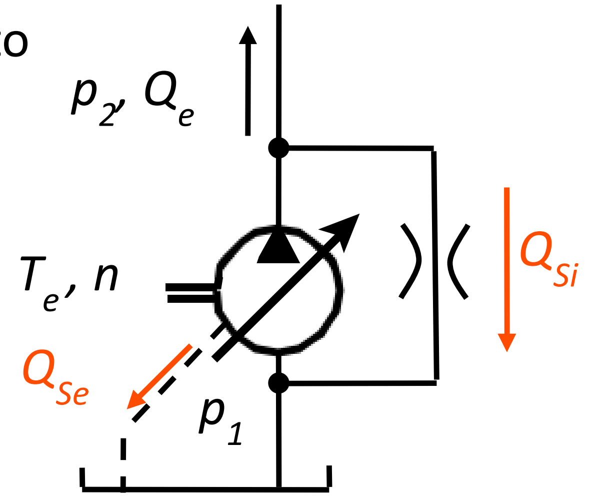
Effective torque: $T_e = \dfrac{\Delta p \cdot \alpha \cdot V_{\max}}{2 \cdot \pi} + T_S$

T_S…*torque losses*

Ογκομετρικές απώλειες

$$Q_S = \sum_{i=1}^{n} Q_{Sei} + \sum_{j=1}^{m} Q_{Sij} + Q_{SK} + Q_{Sf}$$

external / internal volumetric losses — losses due to compressibility — losses due to incomplete filling

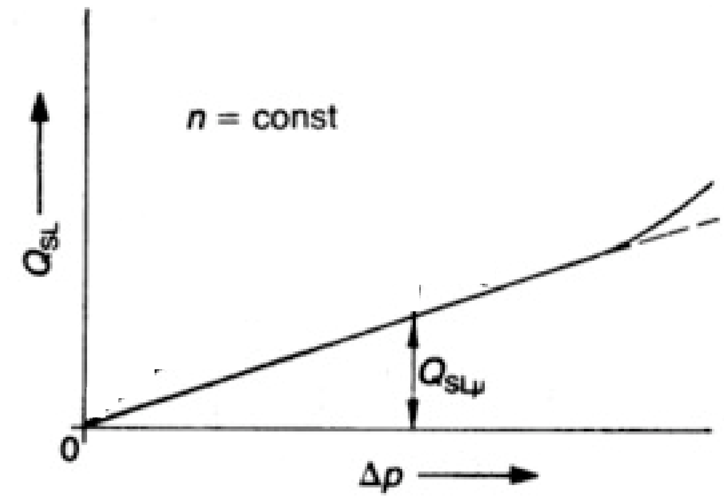


Q_{SL} external and internal volumetric losses = flow through laminar resistances:

$$Q_{SL} = C_\mu \cdot \frac{\Delta p}{\mu}$$

$$Q = \frac{b \cdot h^3 \cdot \Delta p}{12 \cdot \mu}$$

Assuming const. gap height



Dynamic viscosity

$$\mu = f(\theta, p)$$

Στιγμιαία παροχή αντλίας

Instantaneous volumetric flow Q_a

$$Q_a = \frac{dV}{dt} = f(\varphi)$$

Volumetric flow displaced by a displacement chamber

$$Q_{ai} = f(\varphi_i)$$

The instantaneous volumetric flow is given by the sum of instantaneous flows Q_{ai} of each displacement element:

$$Q_a = \sum_{i=1}^{k} Q_{ai}$$

k … number of displacement chambers, decreasing their volume, i.e. being in the delivery stroke

z is an even number $k = \frac{z}{2}$ z … number of displacement elements

z is an odd number $k = \frac{z}{2} + 0.5$ or $k = \frac{z}{2} - 0.5$

→ Flow pulsation of pumps → Pressure pulsation

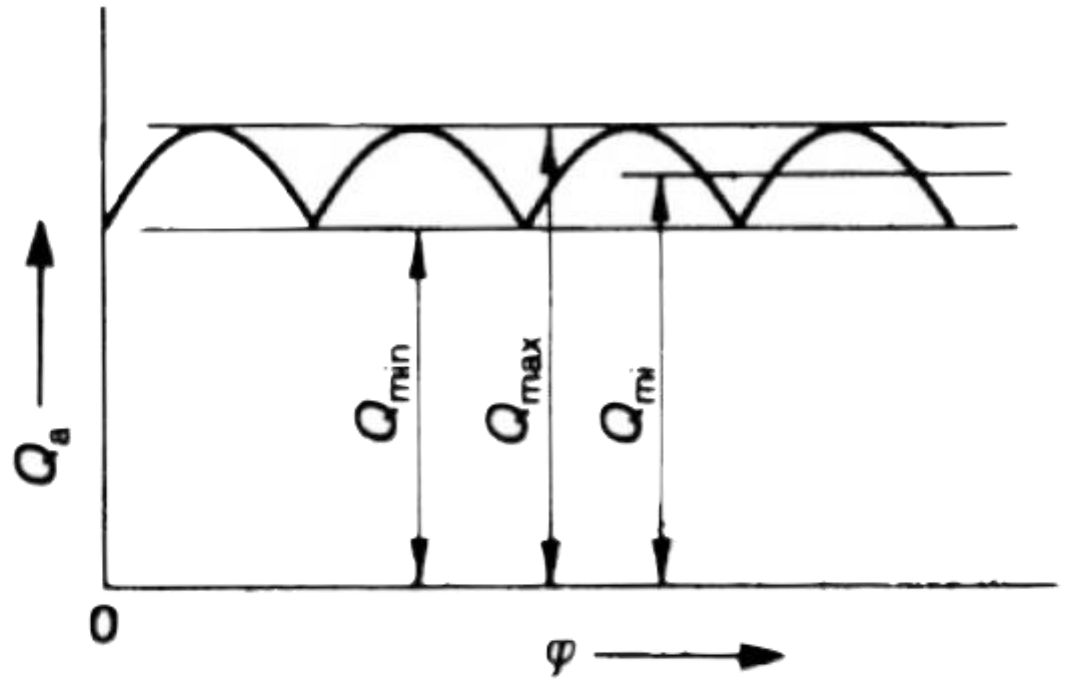
Παλμός παροχής

Non-uniformity grade of volumetric flow is defined:

$$\delta_Q = \frac{Q_{\max} - Q_{\min}}{Q_{mi}}$$

$$Q_{mi} = \frac{Q_{\max} + Q_{\min}}{2}$$

$$\delta_Q = 2\,\frac{Q_{\max} - Q_{\min}}{Q_{\max} + Q_{\min}}$$



Απώλειες ροπής

$$T_S = T_{S\mu} + T_{S\rho} + T_{Sp} + T_{Sc} \Rightarrow$$ constant value

Torque loss due to viscous friction in gaps (laminar flow)

$$T_{S\mu} = k_{T\mu} \cdot \frac{\mu}{h} \cdot n = C_{T\mu} \cdot \mu \cdot n$$

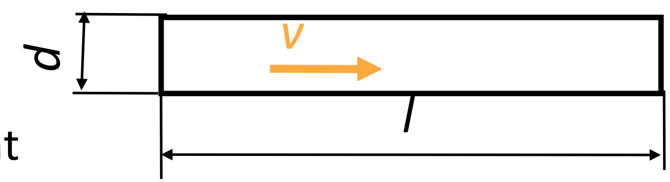
h…gap height

Torque loss to overcome pressure drop caused in turbulent resistances

$$T_{S\rho} = C_{T\rho} \; \rho \; n^2$$

$$\Delta p_s = \lambda \cdot \frac{l}{d} \cdot \rho \cdot \frac{v^2}{2} + \xi \cdot \rho \cdot \frac{v^2}{2}$$

Torque loss linear dependent on pressure



$$T_{Sp} = C_{Tp} \; \Delta p$$

ξ … drag coefficient

λ … flow resistance coefficient $\quad \lambda_{turbulent} = \frac{0.3164}{\sqrt[4]{Re}}$

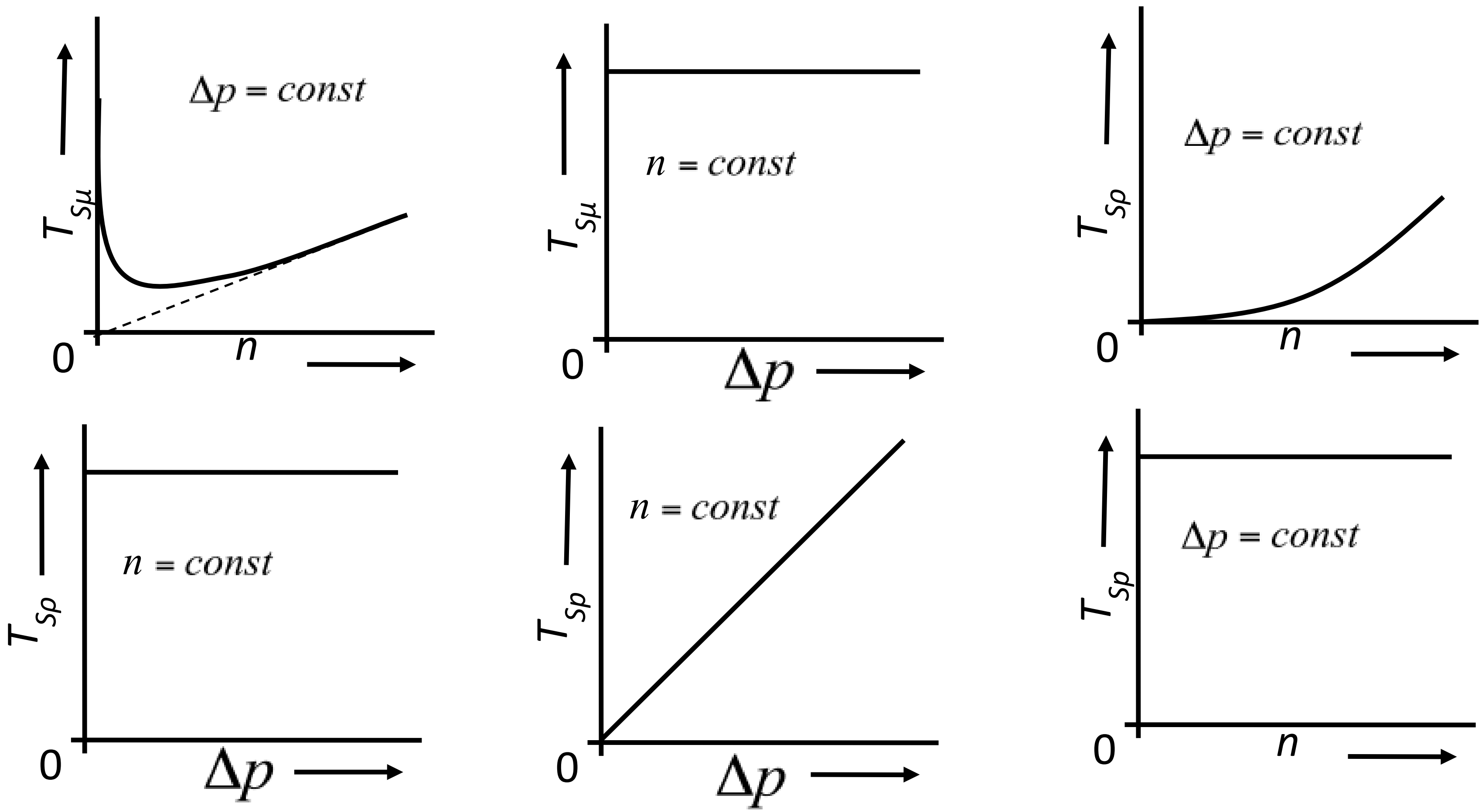
$$\Rightarrow \quad T_e = \frac{\Delta p \cdot \alpha \cdot V_{max}}{2 \cdot \pi} + T_S$$

effective torque required at pump shaft

Συμπεριφορά steady state

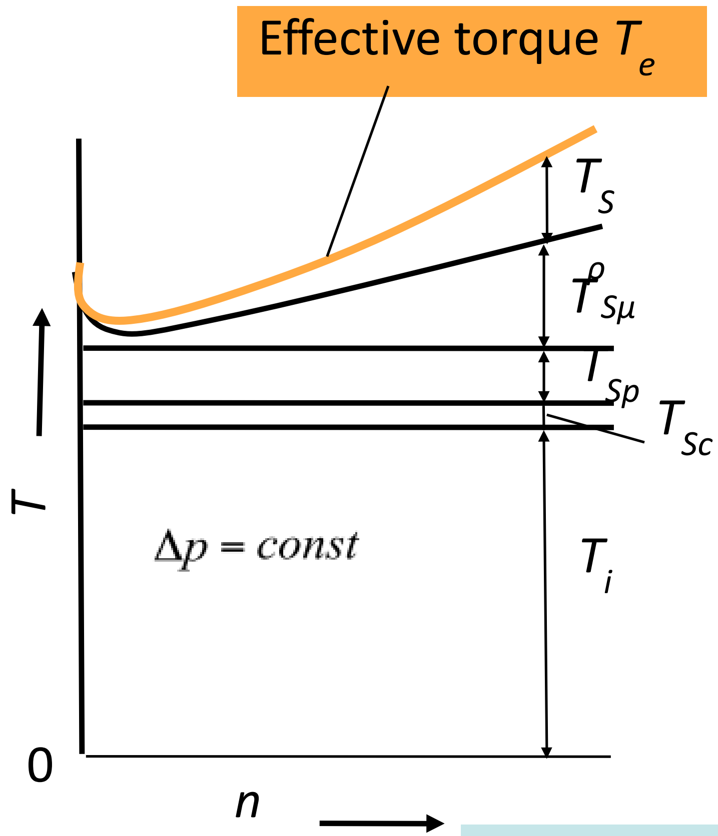
Torque losses of
a real displacement machine

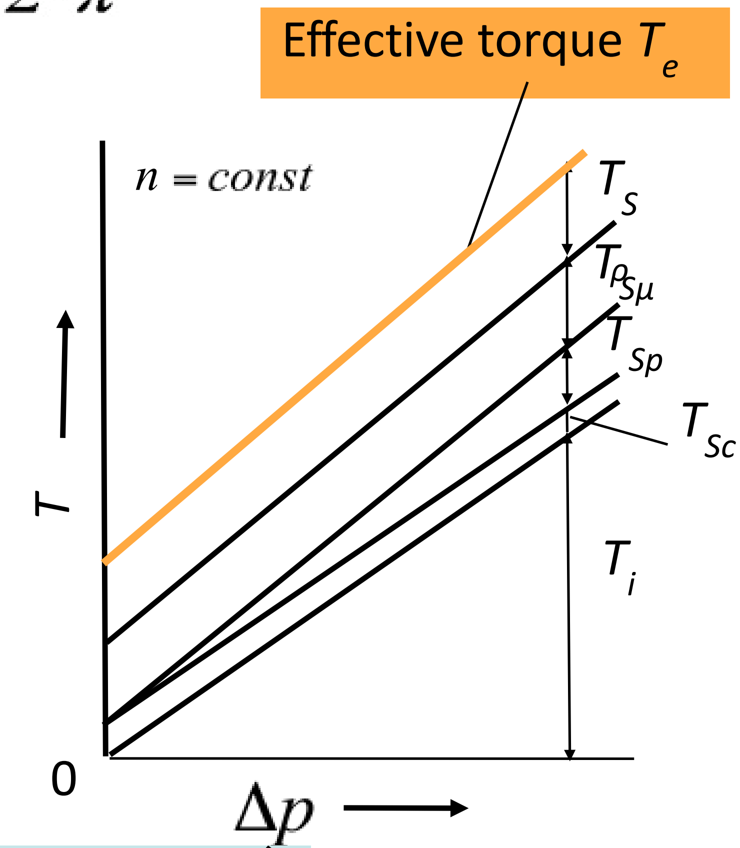
$$T_S = f(n, \Delta p, V, \theta)$$



Συμπεριφορά steady state

$$T_e = T_i + T_S = \frac{\Delta p \cdot \alpha \cdot V_{max}}{2 \cdot \pi} + T_S$$





$$T_S = f(\Delta p, n, V, \theta)$$

Αντλία αξονικών εμβόλων-Κινηματικά

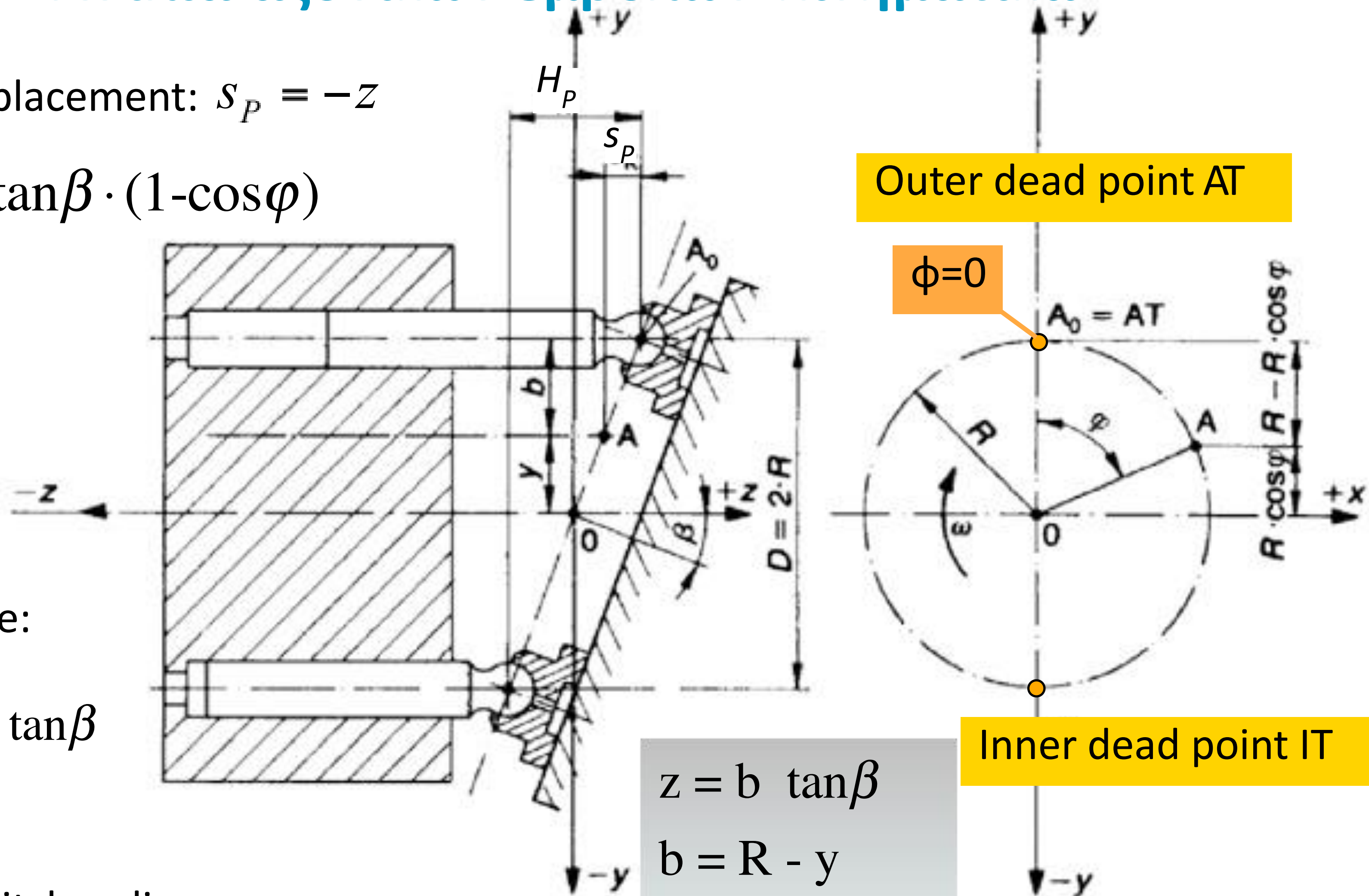
Piston displacement: $s_P = -z$

$$s_p = -R \cdot \tan\beta \cdot (1 - \cos\varphi)$$

Piston stroke:

$$H_P = 2 \cdot R \cdot \tan\beta$$

R … pitch radius



$$z = b \ \tan\beta$$

$$b = R - y$$

$$y = R \cdot \cos\varphi$$

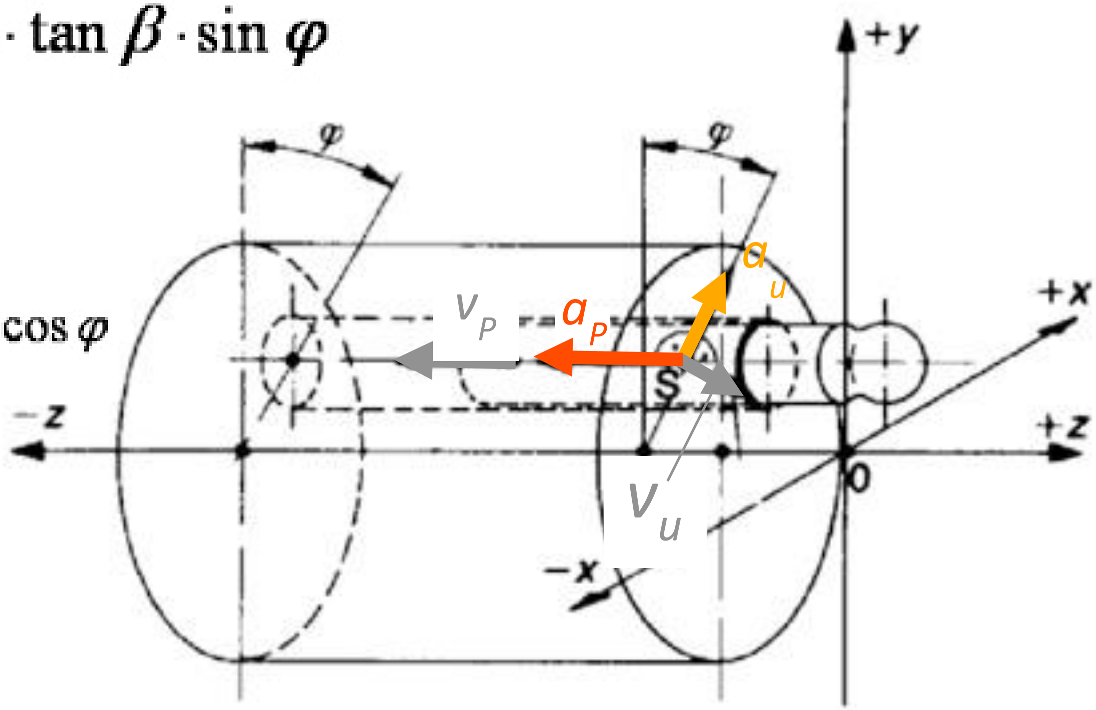
Αντλία αξονικών εμβόλων-Κινηματικά

Piston velocity in z-direction:

$$v_P = \frac{ds_P}{dt} = \frac{ds_P}{d\varphi} \cdot \frac{d\varphi}{dt} = -\omega \cdot R \cdot \tan\beta \cdot \sin\varphi$$

Piston acceleration in z-direction:

$$a_P = \frac{dv_P}{dt} = \frac{dv_P}{d\varphi} \cdot \frac{d\varphi}{dt} = -\omega^2 \cdot R \cdot \tan\beta \cdot \cos\varphi$$



Circumferential speed

$$v_u = R\ \omega$$

Centrifugal acceleration:

$$a_u = R\ \omega^2$$

Coriolis acceleration a_c is just zero, as the vector of angular velocity ω and the piston velocity v_P run parallel

Στιγμιαία παροχή

Geometric displacement volume:

$V_g = z \; A_p \; H_P$ $\quad$ *z* … number of pistons

For an ideal pump without losses

In case of pistons arranged parallel to shaft axis:

$$V_g = z \cdot \frac{\pi \cdot d_P^2}{2} \cdot R \cdot \tan \beta$$

Geometric flow rate: $Q_g = n \cdot z \cdot \frac{\pi \cdot d_P^2}{2} \cdot R \cdot \tan \beta$ → Mean value over time

Instantaneous volumetric flow: $Q_a = \sum_{i=1}^{k} Q_{ai}$ $\quad$ *k* …number of pistons, which are in the delivery stroke

with Q_{ai} instantaneous volumetric flow of individual piston $\quad$ $Q_{ai} = f(\varphi_i)$

$$\mathrm{v_P} = \omega \cdot \mathrm{R} \cdot \tan\beta \cdot \sin\varphi$$

$$\mathrm{Q_{ai}} = \mathrm{v_p} \cdot \mathrm{A_p} = \omega \cdot \mathrm{A_p} \cdot \mathrm{R} \cdot \tan\beta \cdot \sin\varphi_\mathrm{i}$$

Στιγμιαία παροχή

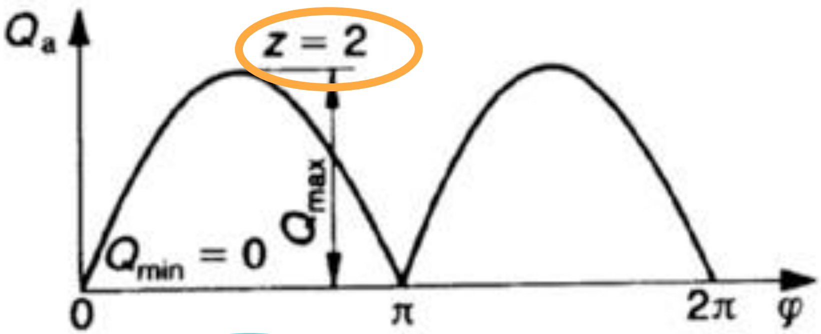
In case of even number of pistons: $k = 0.5 \cdot z$

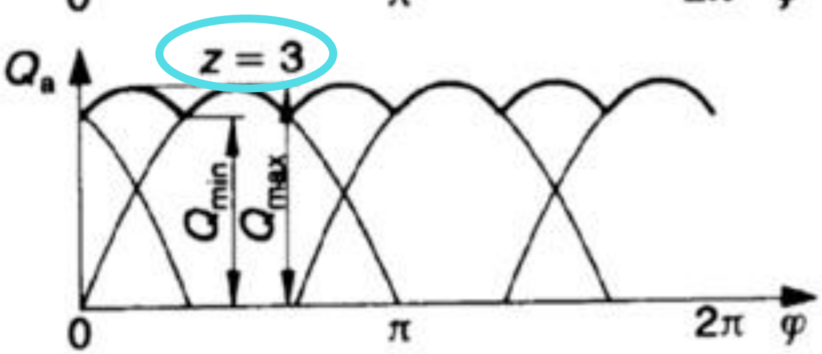
$$Q_a = \sum_{i=1}^{k} Q_{ai}$$

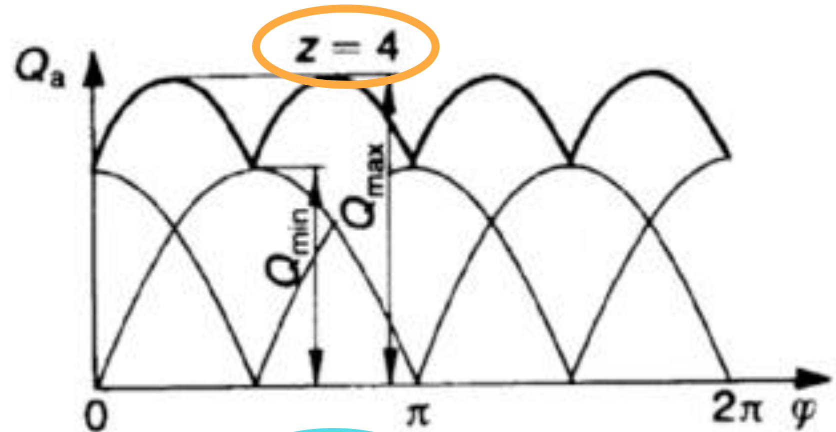
In case of odd number of pistons:

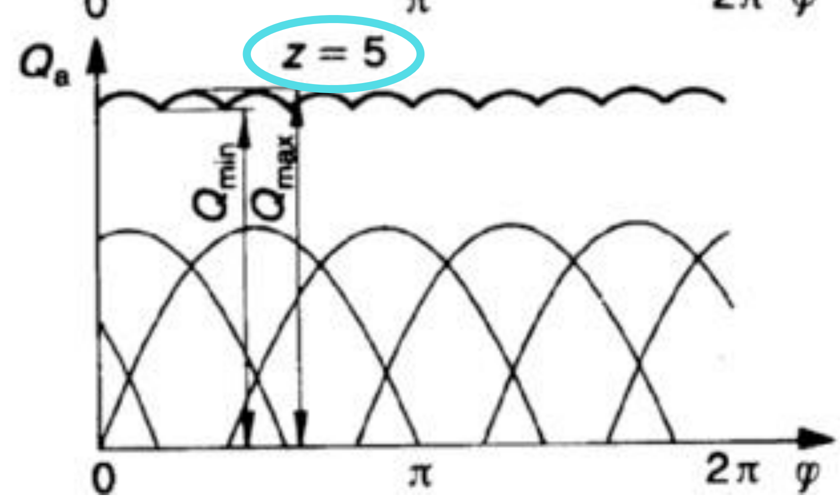
$$k_1 = \frac{z}{2} + 0.5 \quad \text{for} \quad 0 < \varphi \leq \frac{\pi}{z}$$

$$\text{and} \quad k_2 = \frac{z}{2} - 0.5 \quad \text{for} \quad \frac{\pi}{z} < \varphi \leq 2 \cdot \frac{\pi}{z}$$

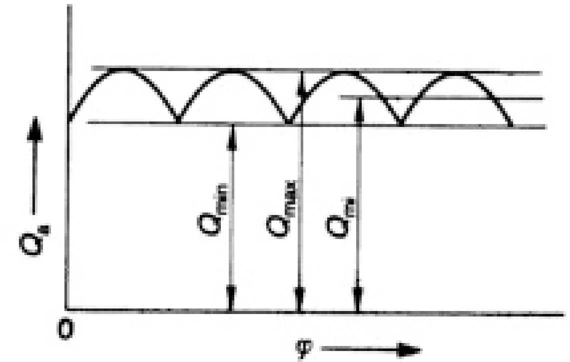








Παλμός παροχής και ροπής



kinematic flow and torque pulsation due to a finite number of piston

Flow Pulsation:

Non-uniformity grade:

$$\delta_Q = \frac{Q_{\max} - Q_{\min}}{Q_{mi}} \quad \text{with} \quad Q_{mi} = \frac{Q_{\max} + Q_{\min}}{2}$$

Even number of pistons:

$$\delta_Q = \frac{\pi}{z} \tan \frac{\pi}{2z}$$

Odd number of pistons:

$$\delta_Q = \frac{\pi}{2z} \tan \frac{\pi}{4z}$$

Torque Pulsation

$$\delta_T = \frac{T_{\max} - T_{\min}}{T_{mi}} \quad \text{with} \quad T_{mi} = \frac{T_{\max} + T_{\min}}{2}$$

Παλμός παροχής και ροπής

kinematic flow and torque pulsation due to a finite number of piston

z… number of pistons

Non-uniformity

$$\delta_Q = \delta_T = \frac{Q_{max} - Q_{min}}{Q_{mean}} = \frac{T_{max} - T_{min}}{T_{mean}}$$

Even number of pistons:

$$\delta_Q = \frac{\pi}{z} \cdot \tan \frac{\pi}{2 \cdot z}$$

Odd number of pistons:

$$\delta_Q = \frac{\pi}{2 \cdot z} \cdot \tan \frac{\pi}{4 \cdot z}$$

NUMBER OF PISTONS	3	4	5	6	7	8	9	10	11	12	13	14
NON-UNIFORMITY of FLOW / TORQUE	0,1403	0,3253	0,0498	0,1403	0,0253	0,0781	0,0153	0,0498	0,0102	0,0345	0,0073	0,0253

NUMBER OF PISTONS	15	16	17	18	19	20	21	22	23	24	25	26
NON-UNIFORMITY of FLOW / TORQUE	0,0055	0,0193	0,0043	0,0153	0,0034	0,0124	0,0028	0,0102	0,0023	0,0086	0,0020	0,0073

Flow and torque pulsation frequency f:

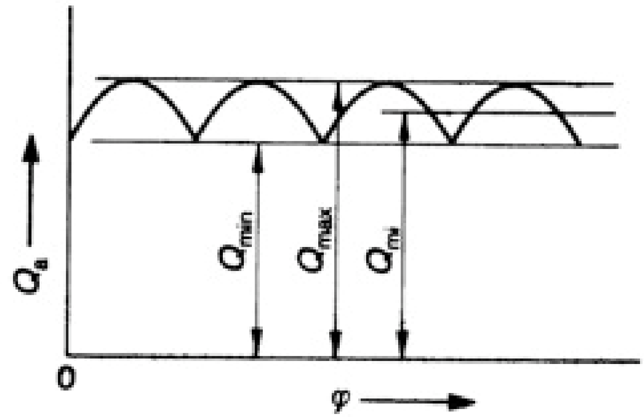
Even number of pistons: $f = z \cdot n$

Odd number of pistons: $f = 2 \cdot z \cdot n$

Παλμός παροχής

Non-uniformity grade:

$$\delta_Q = \frac{Q_{\max} - Q_{\min}}{Q_{mi}} \quad \text{with} \quad Q_{mi} = \frac{Q_{\max} + Q_{\min}}{2}$$



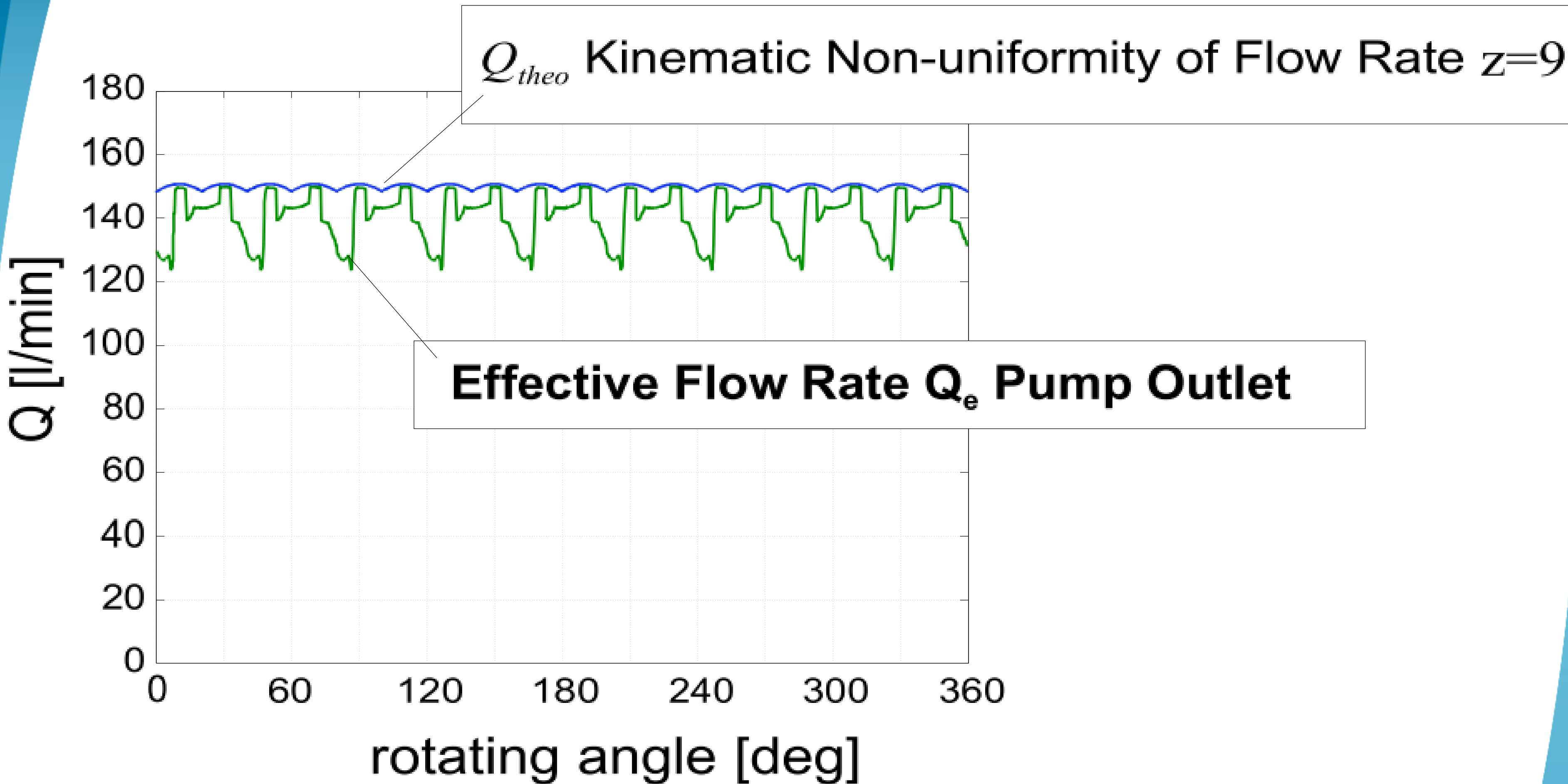
Kinematic non-uniformity grade for piston machines:

Number of pistons z	3	4	5	6	7	8	9	10	11
Non-uniformity grade δ	0.140	0.325	0.049	0.140	0.025	0.078	0.015	0.049	0.010

Volumetric losses $Q_s = f(\phi)$ and $Q_S = f(\Delta p, n, V_i, \theta)$

Flow pulsation of a real displacement machine is much larger than the flow pulsation given by the kinematics

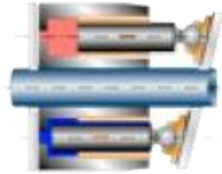
Παλμός παροχής



Flow pulsation leads to pressure pulsation at pump outlet

Μηχανές θετικής μετατόπισης

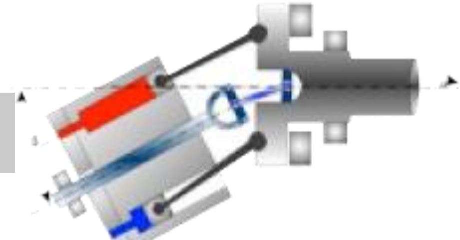
Axial Piston Machines



Swash Plate Machines

Piston Machines

Bent Axis machines

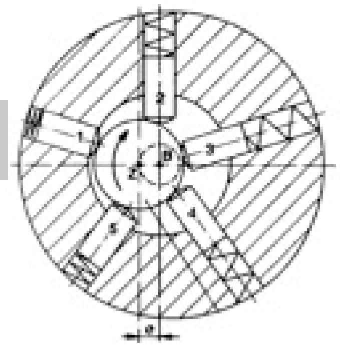


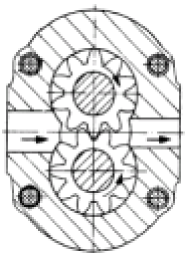
In-line Piston Machines

with external piston support

Radial Piston Machines

with internal piston support

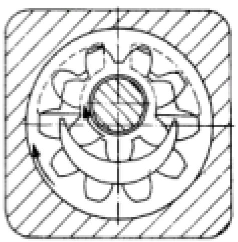




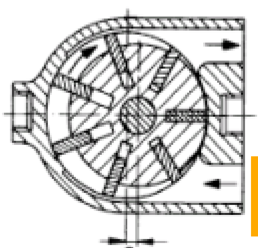
External Gear

Gear Machines

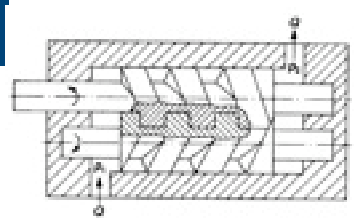
Internal Gear



Annual Gear



Vane Machines

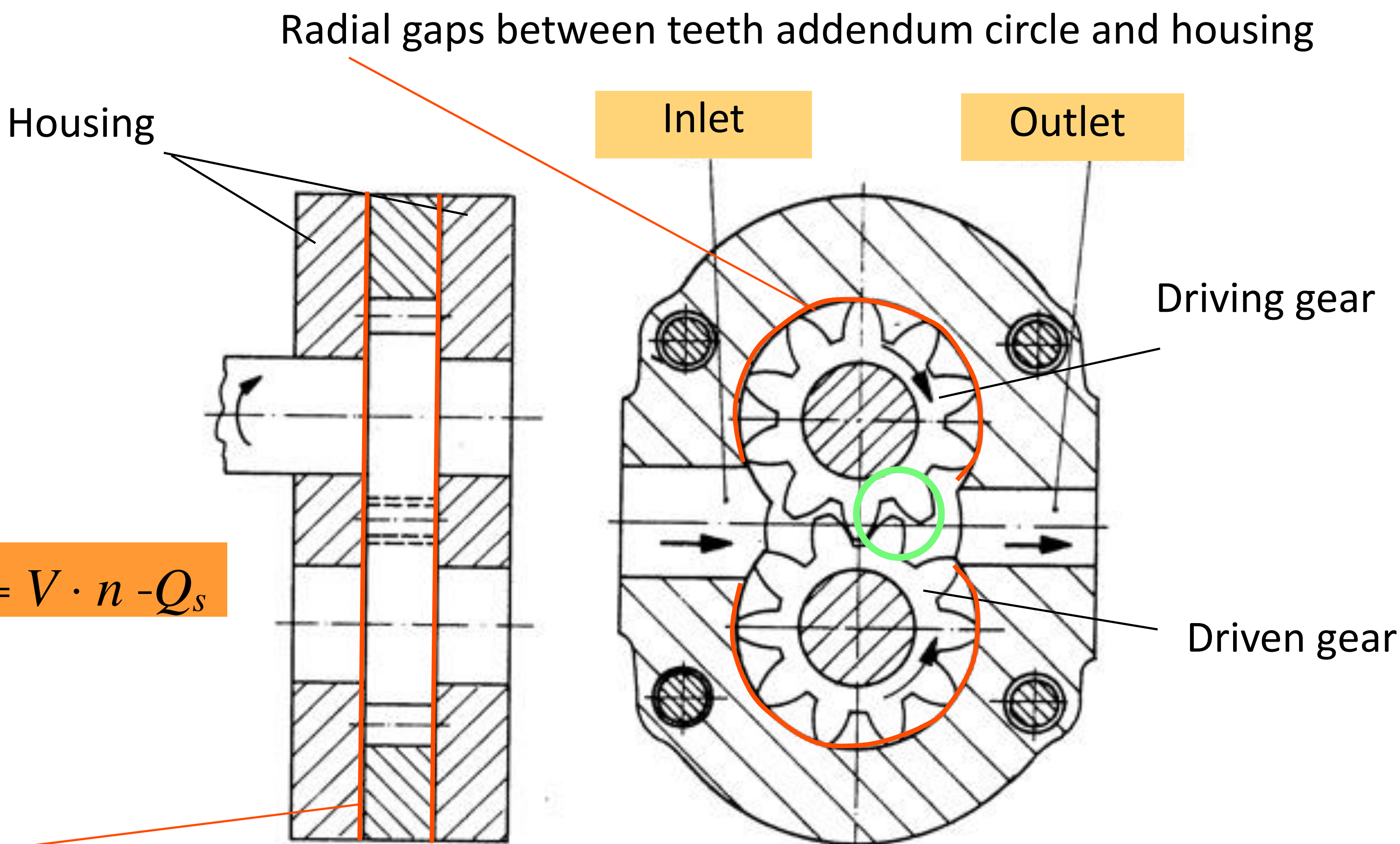


Screw Machines

others

Fixed displacement machines ⟷ Variable displacement machines

Γραναζωτή αντλία εξωτερικής οδόντωσης



$$Q_e = V \cdot n - Q_s$$

Axial gaps between housing and the gear pair must be very small to seal the displacement chamber

Expression for the theoretical flow rate of an external gear pump

D_o =the outside diameter of gear teeth
D_i= the inside diameter of gear teeth
L =the width of gear teeth
N=the speed of pump in RPM
V_D=the displacement of pump in m/rev
M= module of gear
z=number of gear teeth
α= pressure angle
Volume displacement is

$$V_D = \frac{\pi}{4}(D_o^2 - D_i^2)L$$

$$D_i = D_o - 2(\text{Addendum} + \text{Dendendum})$$

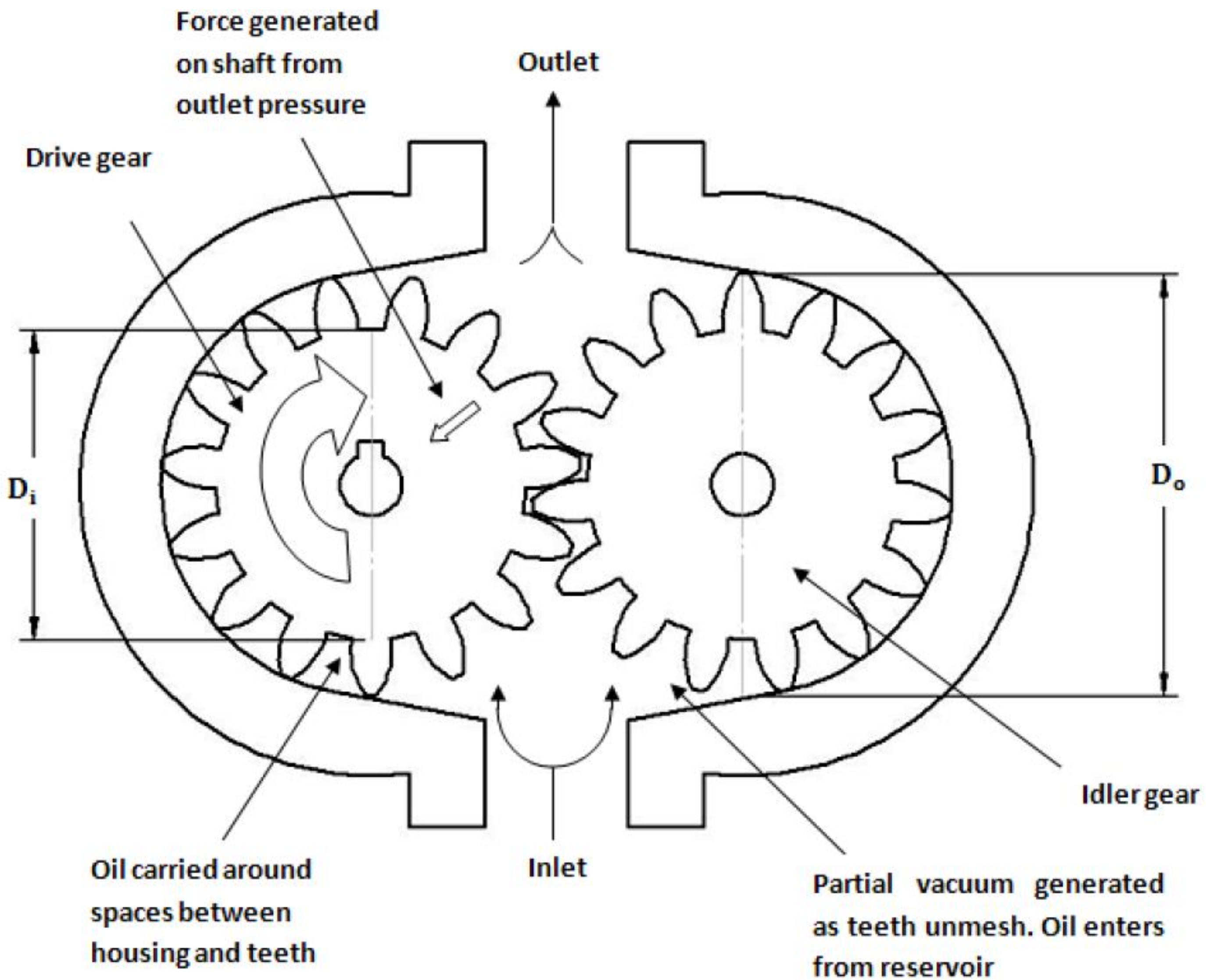
Theoretical discharge is

$$Q_T\ (\text{m}^3/\text{min}) = V_D(\text{m}^3/\text{rev}) \times N\ (\text{rev/min})$$

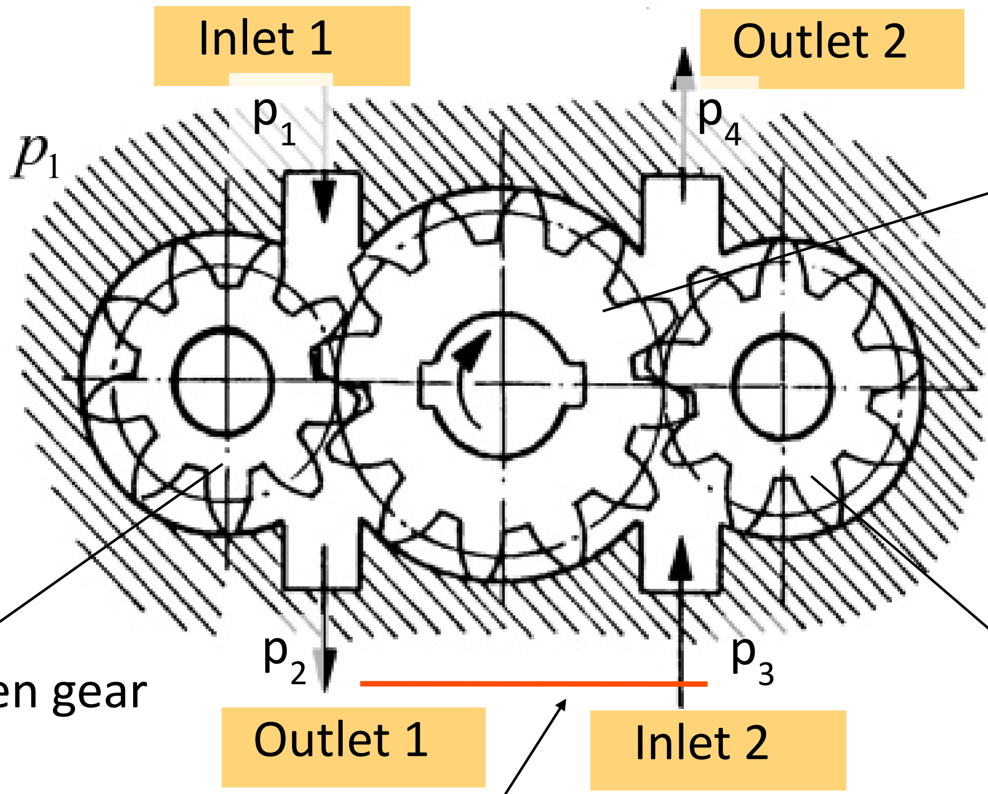
If the gear is specified by its module and number of teeth, then the theoretical discharge can be found by

$$Q_T = 2\pi L m^2 N\left[z + \left(1 + \frac{\pi^2 \cos^2 20}{12}\right)\right] \text{m}^3/\text{min}$$

Expression for the theoretical flow rate of an external gear pump



Γραναζωτή αντλία εξωτερικής οδόντωσης δυο βαθμίδων



$$\Delta p_1 = p_2 - p_1$$

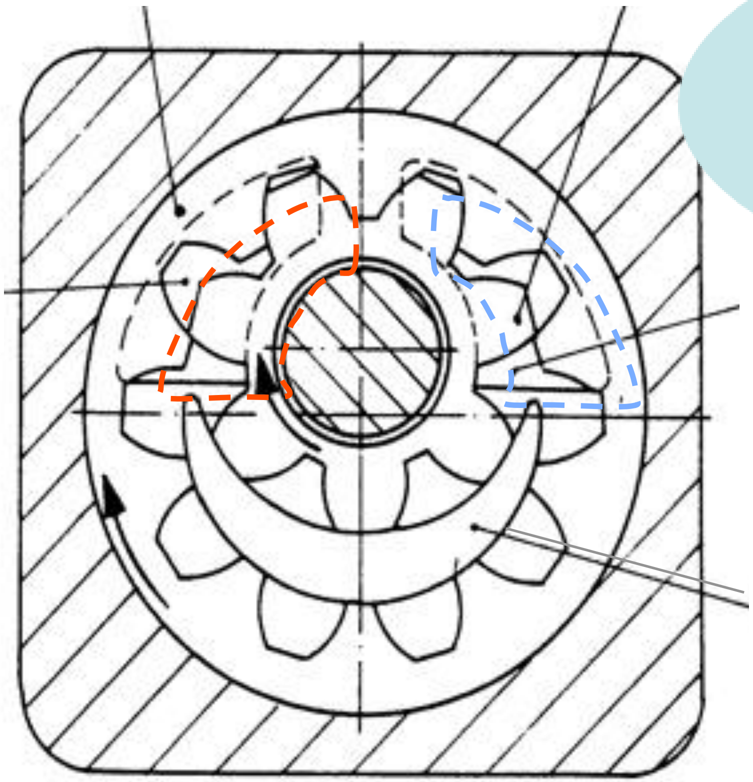
$$\Delta p_2 = p_4 - p_3$$

Outlet 1 and inlet 2 can be connected ⇨ $p_2 = p_3$

or the pump can have two separate outlets ⇨ $p_1 = p_3$

$\Delta p_1 \approx \Delta p_2$ ⇨ the driving gear is pressure balanced!

Γραναζωτή αντλία εσωτερικής οδόντωσης

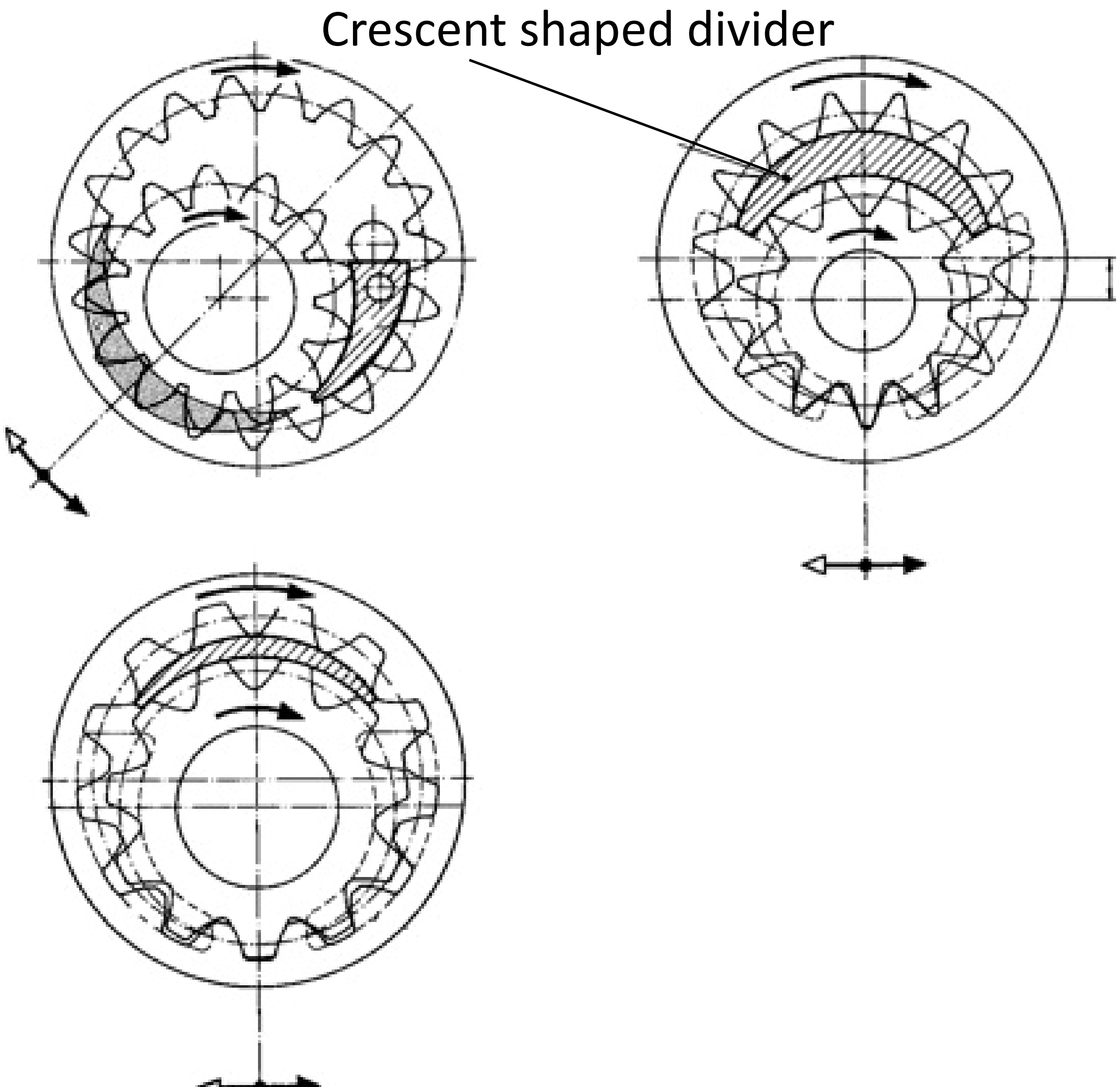


Advantages:
Better suction ability
Higher efficiency
More compact design
Less noise emission

Using teeth of standard involute design requires a combination where the pinion has two or more fewer teeth than the ring gear! Pinion and ring gear are then separated by a crescent shaped divider.

→ Longer duration of teeth meshing leads to better sealing function

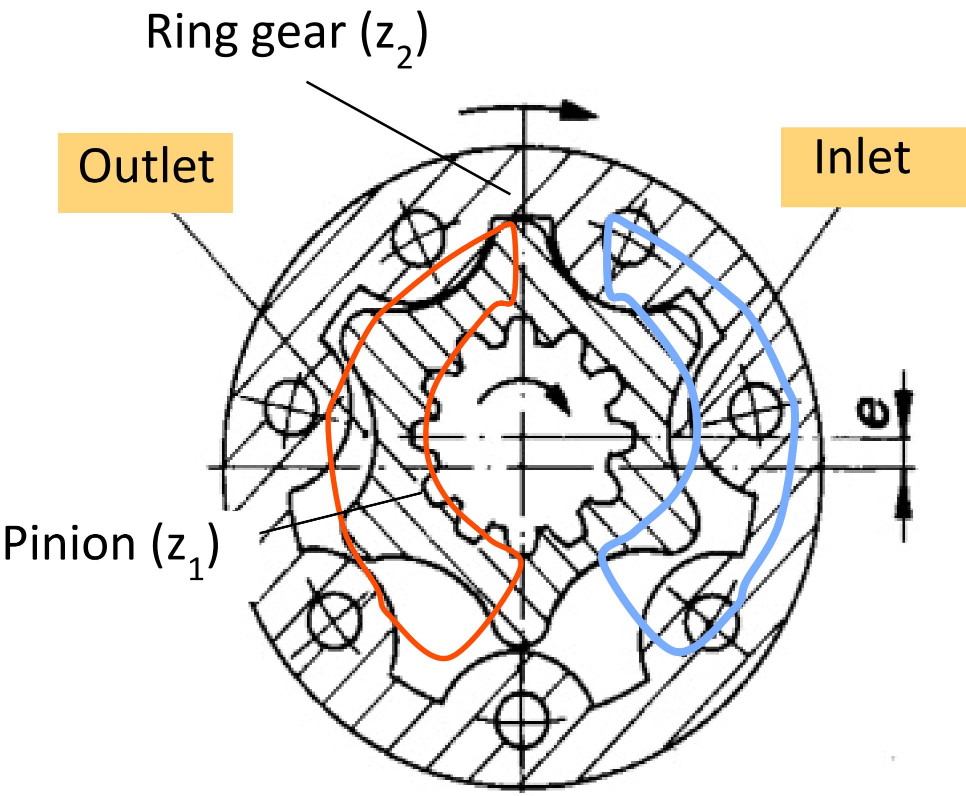
Γραναζωτή αντλία εσωτερικής οδόντωσης



Many different tooth profiles have been applied in the recent past.

Αντλίες τύπου gerotor

Applying specially generated tooth curves it can be achieved, that the inner rotor (the pinion) has only one tooth less than the ring gear, thus eliminating the crescent-shaped divider.



Each tooth of the pinion maintains continuous sliding contact with a tooth of the ring gear, providing fluid tight engagement.

Relative sliding velocity between pinion and ring gear is very small

quiet operation and long service life

Gerotor pump

$$z_2 = z_1 + 1$$

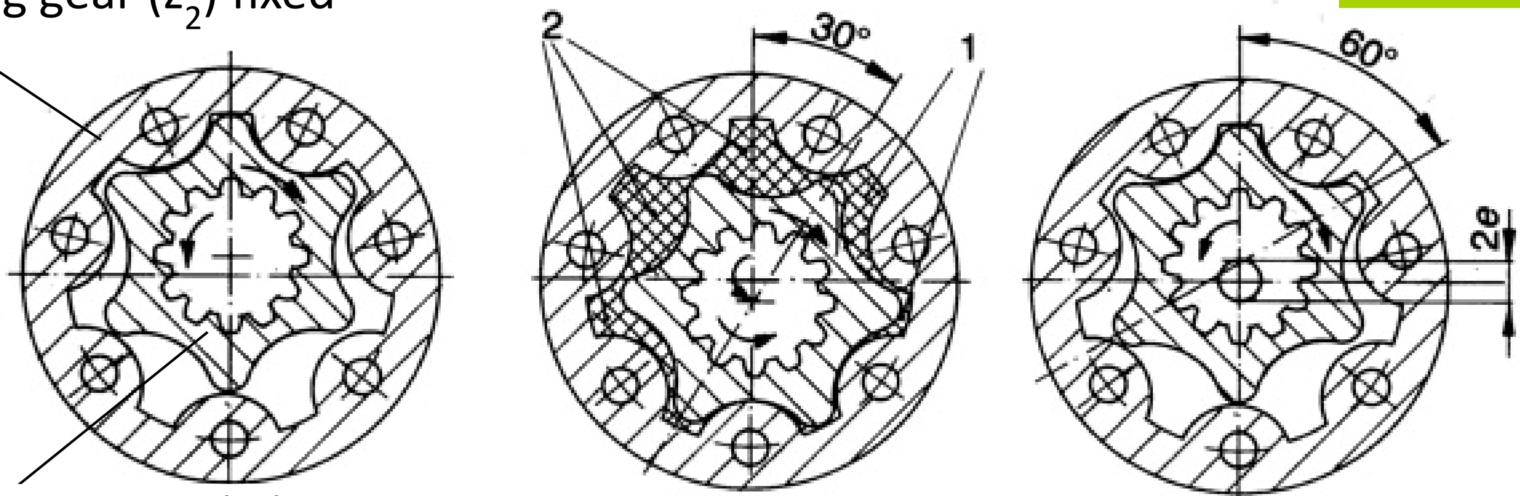
$$n_2 = n_1 \cdot \frac{z_1}{z_2} = n_1 \cdot \left(1 - \frac{1}{z_2}\right) = n_1 \cdot \left(1 + \frac{1}{z_1}\right)^{-1}$$

Αντλίες τύπου Gerotor

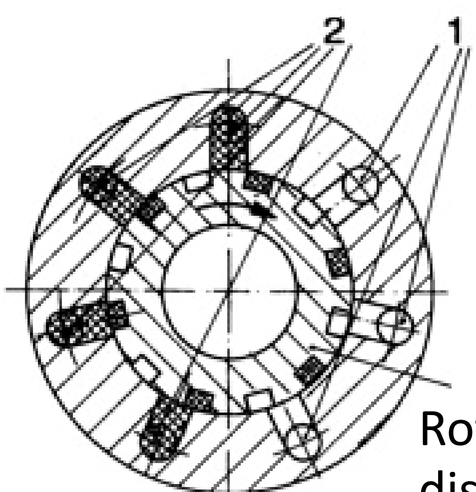
2 Pressure port

$z_2 = z_1 + 1$

Ring gear (z_2) fixed



Rotating pinion (z_1)

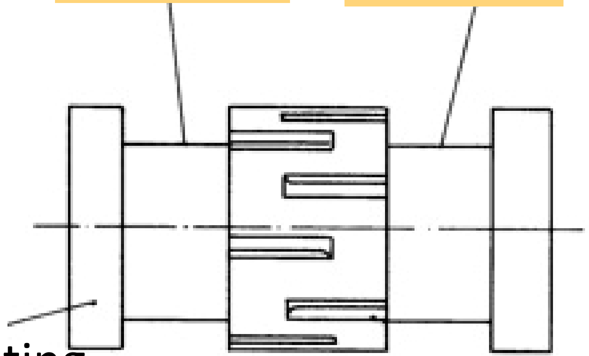


1 Suction port

Rotating distributor

Outlet

Inlet

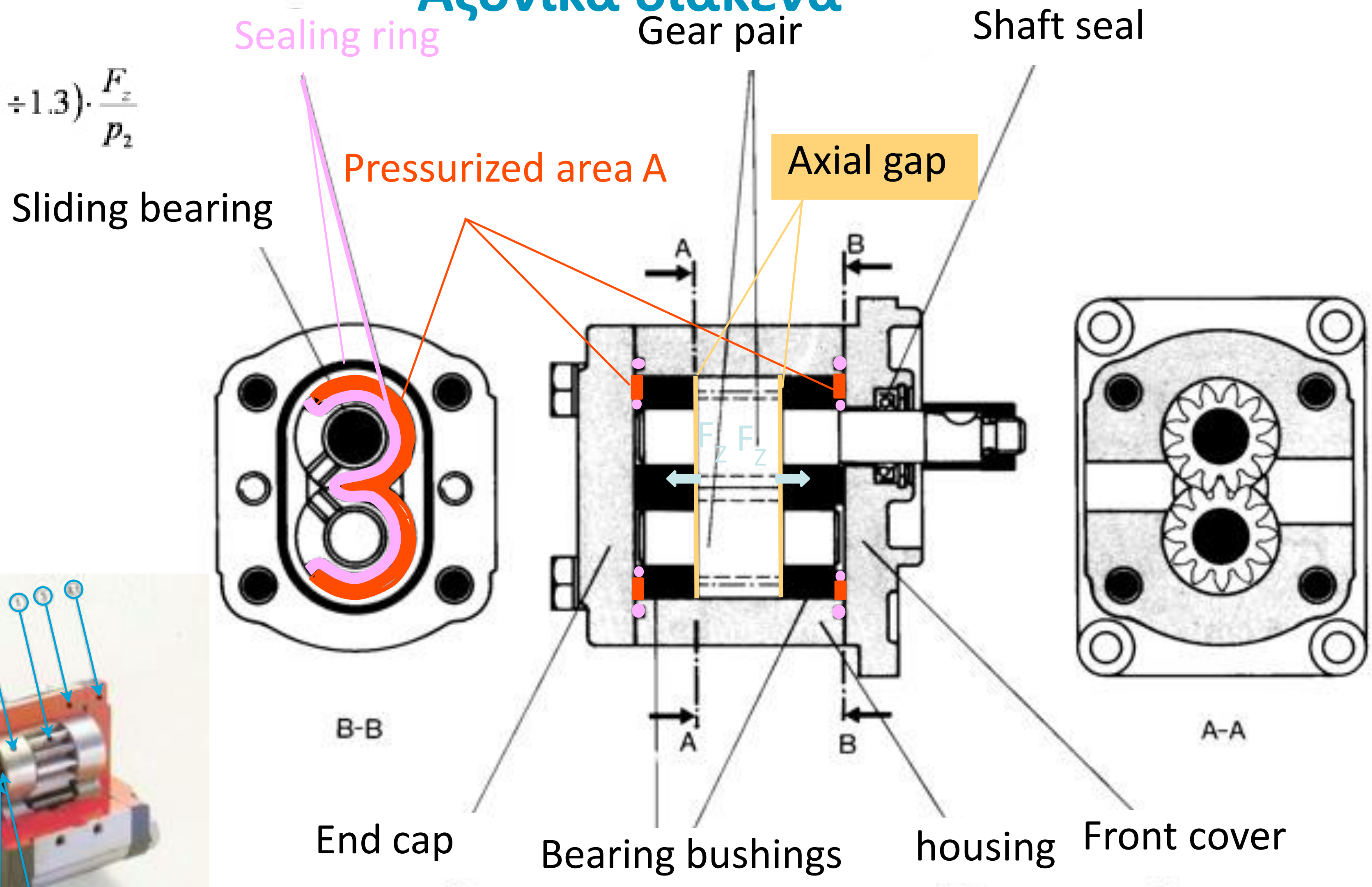


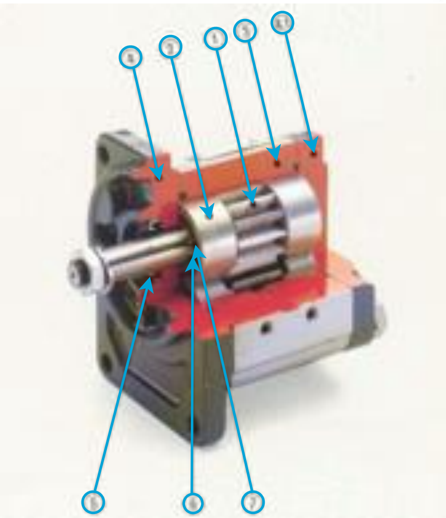
Displacement volume is given by z_1 times z_2 tooth spaces

Multiple delivery of each tooth space

Αξονικά διάκενα

$$A = (1.1 \div 1.3) \cdot \frac{F_z}{p_2}$$

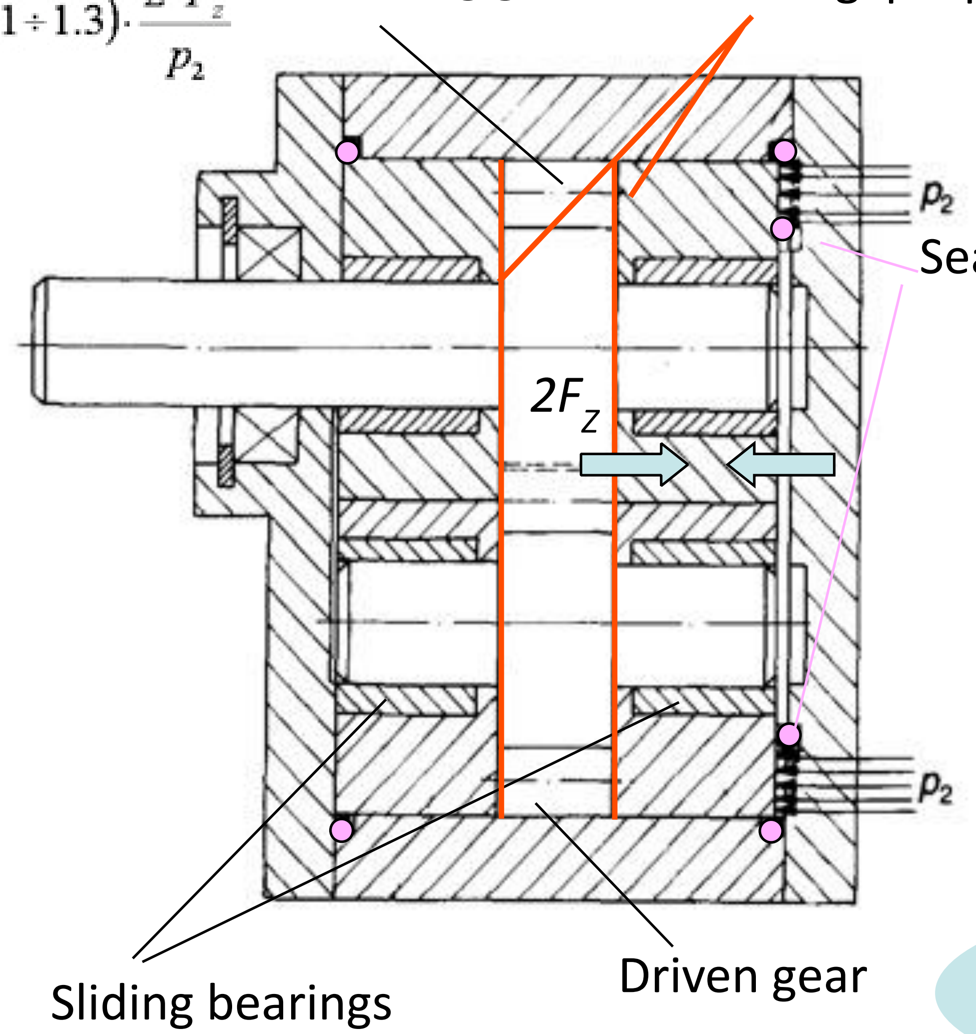


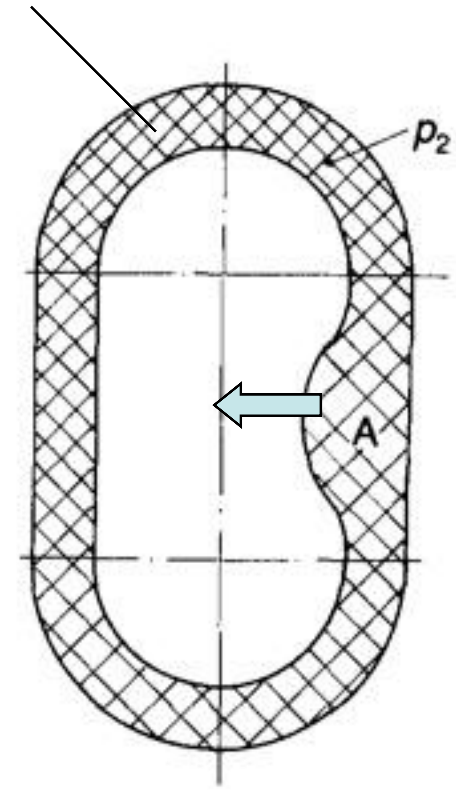


Only one direction of shaft rotation possible!

Αξονικά διάκενα

$$A = (1.1 \div 1.3) \cdot \frac{2 \cdot F_z}{p_2}$$

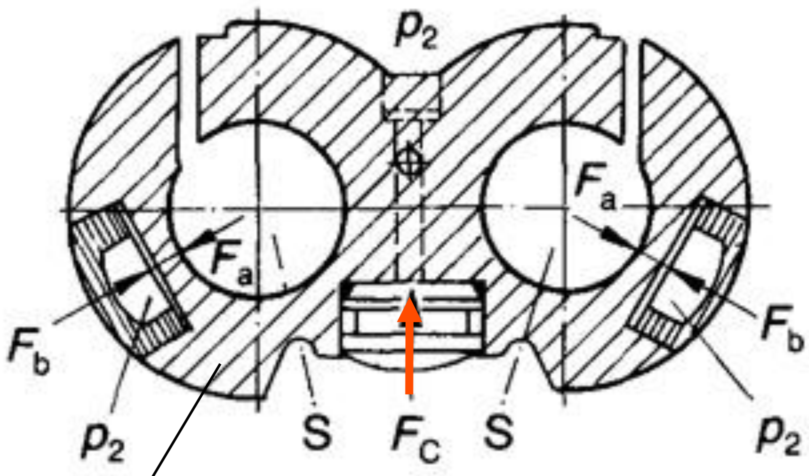




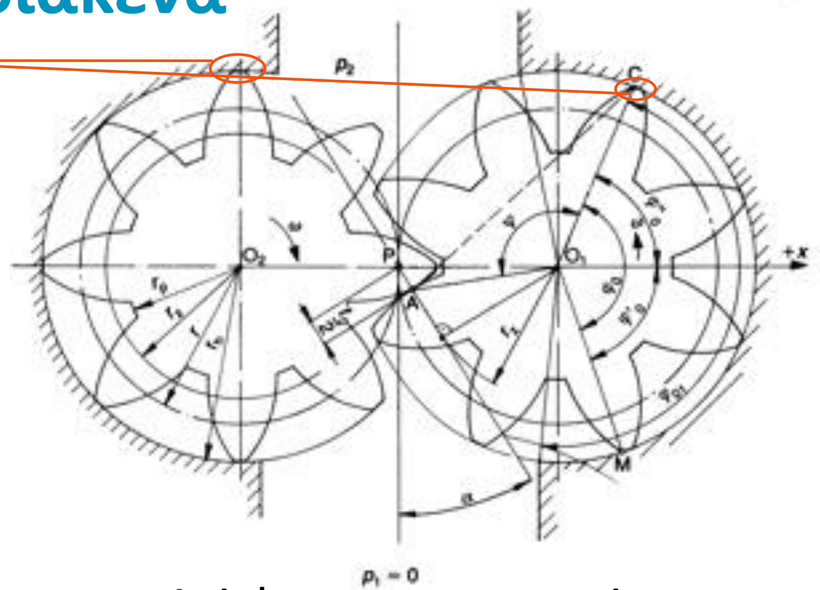
Improved volumetric efficiency

Ακτινικά διάκενα

Radial gap compensation

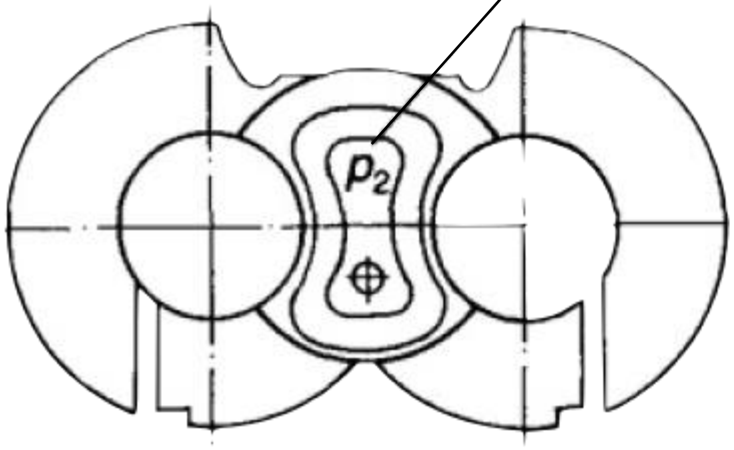


Bearing bushing



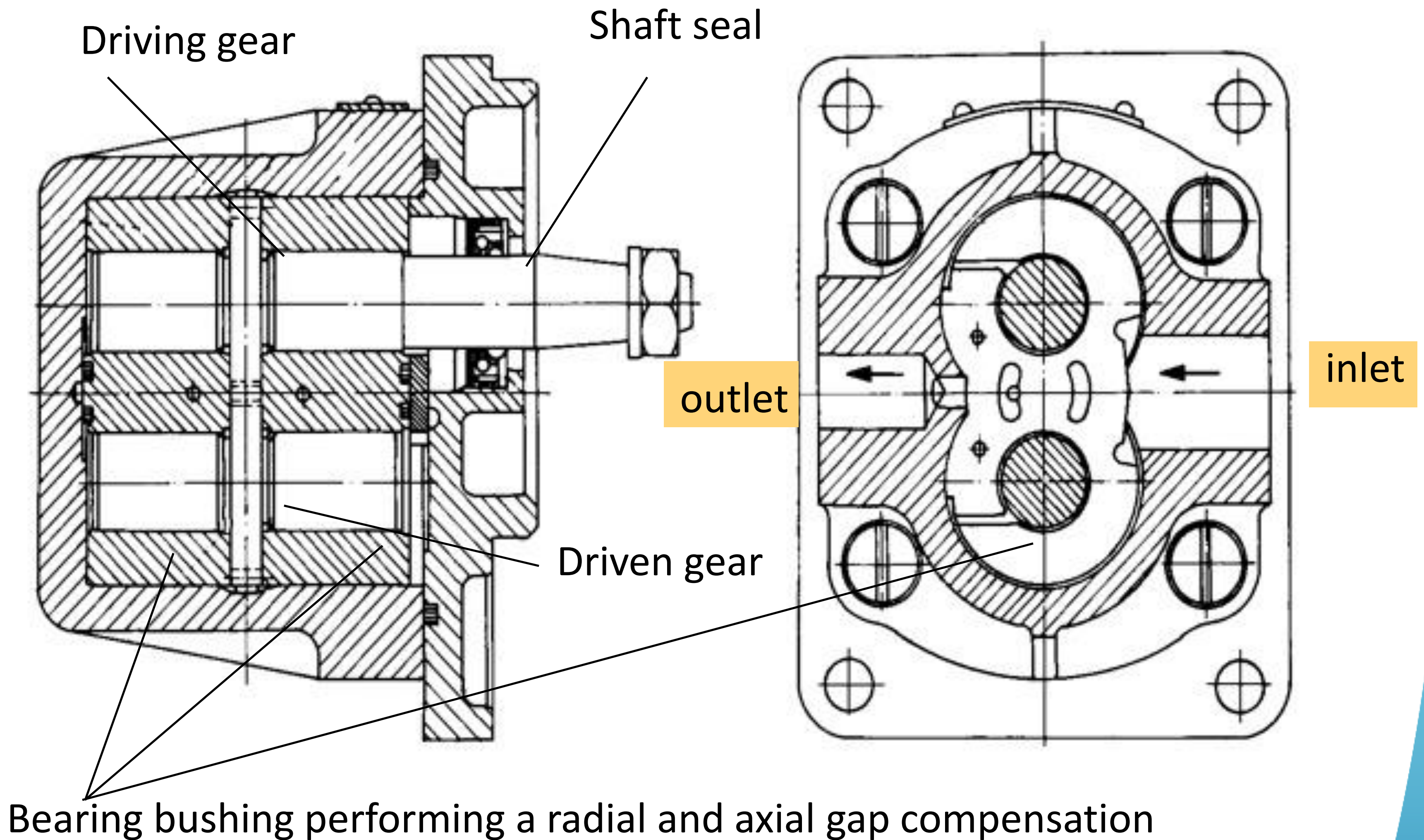
Axial gap compensation

Pressurized area



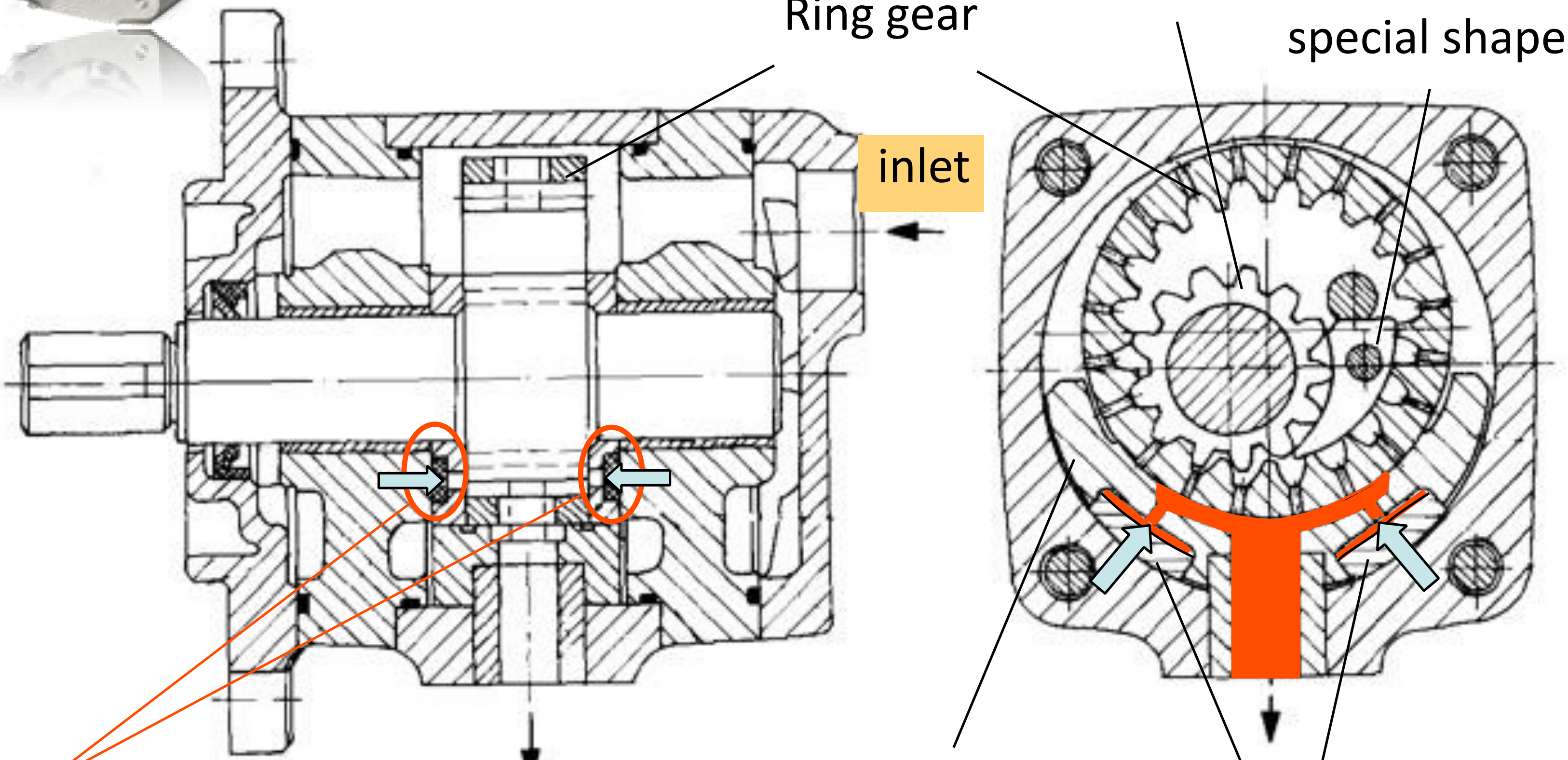
Small pressure zone achievable

Παράδειγμα σχεδιασμού



Παράδειγμα σχεδιασμού

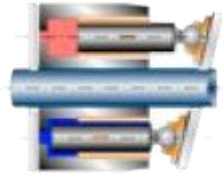
Internal gear pump with axial and radial gap compensation



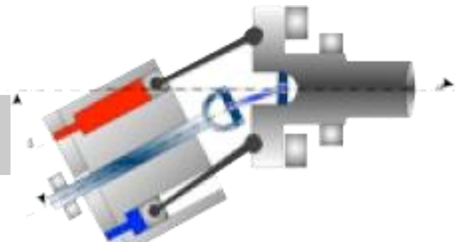
Μηχανές θετικής μετατόπισης

Piston Machines

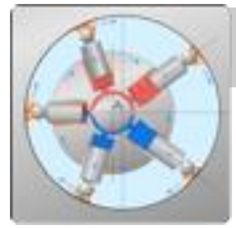
- Axial Piston Machines
  - Swash Plate Machines



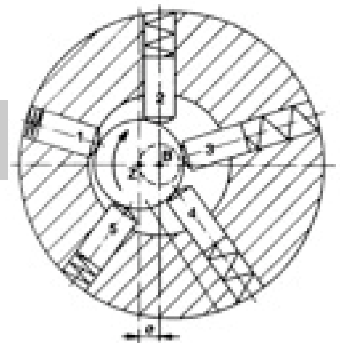
  - Bent Axis machines



- In-line Piston Machines
- Radial Piston Machines
  - with external piston support

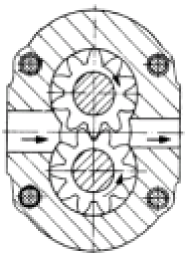


  - with internal piston support

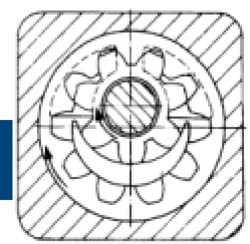


Gear Machines

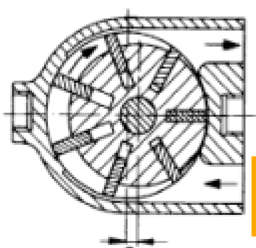
- External Gear



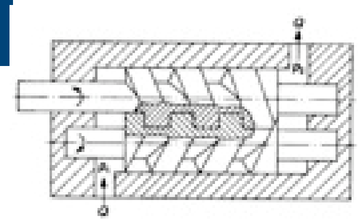
- Internal Gear



- Annual Gear



Vane Machines



Screw Machines

others

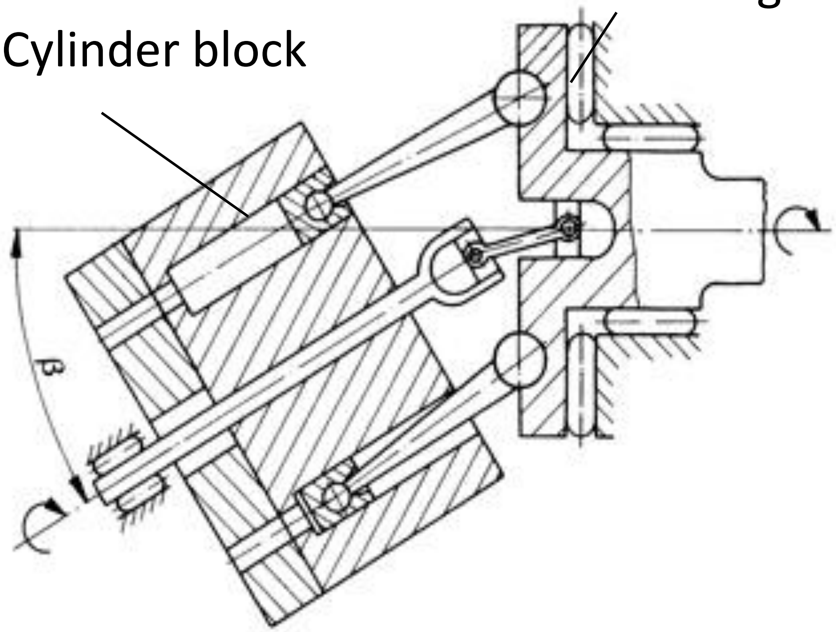
Fixed displacement machines ⟷ Variable displacement machines

Αντλίες αξονικών εμβόλων bent axis

Synchronization of cylinder block

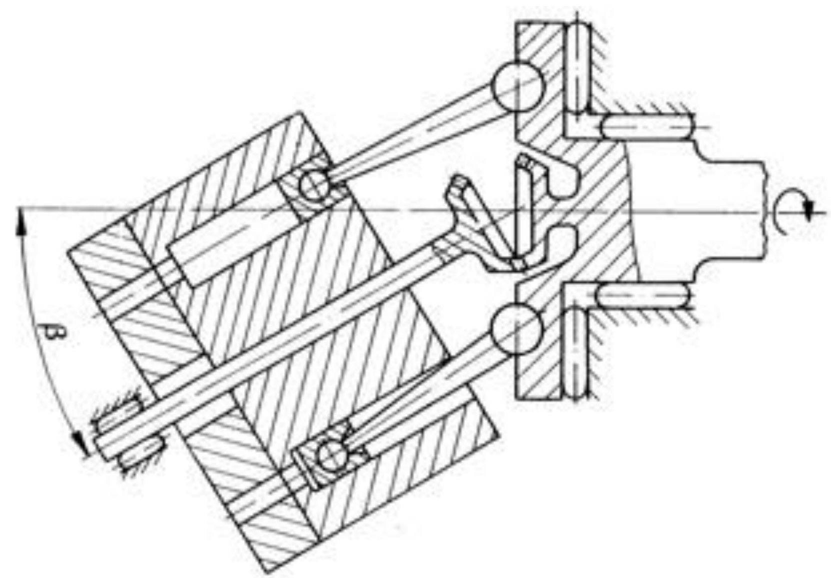
Cylinder block

Driving flange

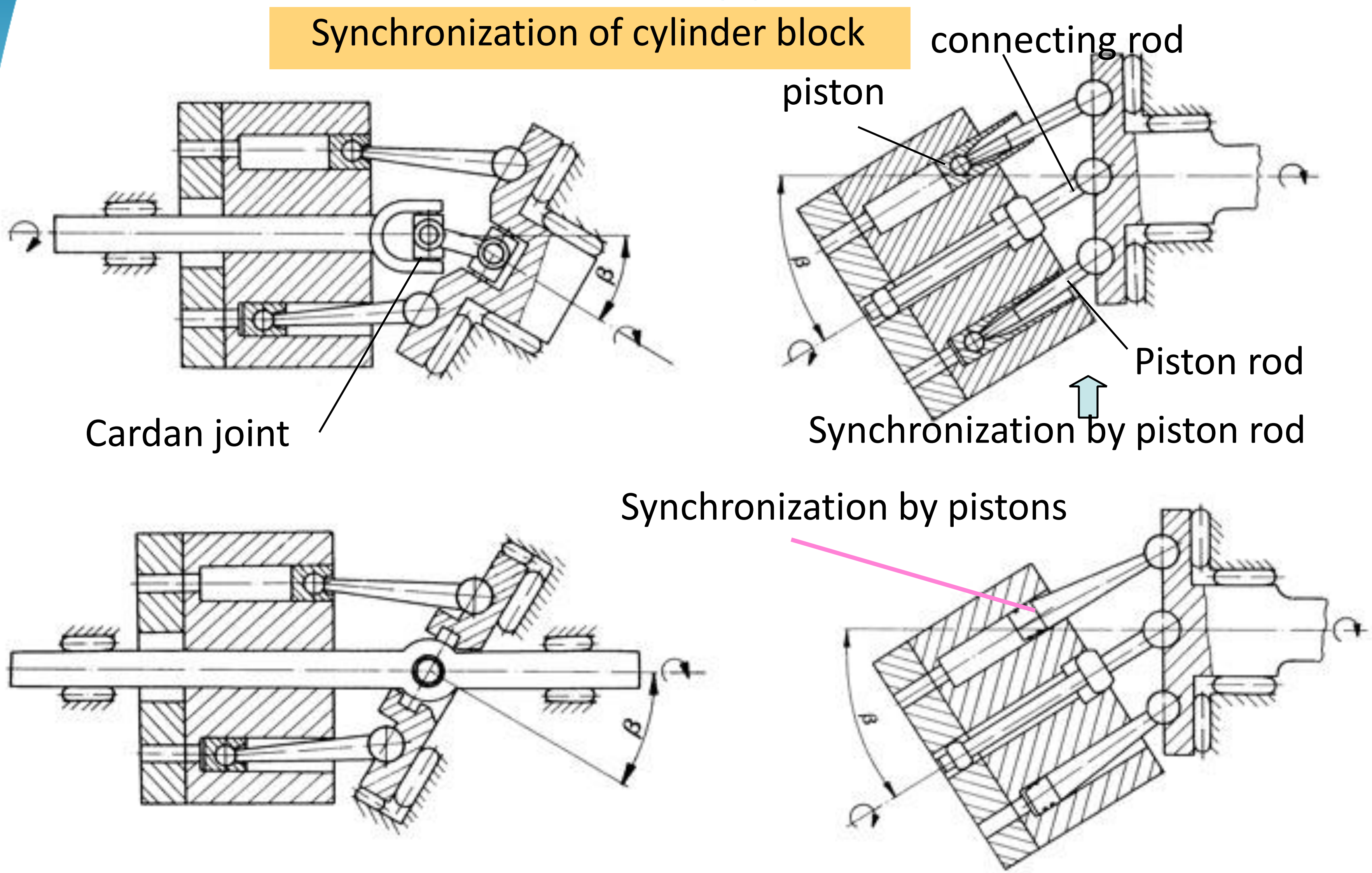


Using a universal joint

Using a bevel gear

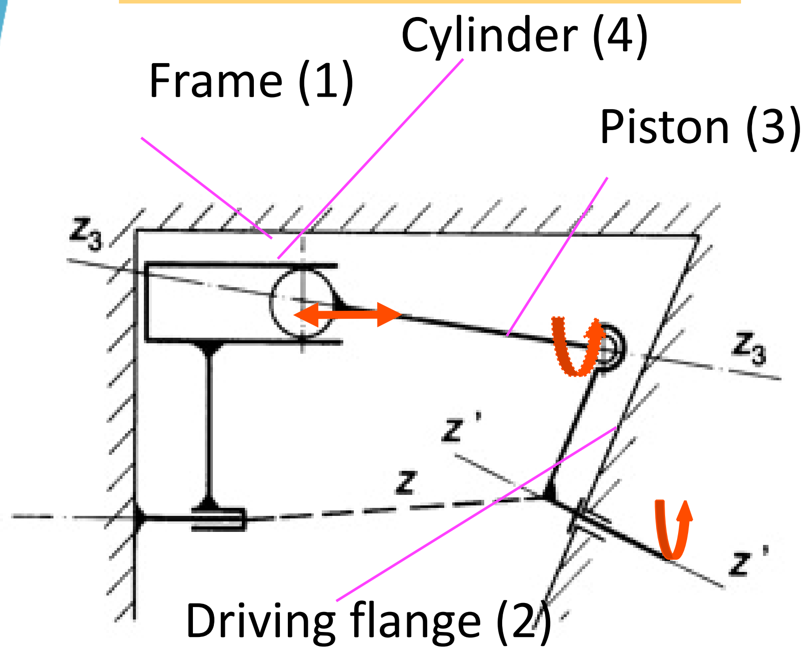


Αντλίες αξονικών εμβόλων bent axis

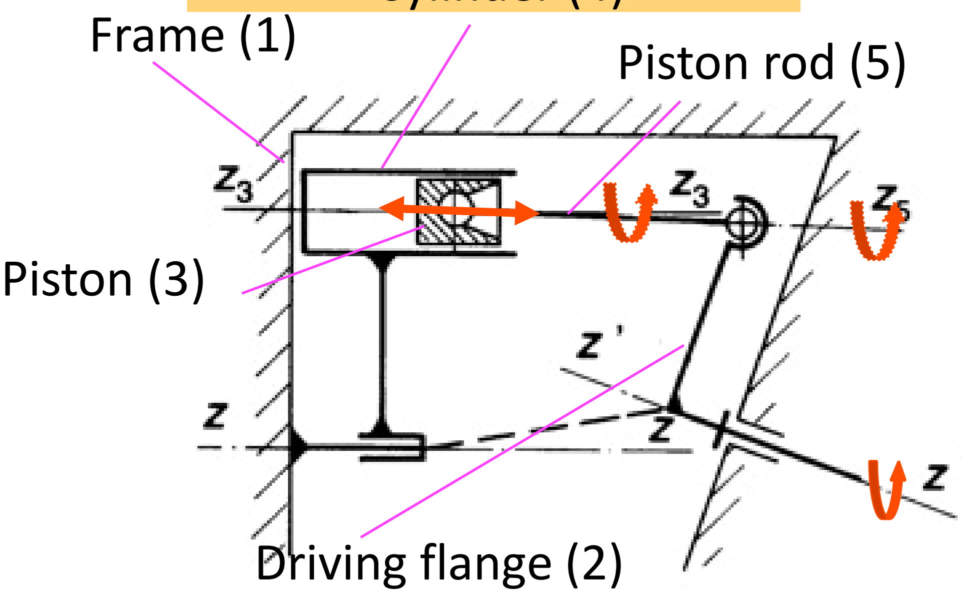


Αντλίες αξονικών εμβόλων bent axis-κινηματικά

Four link 3D mechanism



Five link 3D mechanism
Cylinder (4)



Assuming a fixed connection between link 2 and link 4, achieved by synchronization

mechanism has finally two degrees of freedom

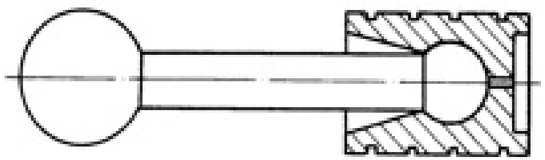
Piston can rotate about z_3-axis

the mechanism has finally three degrees of freedom

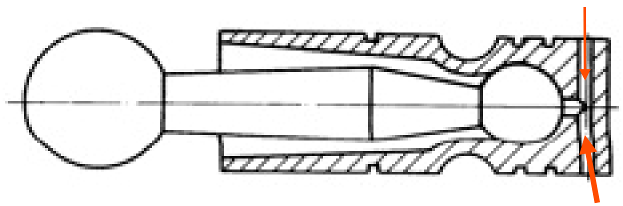
Piston can rotate about z_3-axis and piston rod can rotate about z_5-axis

Σχεδιασμός εμβόλου

Short piston with piston rod



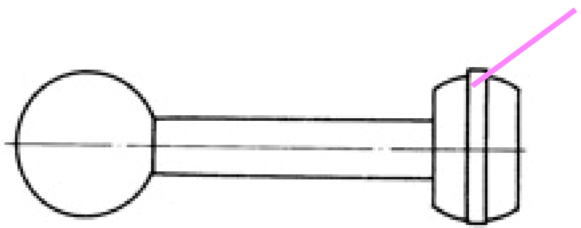
Long piston with piston rod



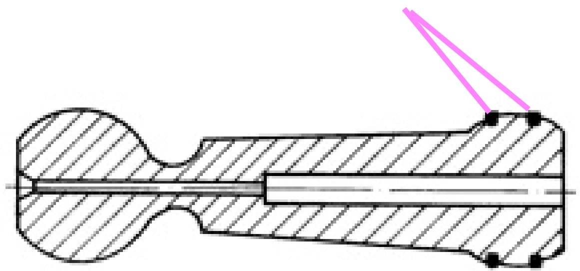
Synchronization by
universal joint or bevel gear

Synchronization by
pistons or piston rods

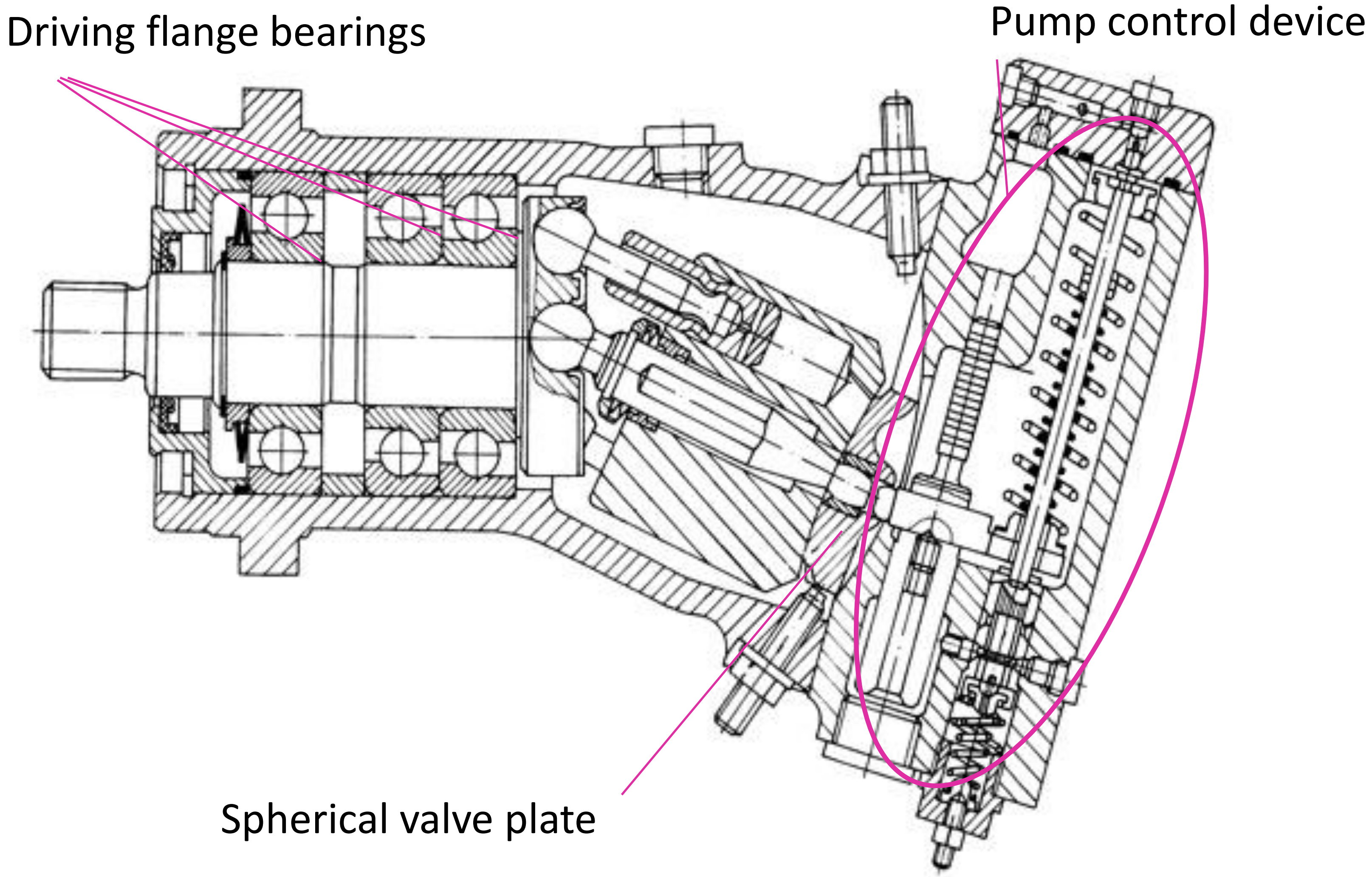
Spherical piston with piston ring



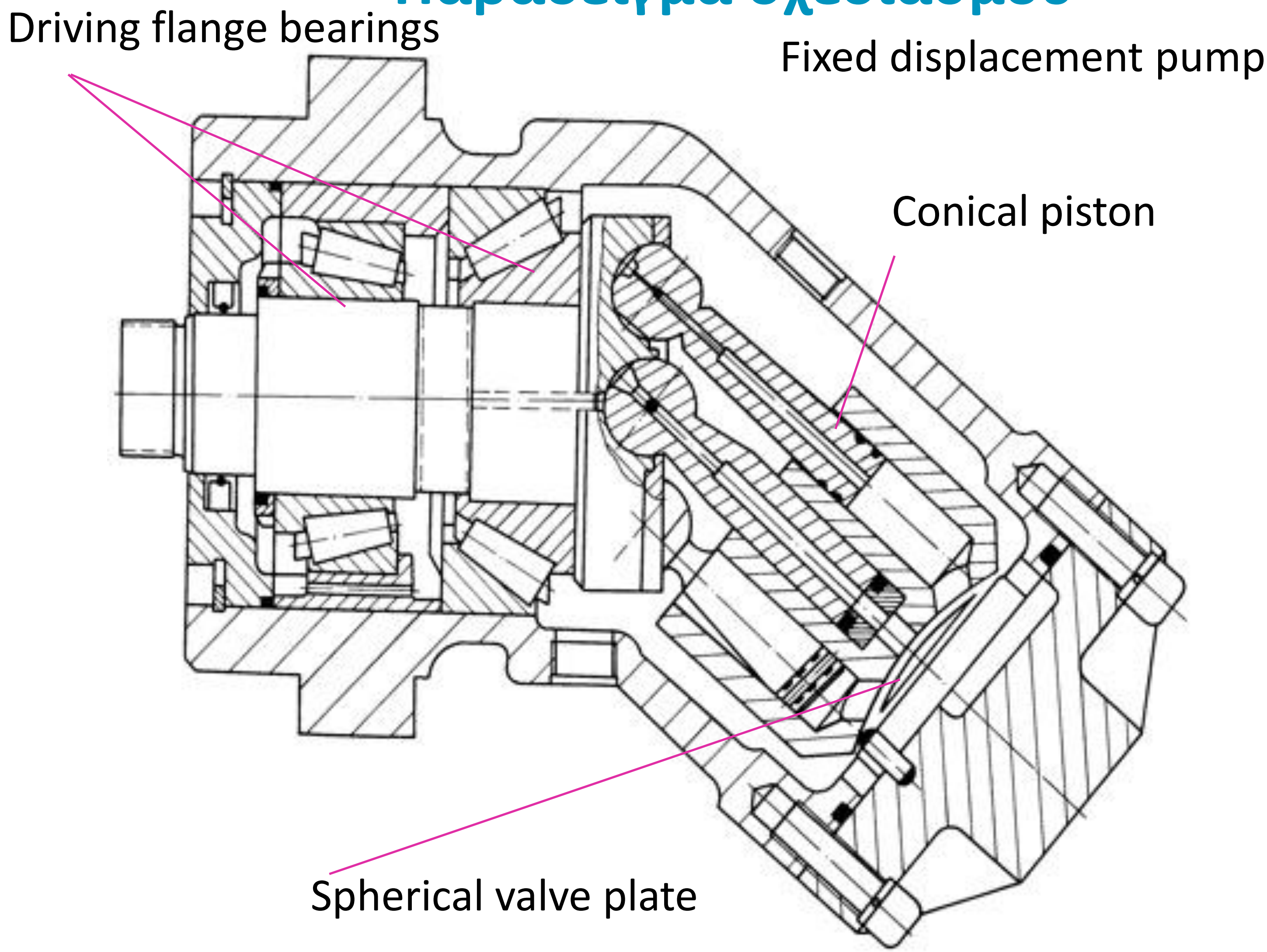
Conical piston with piston rings



Παράδειγμα σχεδιασμού



Παράδειγμα σχεδιασμού



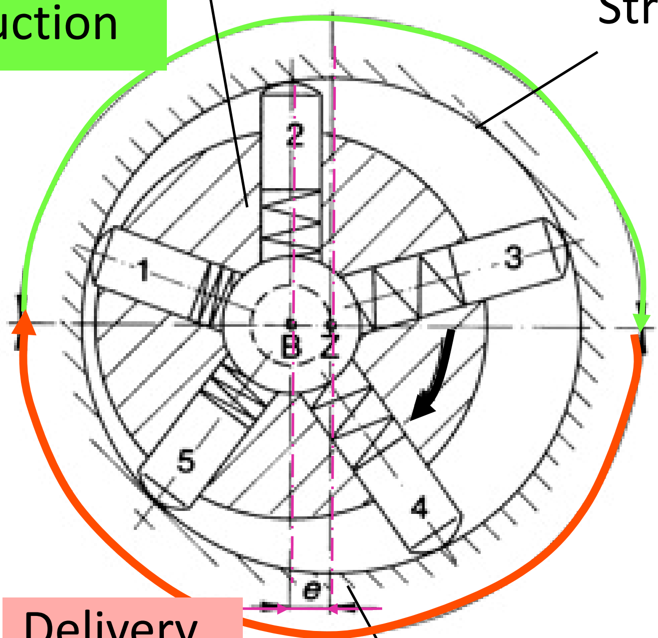
Αντλίες ακτινίκων εμβόλων

with external piston support

Rotating cylinder body

Stroke ring

Suction



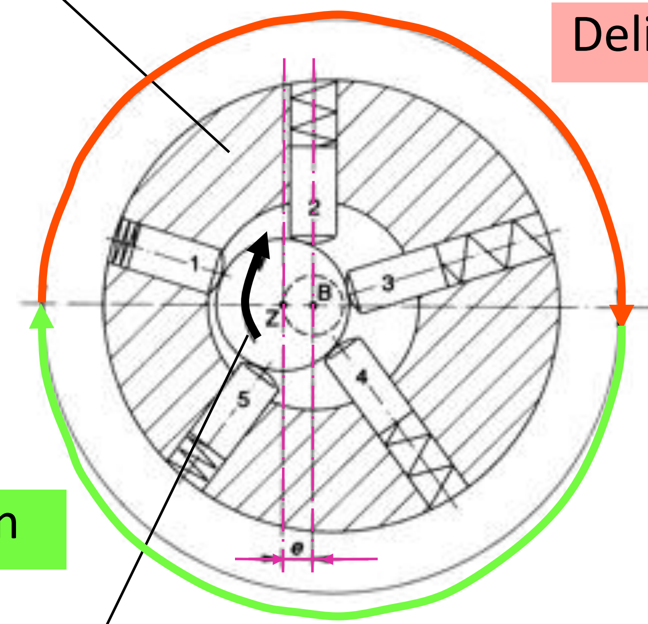
Delivery

eccentricity

with internal piston support

Stationary cylinder body

Delivery



Suction

Rotating cam or crankshaft

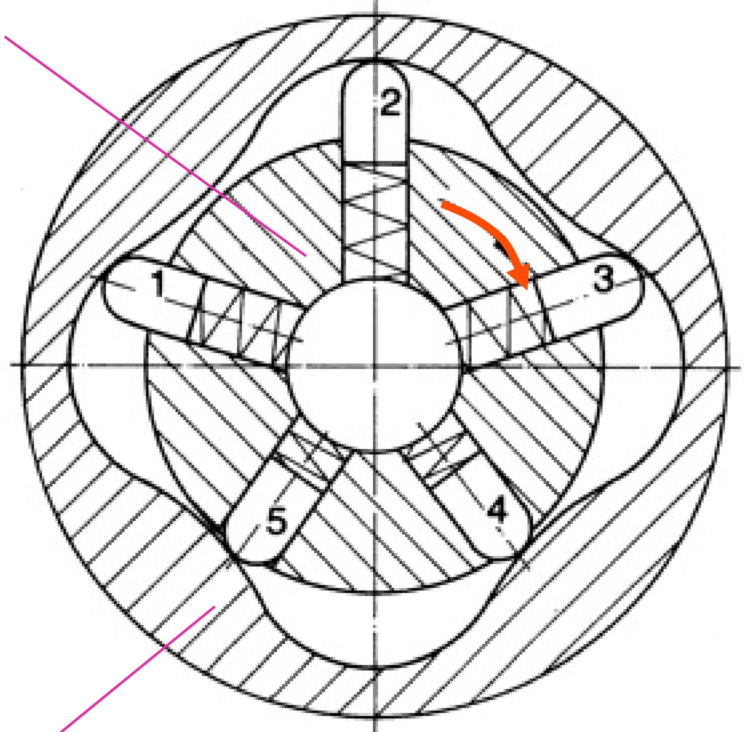
Displacement volume adjustable by changing eccentricity e

Radial Piston Pumps

Multiple stroke radial piston pumps

with external piston support

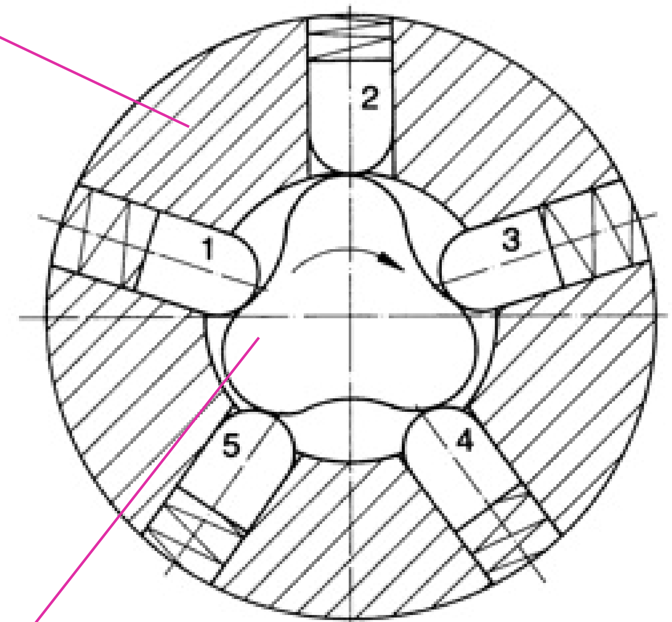
Rotating cylinder body



Stationary stroke ring

with internal piston support

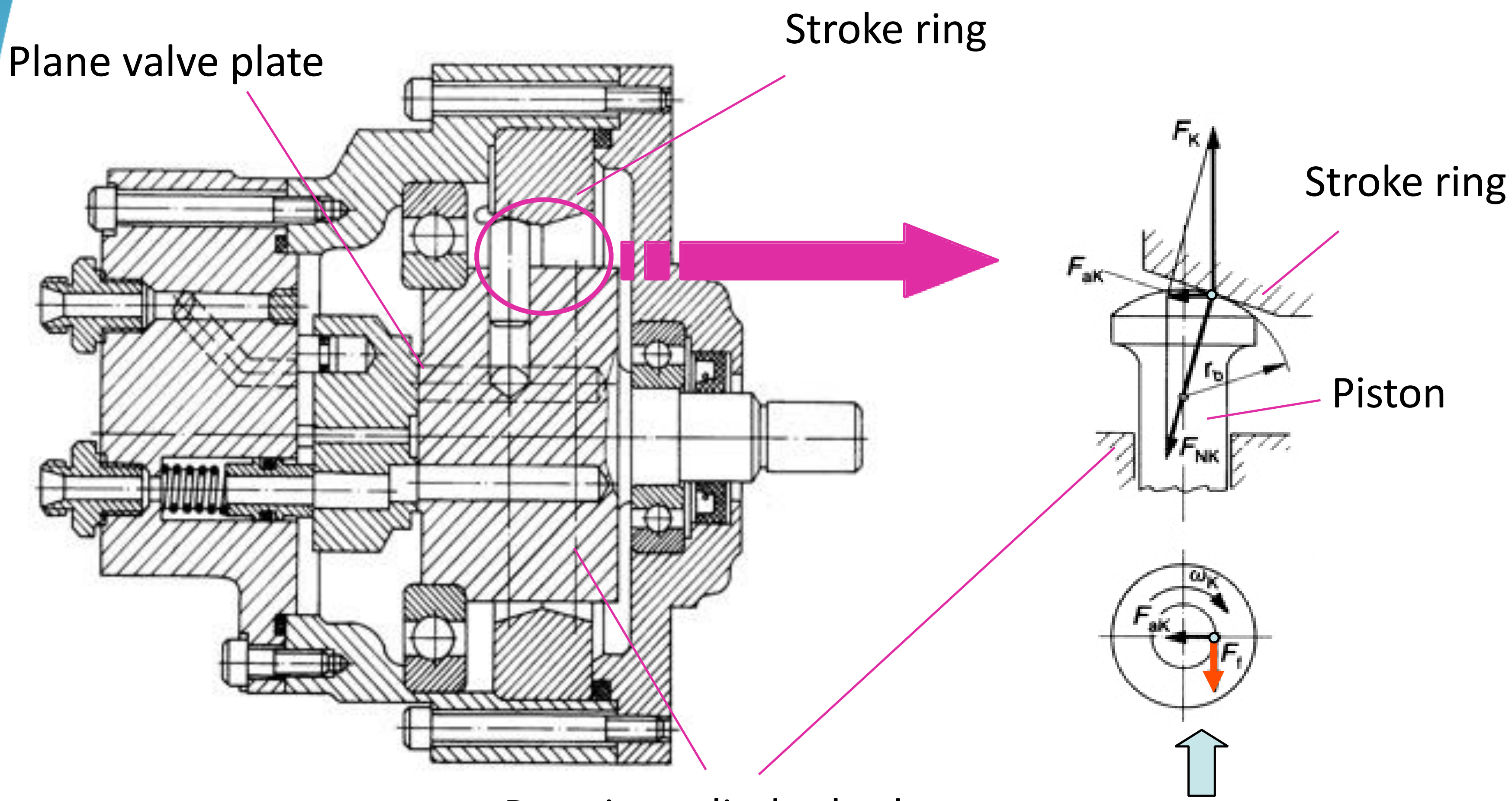
Stationary cylinder body



Rotating cam

Only fixed displacement pumps realizable!

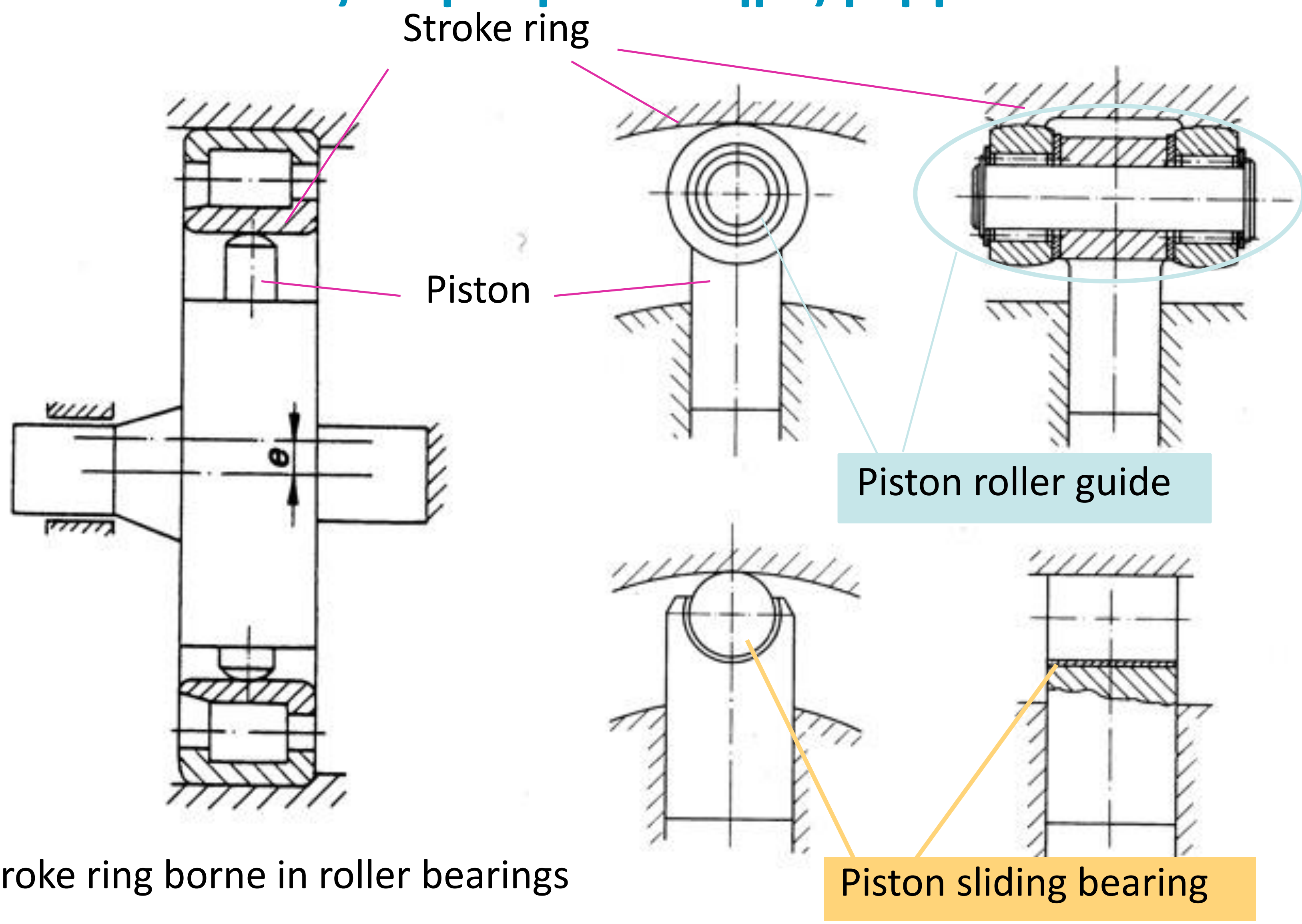
Εξωτερική υποστήριξη εμβόλων



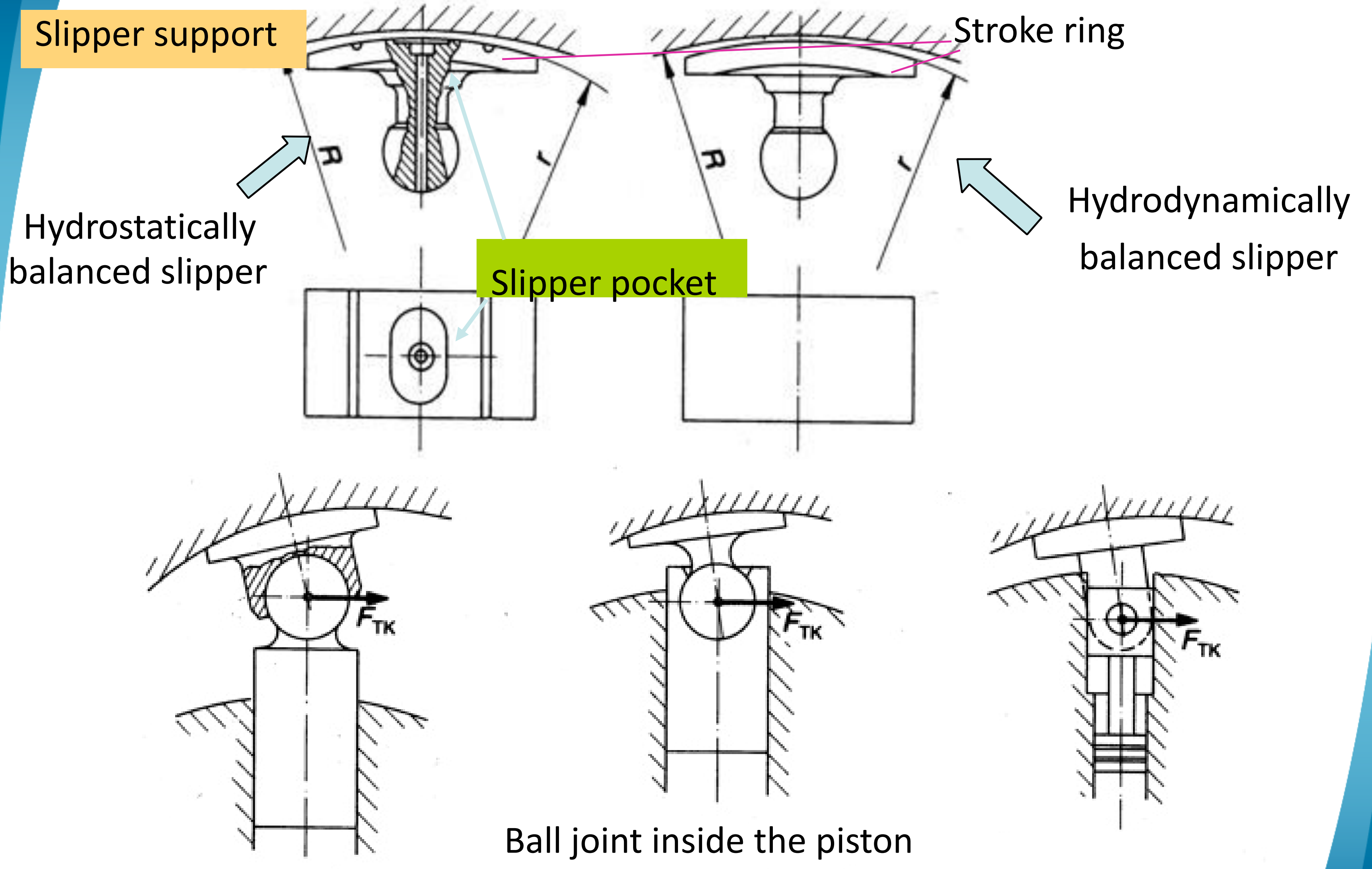
Piston rotation enforced by friction force F_f

Pison support on outer stroke ring

Εξωτερική υποστήριξη εμβόλων



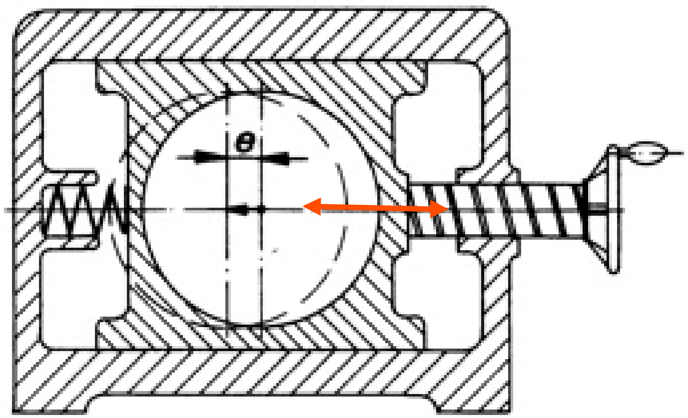
Εξωτερική υποστήριξη εμβόλων



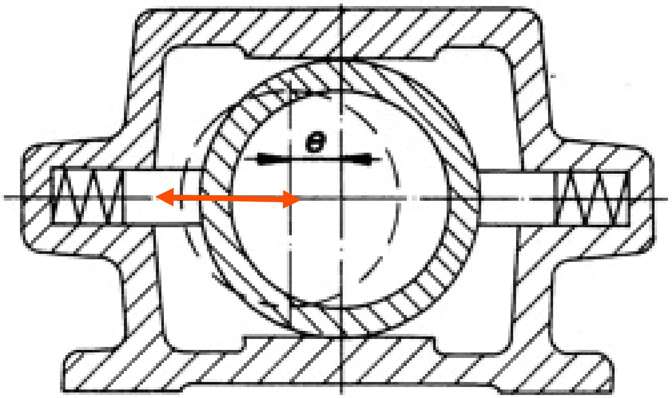
Ball joint inside the piston

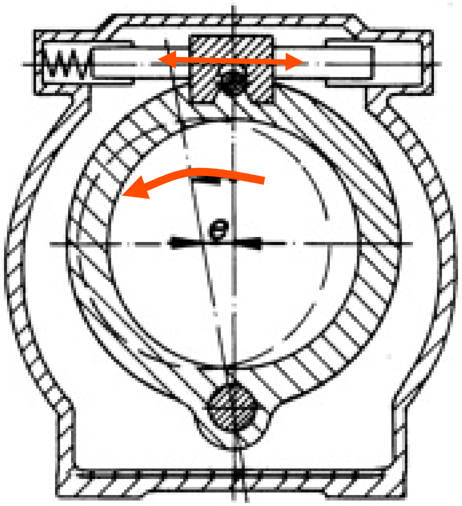
Εναλλακτική υποστήριξη

Using a sliding carriage



supported using line contact

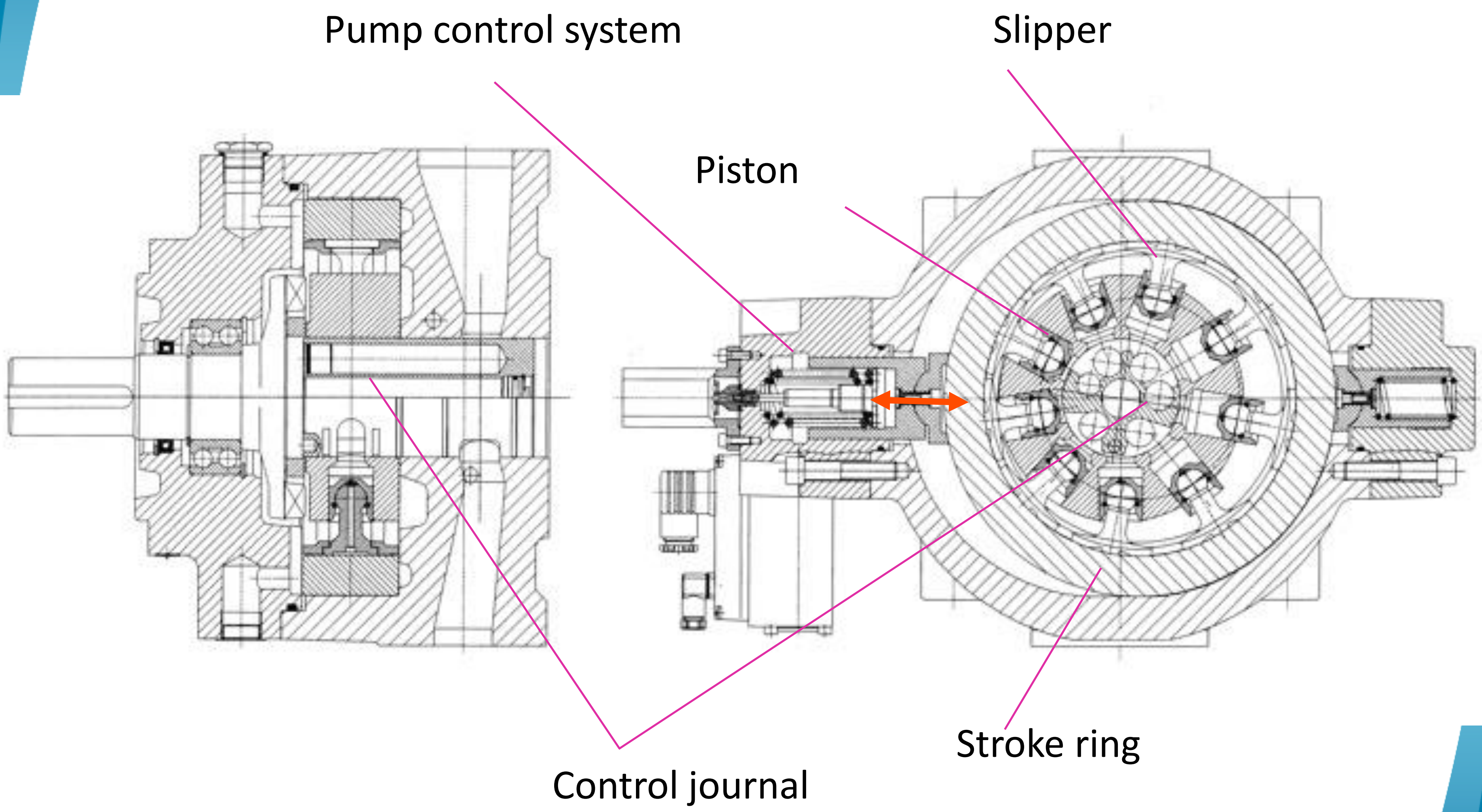




Stroke ring mounted on a pivot

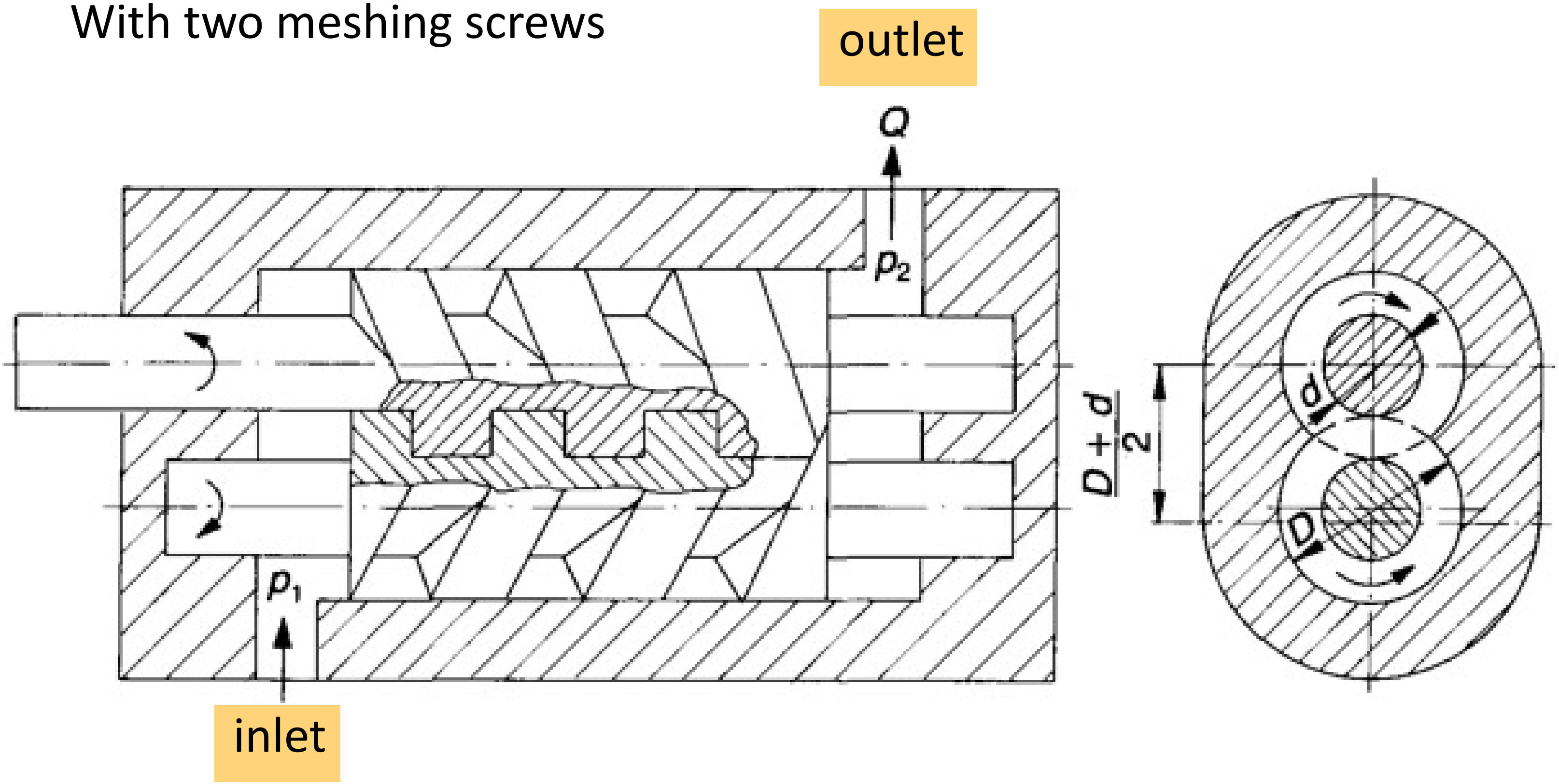
Change of eccentricity by pivoting the stroke ring about pivot axis

Παράδειγμα σχεδιασμού

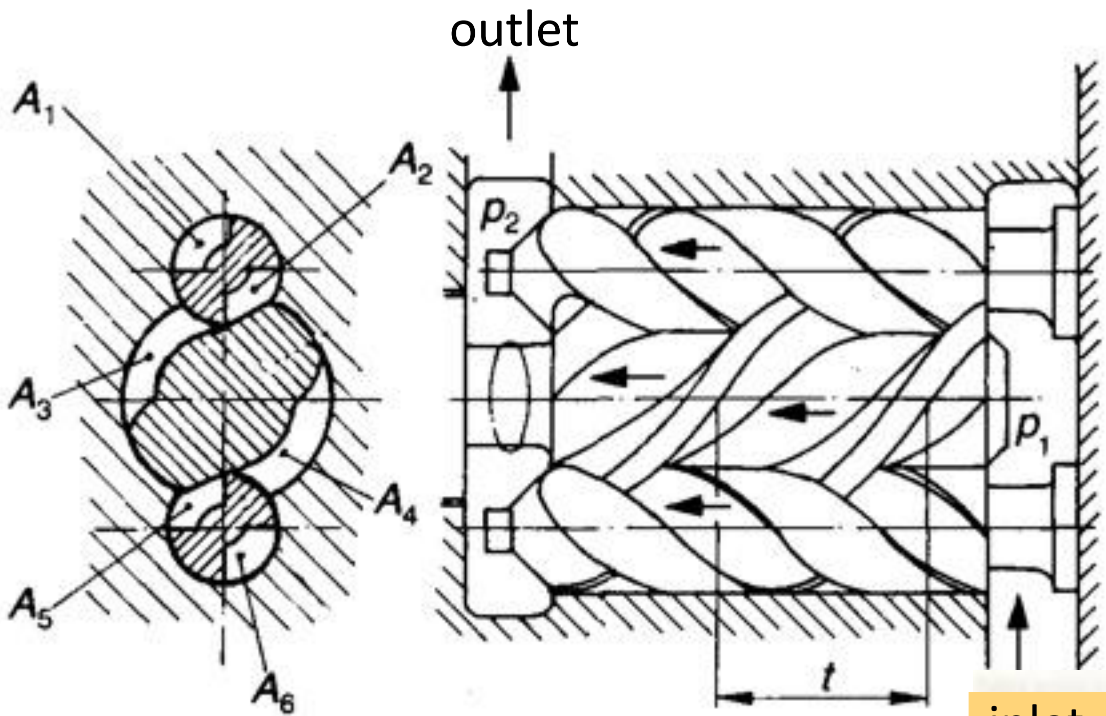


Κοχλιωτή αντλία

With two meshing screws

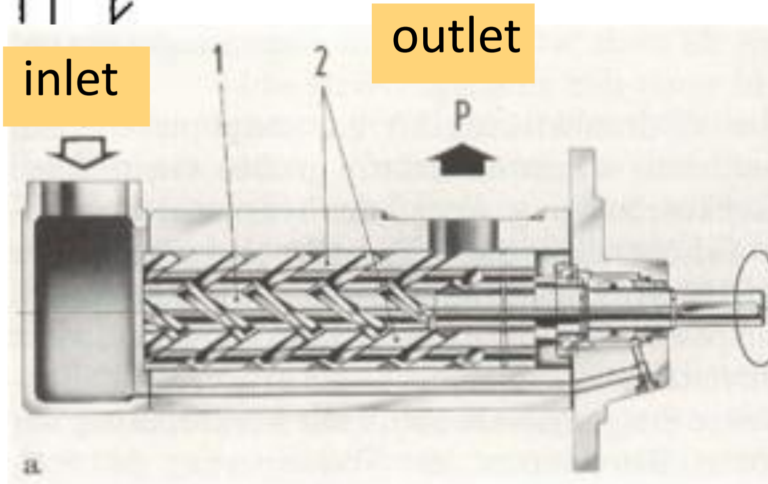


Κοχλιωτή αντλία



t…thread pitch

With three meshing screws



Κοχλιωτή αντλία

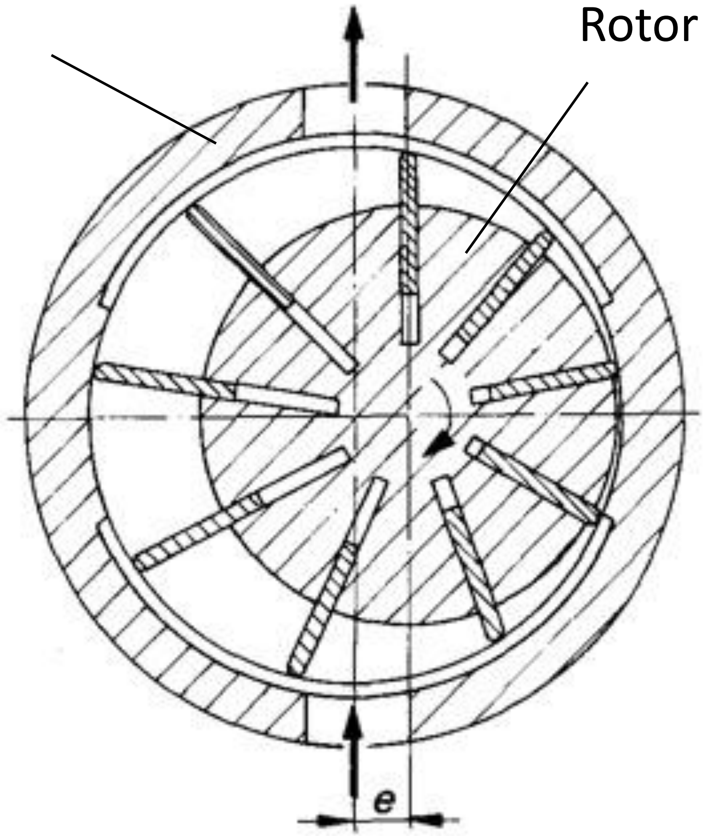
- Σελίδα 14 διάλεξη 7

Πτερυγιοφόρος αντλία

Unbalanced vane pump

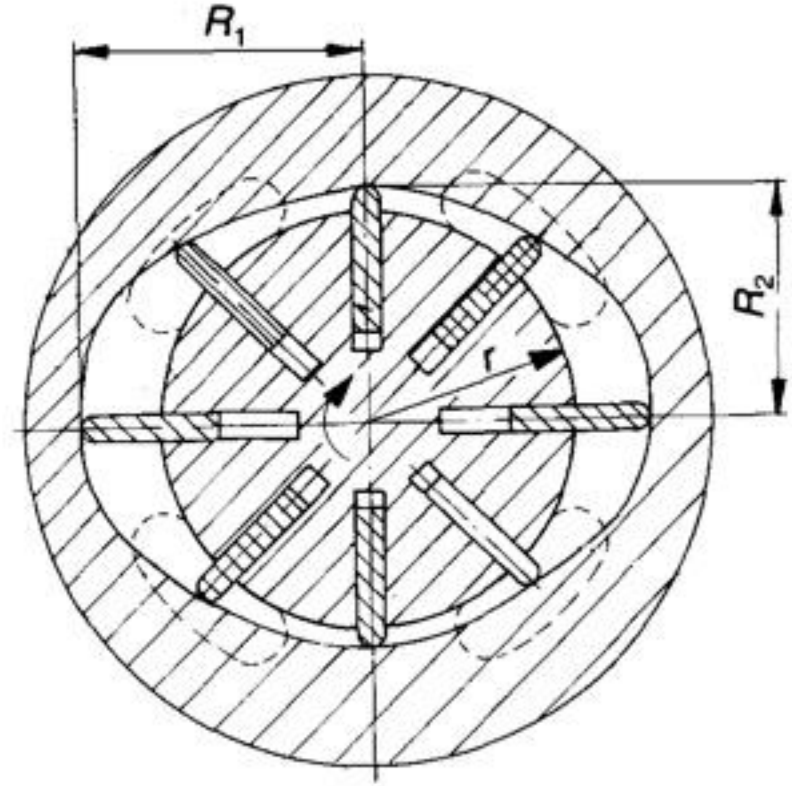
Stator

Rotor



Fixed and variable pump design

Balanced vane pump



Only fixed displacement pump

Πτερυγιοφόρος αντλία

Single stroke vane pump – variable displacement volume

Overcenter pump – the direction of flow can be re-versed by change of eccentricity, i.e. without changing the direction of rotation of the drive shaft

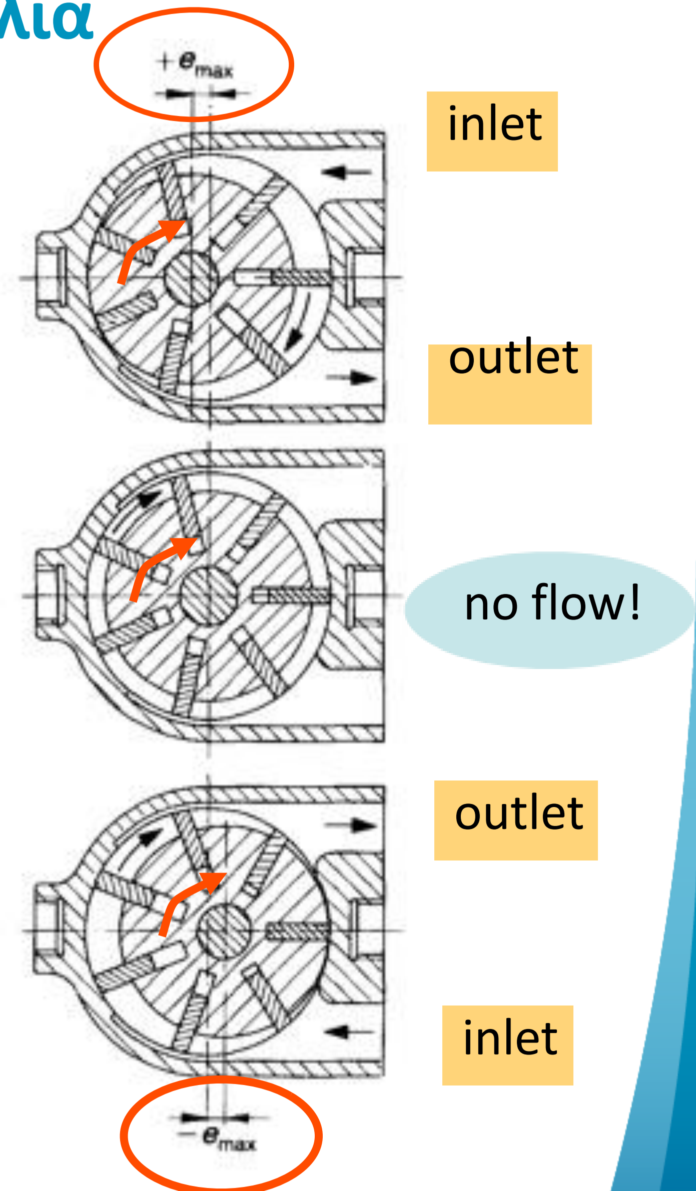
Relatively high friction between axial moveable vanes and rotor

&

between vanes and stator

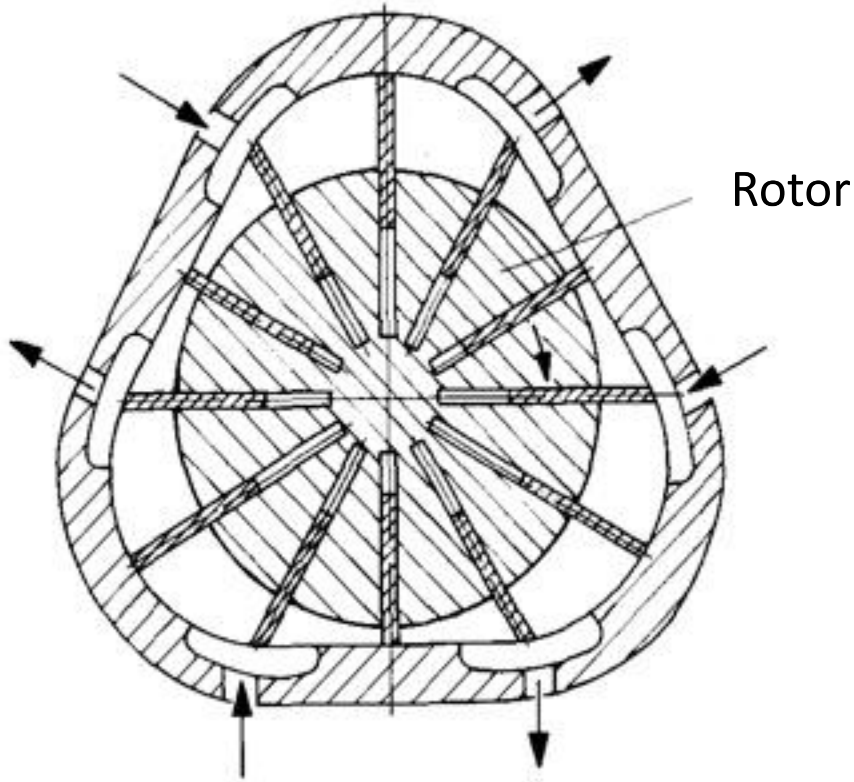
Large radial forces exerted on the rotor

Limitation of max. operating pressure (20 MPa)

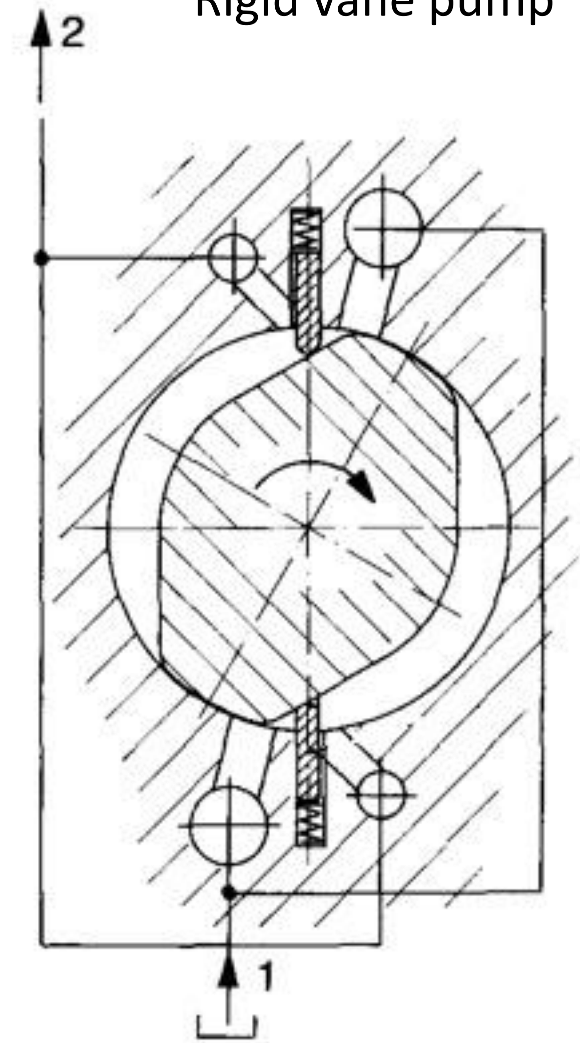


Πτερυγιοφόρος αντλία

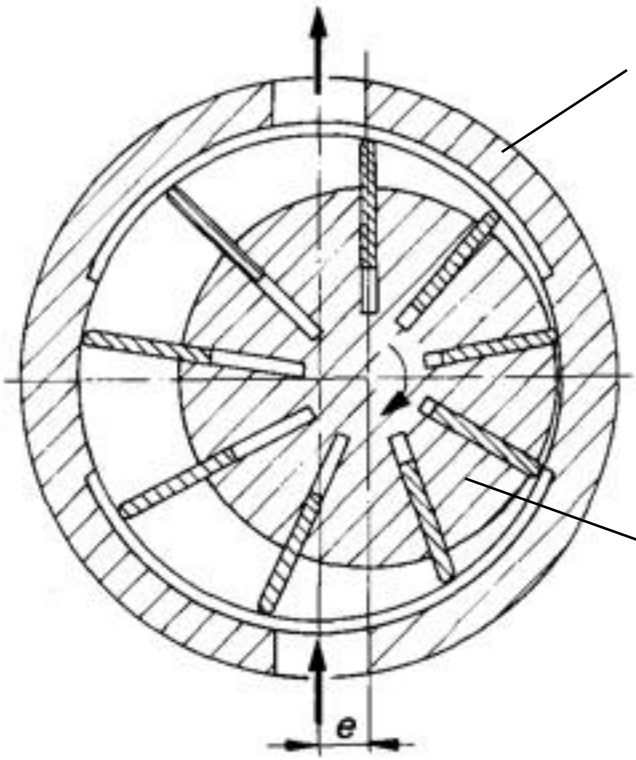
Multiple stroke vane pump

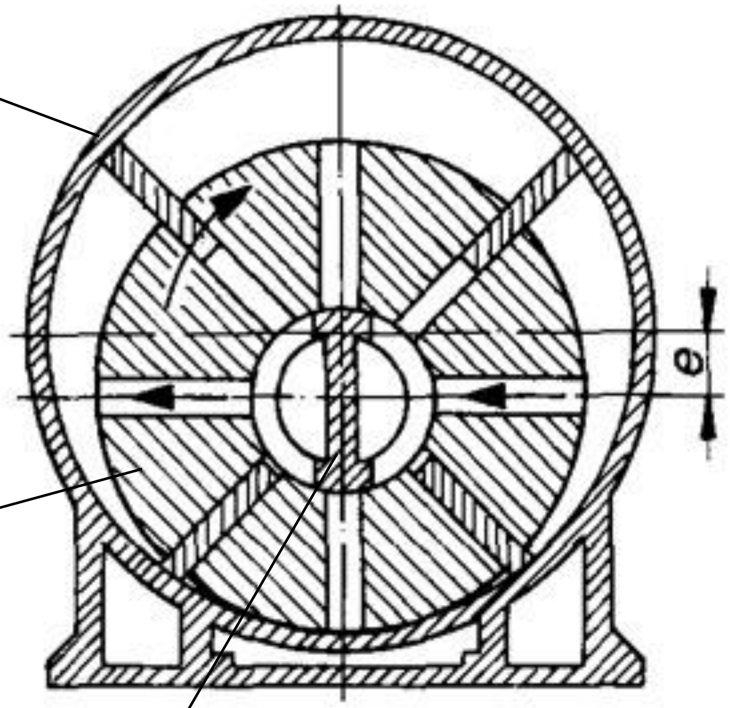


Rigid vane pump



Πτερυγιοφόρος αντλία

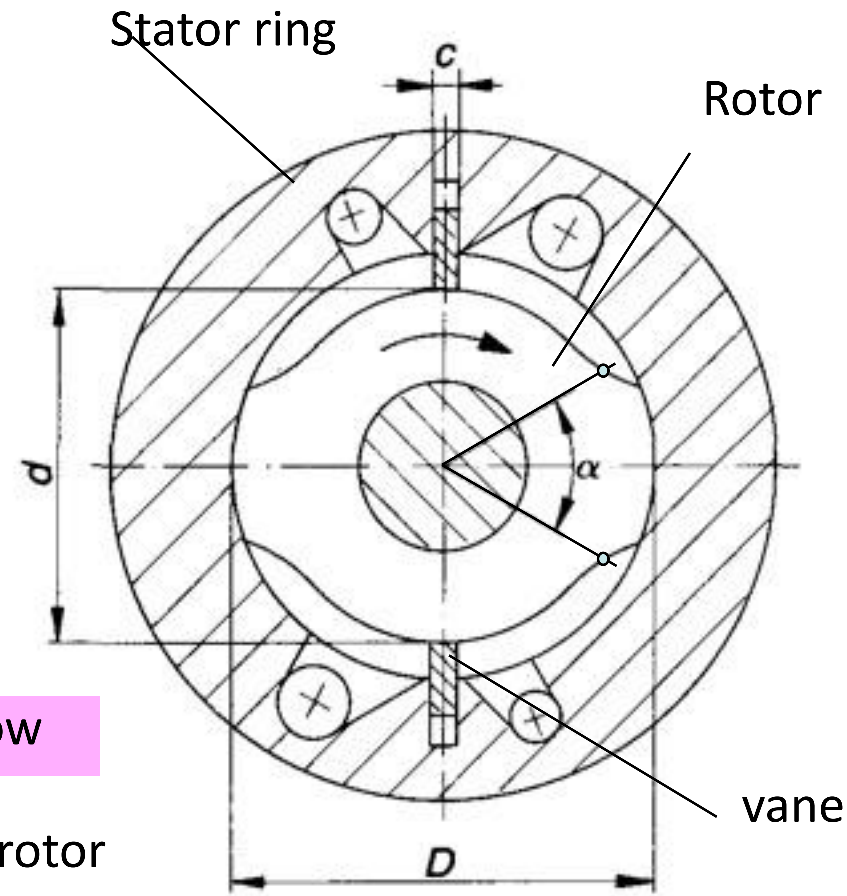


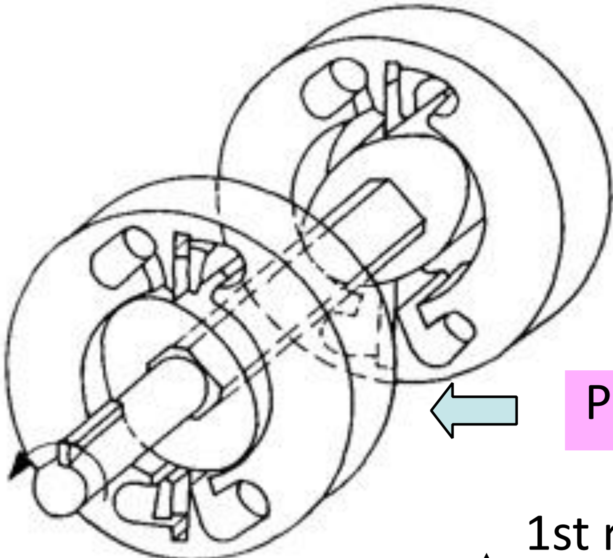


Πτερυγιοφόρος αντλία

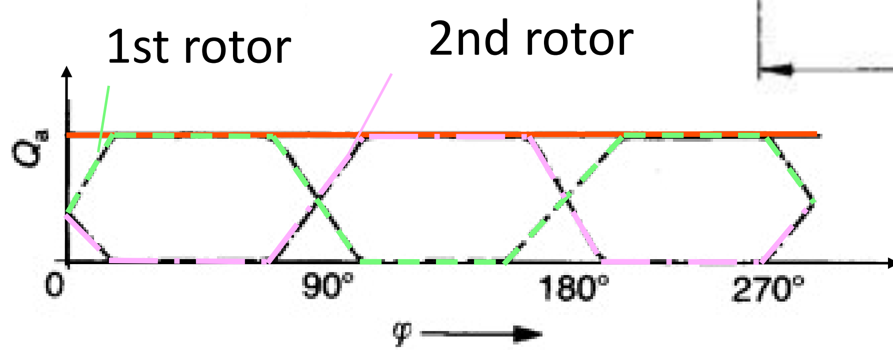
Displacement volume:

$$V_g = 2 \cdot \frac{\pi \cdot (D^2 - d^2)}{4} \cdot \frac{180 - \alpha}{180} \cdot b - 2 \cdot \left(\frac{D}{2} - \frac{d}{2}\right) \cdot c$$





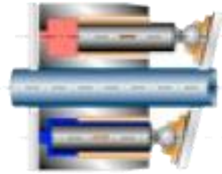
Pulsation free flow



Μηχανές θετικής μετατόπισης

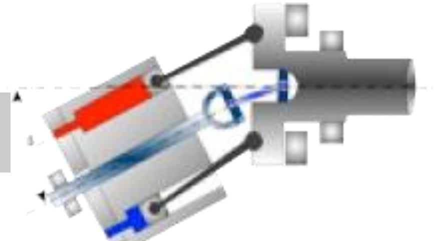
Piston Machines

Axial Piston Machines



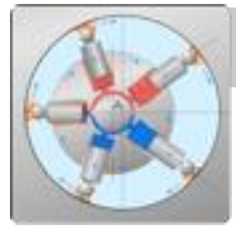
Swash Plate Machines

Bent Axis machines



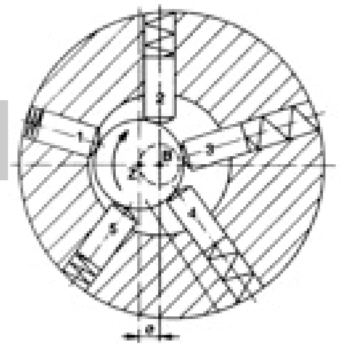
In-line Piston Machines

Radial Piston Machines

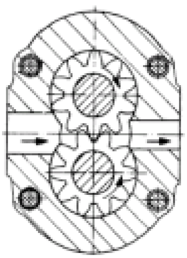


with external piston support

with internal piston support

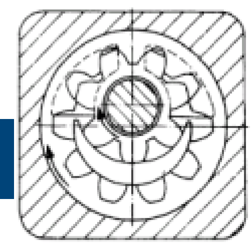


Gear Machines

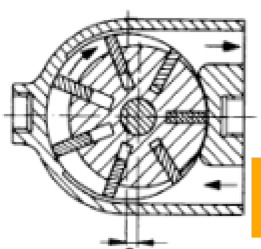


External Gear

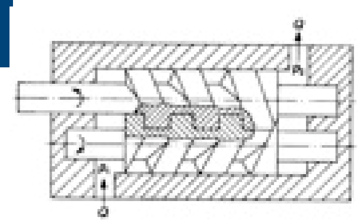
Internal Gear



Annual Gear



Vane Machines



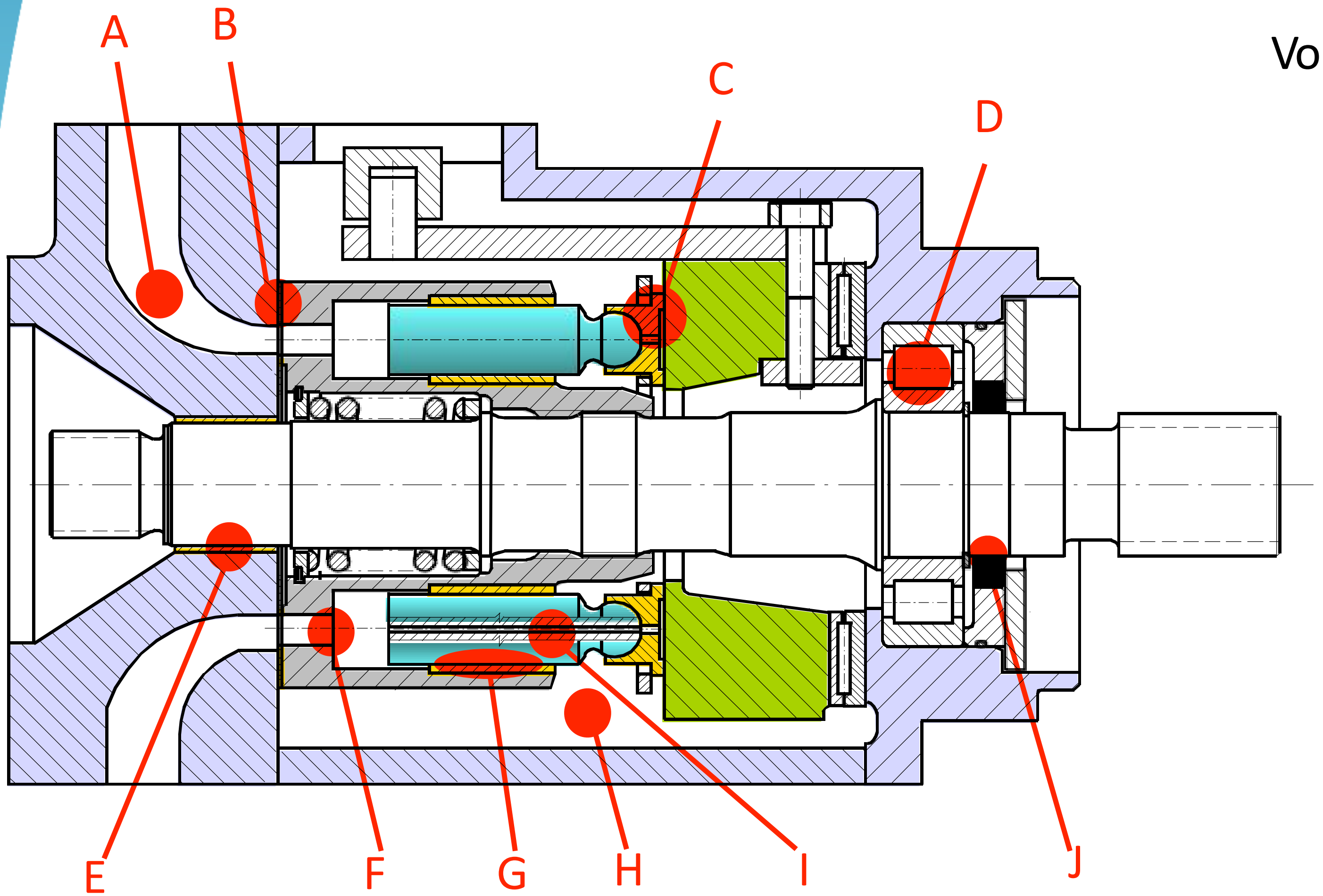
Screw Machines

others

Fixed displacement machines ⟺ Variable displacement machines

Απώλειες στις αντλίες θετικής μετατόπισης

Axial piston machine – swash plate design



Volumetric Losses

B C F G I

Torque Losses

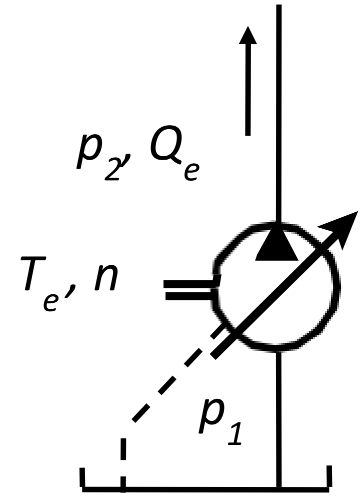
A B C D E

G H J

Βαθμοί απόδοσης

Volumetric efficiency: $$\eta_v = \frac{Q_e}{n \cdot V_i}$$

where V_i …. represents the derived displacement volume



Torque efficiency
(hydraulic-mechanical efficiency): $$\eta_{hm} = \frac{T_i}{T_e} = \frac{\Delta p \cdot V_i}{2 \cdot \pi \cdot T_e}$$

$\Delta p = p_2 - p_1$

Total efficiency: $$\eta_t = \frac{P_{out}}{P_{in}} = \frac{Q_e \cdot \Delta p}{T_e \cdot \omega} = \eta_v \cdot \eta_{hm}$$

The derived displacement volume can only be determined by measurement

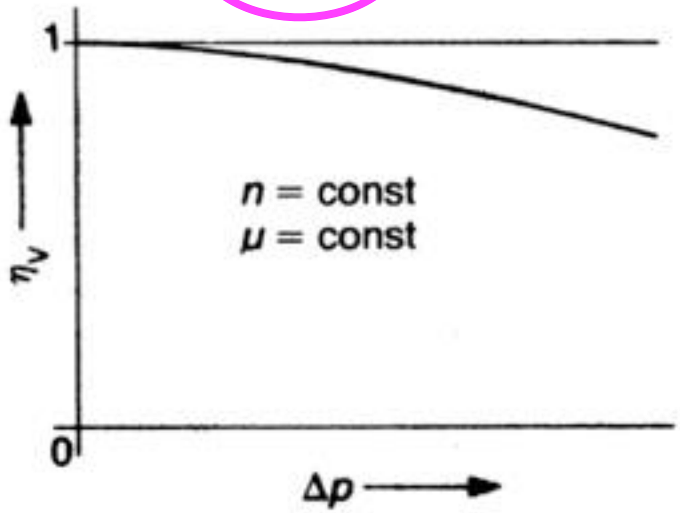
Ογκομετρικός βαθμος απόδοσης

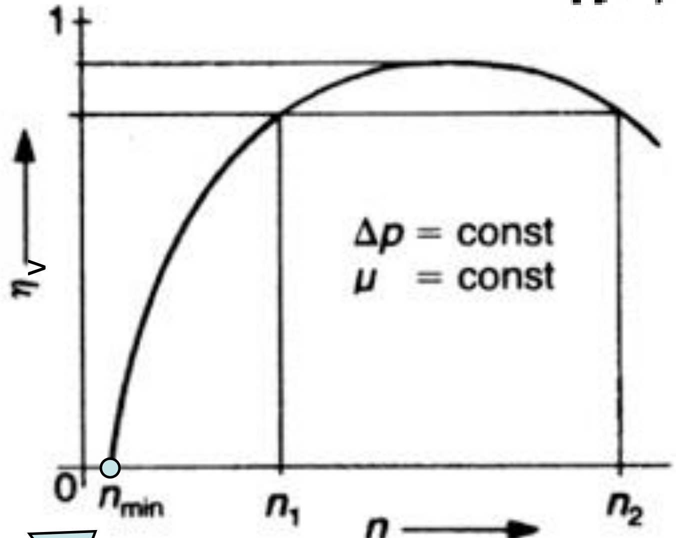
Volumetric efficiency

$$\eta_v = f(\Delta p, n, V_i, \mu)$$

$$\eta_v = \frac{Q_e}{n \cdot V_i}$$

$$Q_e = \alpha \cdot V_{max} \cdot n - Q_s$$





$$\alpha = \frac{V}{V_{\max}}$$

$$V_i \cdot n_{min} = Q_s$$

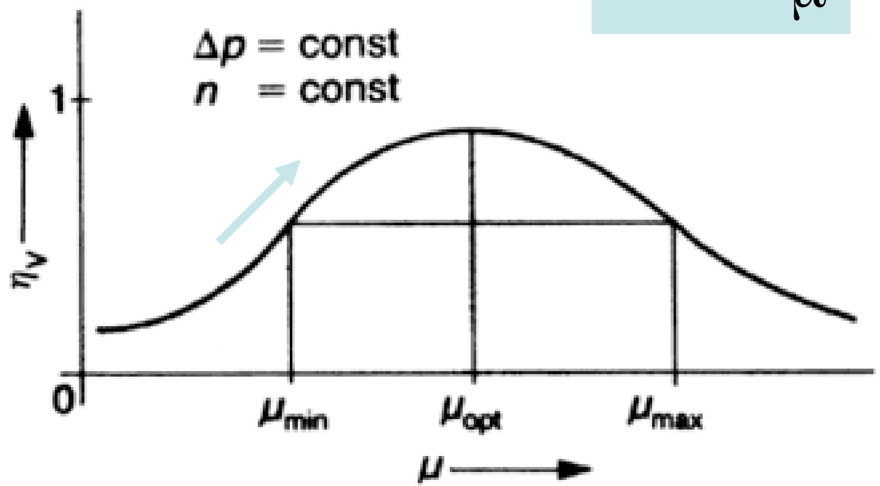
Dynamic viscosity of fluid: $\mu = f(\theta, p)$ [Pa·s]

Ογκομετρικός βαθμος απόδοσης

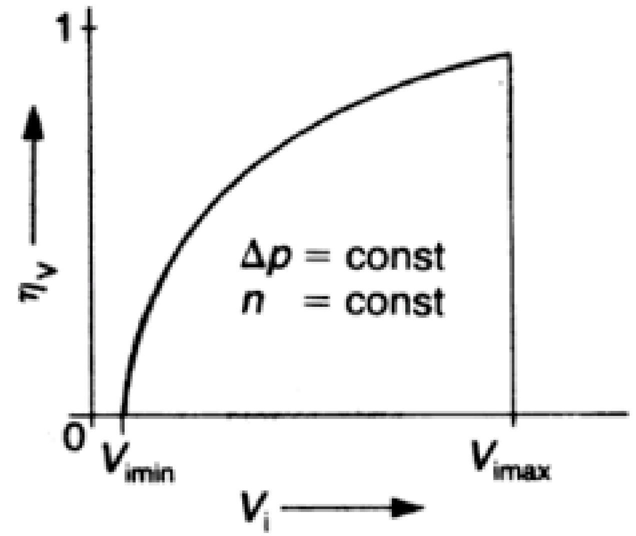
Volumetric efficiency $\eta_v = f(\Delta p, n, V_i, \mu)$ $\eta_v = \frac{Q_e}{n \cdot V_i}$

$Q_e = \alpha \cdot V_{max} \cdot n - Q_s$

$Q_{SL\mu} \approx \frac{1}{\mu}$



Variable displacement pump



Typical values of dynamic viscosity used in displacement machines: $0.0435\ Pa \cdot s \div 0.0087\ Pa \cdot s$

with: Kinematic viscosity ν [cSt, mm^2/s] and $\mu = \nu \cdot \rho$

$\nu = 10 mm^2 \cdot s^{-1} \div 50\ mm^2 \cdot s^{-1}$ with $\rho = 870 kg \cdot m^{-3}$

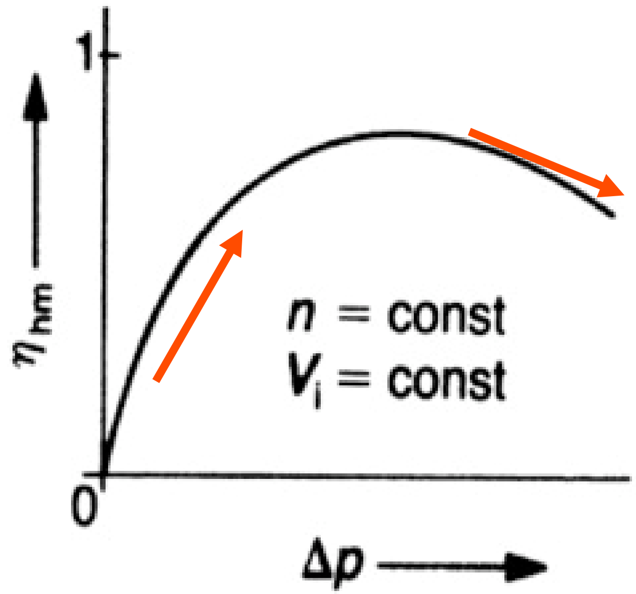
Απόδοση ροπής

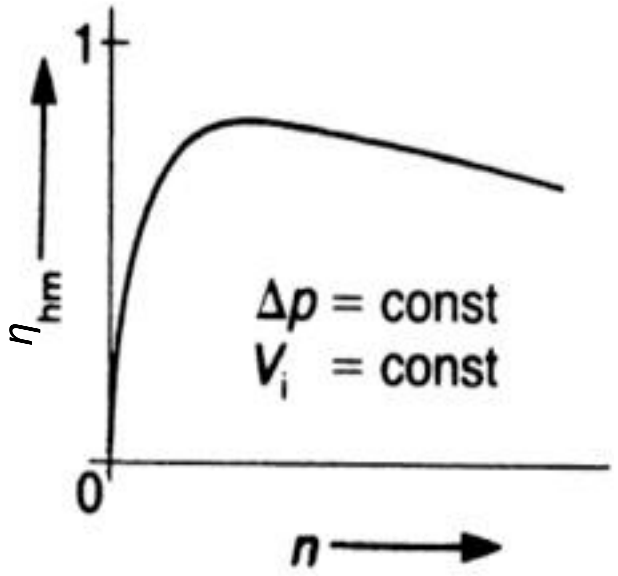
$$\eta_{hm} = f(\Delta p, n, V_i, \mu)$$

$$\eta_{hm} = \frac{T_i}{T_e} = \frac{\Delta p \cdot V_i}{2 \cdot \pi \cdot T_e} = 1 - \frac{T_S}{T_e}$$

$$T_e = \frac{\Delta p \cdot \alpha \cdot V_{\max}}{2 \cdot \pi} + T_S$$

$$T_s = T_{S\mu} + T_{S\rho} + T_{Sp} + T_{Sc} = C_\mu \cdot \mu \cdot n + C_\rho \cdot \rho \cdot n^2 + C_p \cdot \Delta p + T_{Sc}$$

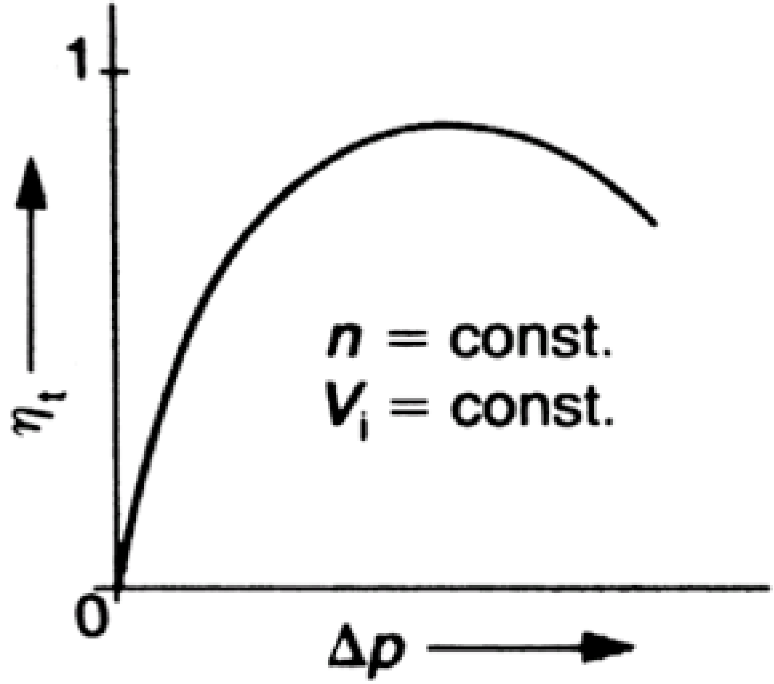


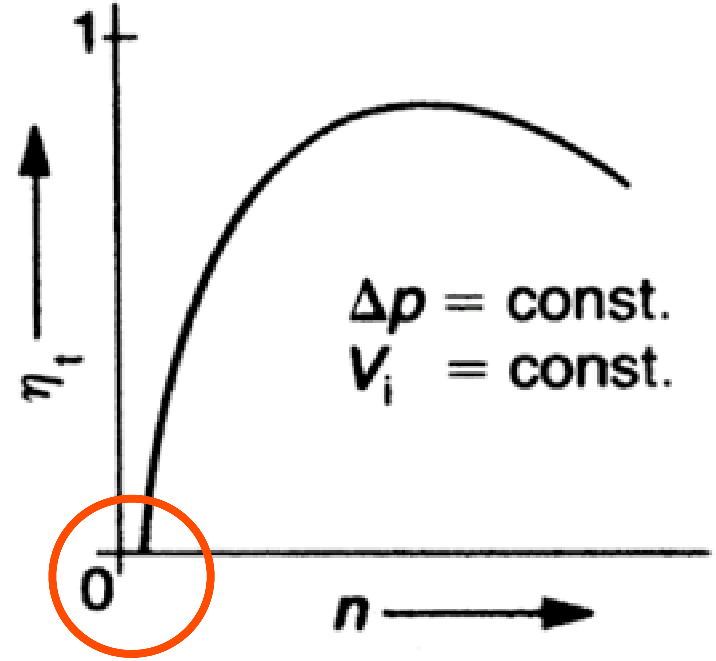


Ολικός βαθμός απόδοσης

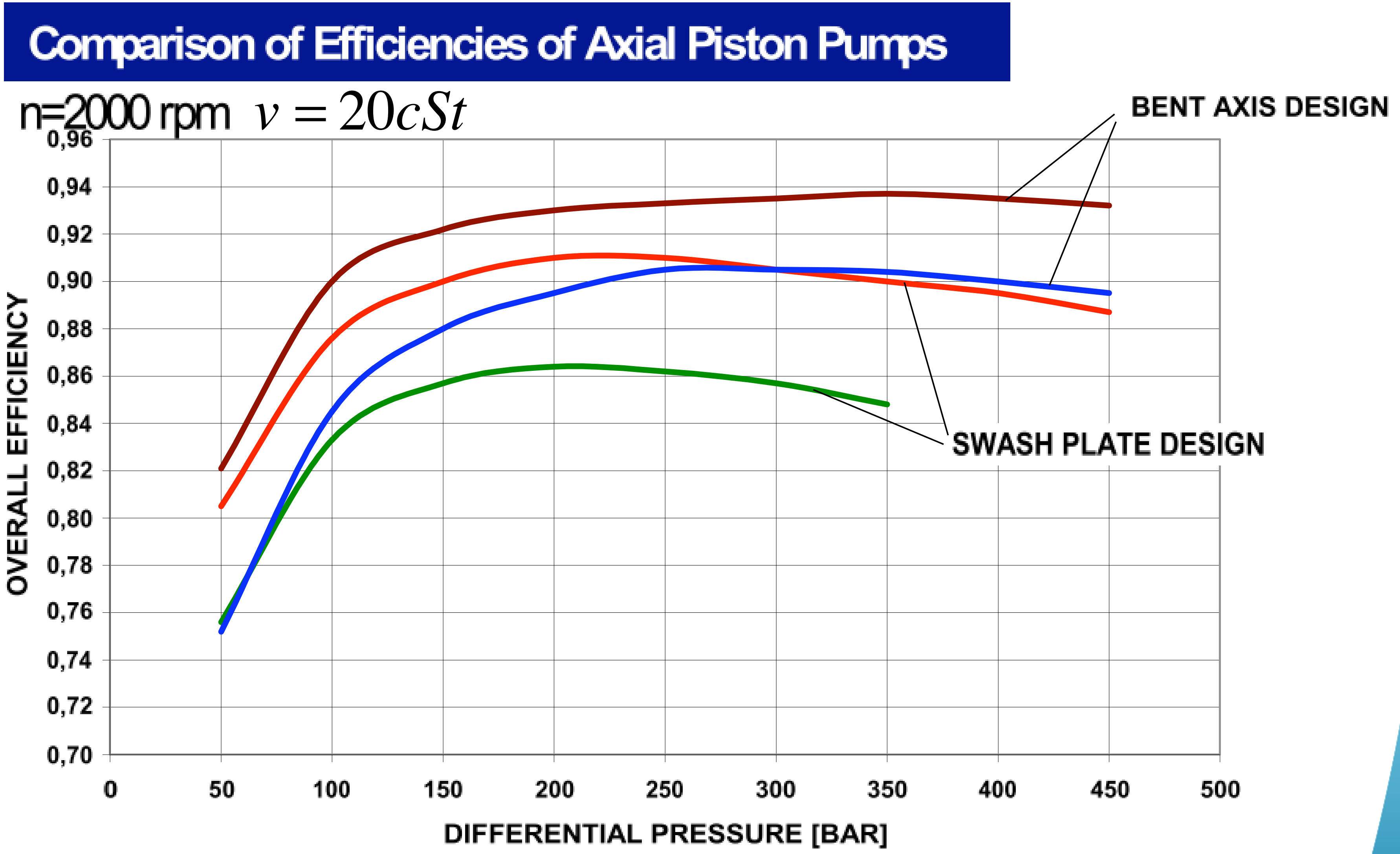
$$\eta_t = f(\Delta p, n, V_i, \mu)$$

$$\eta_t = \frac{P_{out}}{P_{in}} = \frac{Q_e \cdot \Delta p}{T_e \cdot \omega} = \eta_v \cdot \eta_{hm}$$





Χαρακτηριστικά steady state

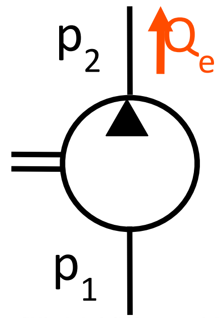


Παράδειγμα

The displacement pump with a displacement volume of $V=100\ cm^3/rev$ is driven by an electric motor at 1800 rpm. At steady state conditions the pressure difference across the pump is $\Delta p= 380$ bar. Determine the effective flow rate at the pump outlet Q_e and the power required to drive this pump. Assume the following values for efficiency:

- volumetric efficiency $\eta_v=0.87$
- torque efficiency $\eta_{hm}=0.95$

$$\Delta p = p_2 - p_1$$



The effective volume flow rate:

$$Q_e = V \cdot n \cdot \eta_v = 100 \cdot 10^{-6}\,\mathrm{m}^3 \cdot 1800 \cdot \frac{1}{60} \cdot s^{-1} \cdot 0.87 = 0.00261\,\mathrm{m}^3 \cdot s^{-1} = 156.6\ \mathrm{l} \cdot \mathrm{min}^{-1}$$

The effective torque yields:

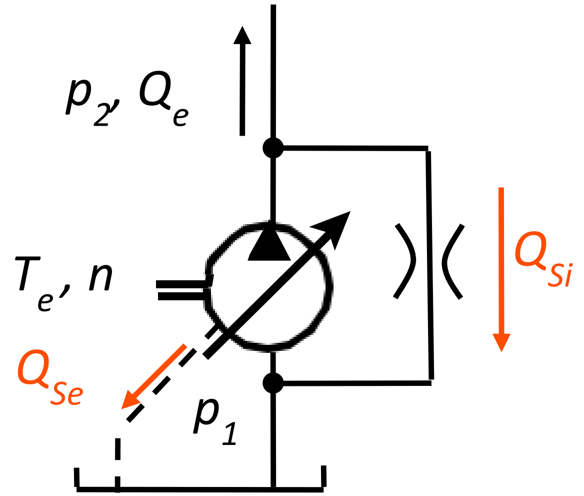
$$T_e = \frac{T_i}{\eta_{hm}} = \frac{\Delta p \cdot V_i}{2 \cdot \pi \cdot \eta_{hm}} = \frac{380 \cdot 10^5\ \mathrm{Pa} \cdot 100 \cdot 10^{-6}\ \mathrm{m}^3}{2 \cdot \pi \cdot 0.95} = 636.62\ \mathrm{Nm}$$

The power required to drive the pump:

$$P = T_e \cdot \omega = T_e \cdot 2 \cdot \pi \cdot n = 636.62\ \mathrm{Nm} \cdot 2 \cdot \pi \cdot \frac{1800}{60} \cdot s^{-1} = 120\ \mathrm{kW}$$

Μετρήσεις steady state

The aim of steady state measurements is determination of steady state characteristics of pumps.



- Losses and their dependency on operating parameters
- Efficiency and its dependency on operating parameters
- Effective torque T_e and effective volumetric Flow rate Q_e in the whole parameter range

Under stable conditions, all parameters remain constant, including temperatures !

Parameters to be measured:

Inlet pressure p_1 Outlet pressure p_2

Torque T_e Shaft speed n

Volume flow rate at pump outlet $Q_e=Q_2$

Temperatures ϑ_1, ϑ_2 , ϑ_{Se}

Συμπεριφορά steady state

V_i derived displacement volume to be determined during measurements

$$Q_e = \alpha \cdot V_{max} \cdot n - Q_s$$

Q_S must be calculated using measurement results

Measured value

$$\Delta p = p_2 - p_1$$

$$T_e = \frac{\Delta p \cdot \alpha \cdot V_{max}}{2 \cdot \pi} + T_S$$

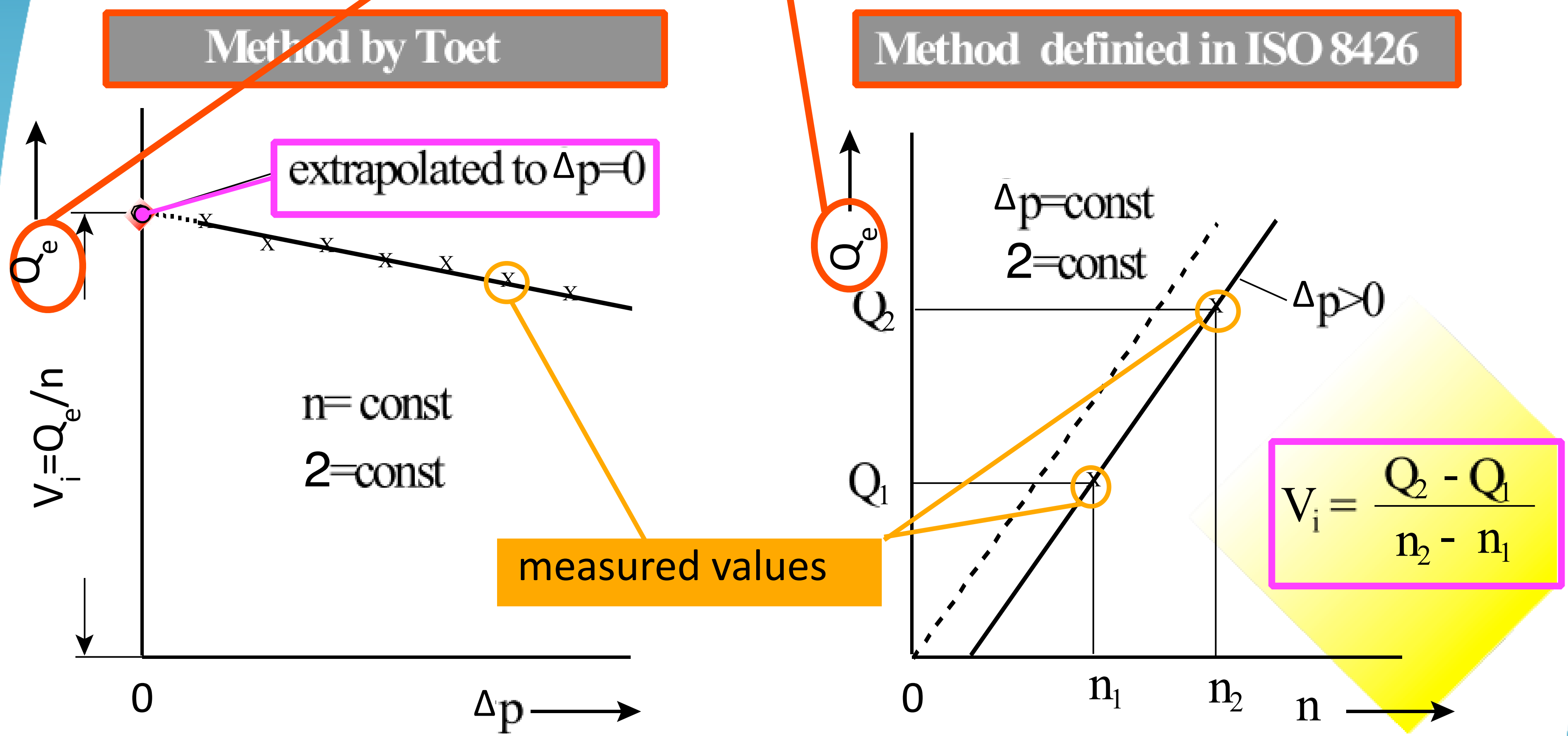
T_S must be calculated using measurement results

$$P_{in} = P_{out} + P_S = \Delta p \cdot Q_e + P_s = T_e \cdot 2 \cdot \pi \cdot n$$

P_S must be calculated using measurement results

Εξαγωγή παροχής όγκου

Measurement of effective volume flow rate at pump outlet
under defined conditions



V_i required for determination of losses and volumetric and torque efficiency

Κύκλωμα δοκιμής

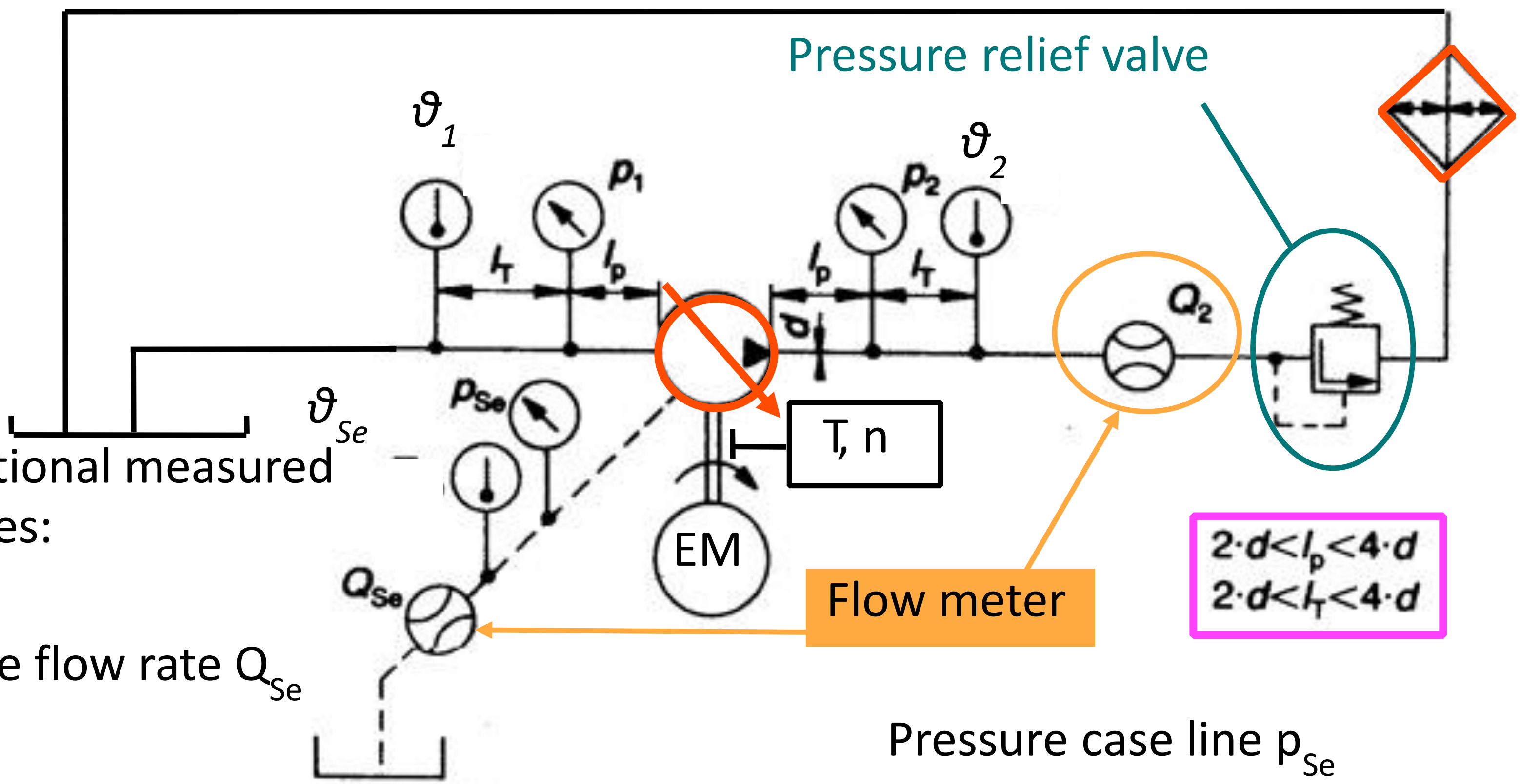
Measured Values: Inlet pressure p_1 Torque T Shaft speed n

Outlet pressure p_2 Temperature $\vartheta_1, \vartheta_2, \vartheta_{Se}$

Volume flow rate at pump outlet Q_2

ISO 4409

⇨ Temperature θ_1 must remain constant during measurements



Additional measured values:

Case flow rate Q_{Se}

Pressure case line p_{Se}

Μετρήσεις steady state

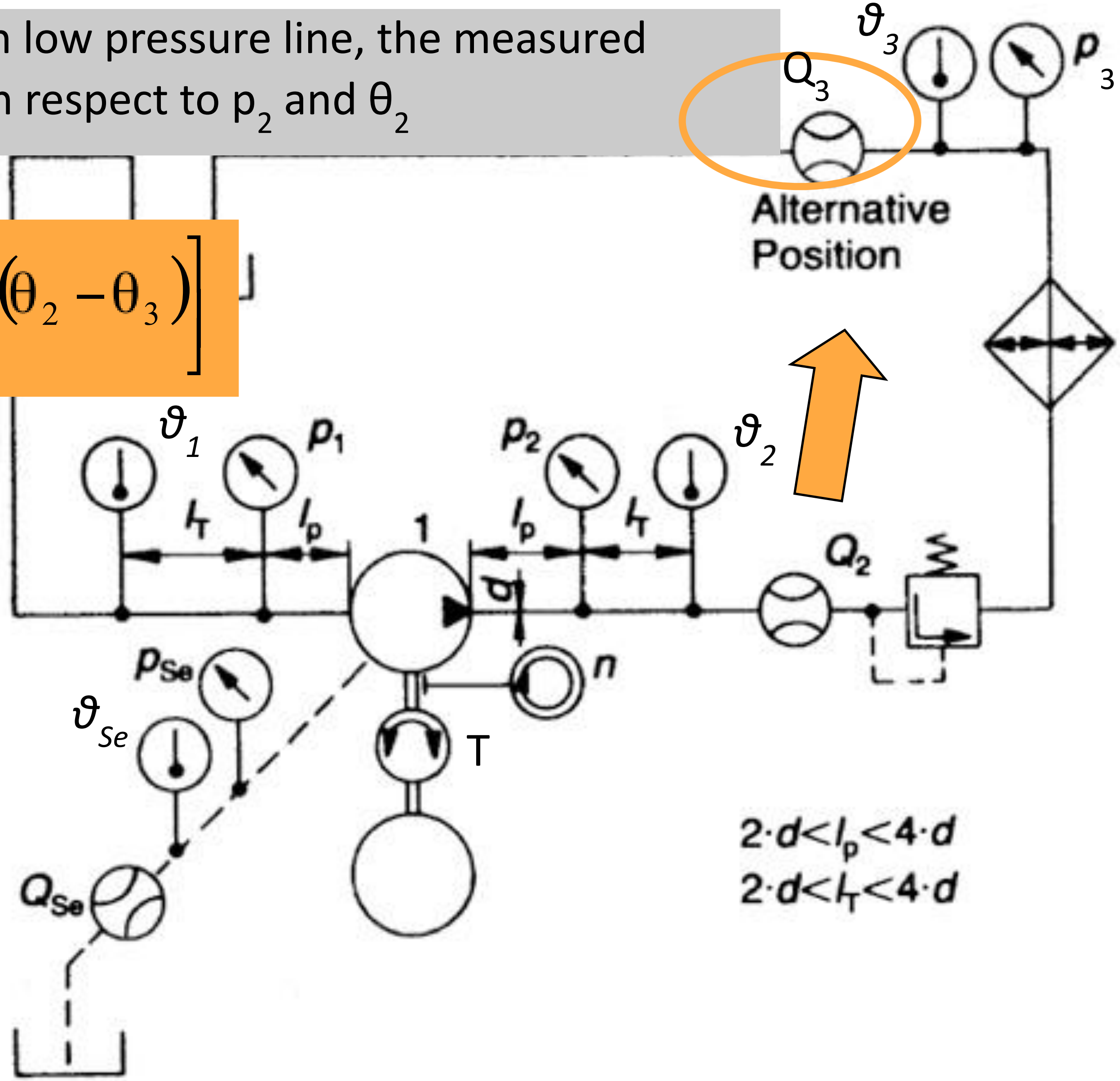
In case that Q_2 is measured in low pressure line, the measured value must be corrected with respect to p_2 and θ_2

$$Q_2 = Q_3\left[1 - \frac{p_2 - p_3}{K} + \beta_\theta\left(\theta_2 - \theta_3\right)\right]$$

with

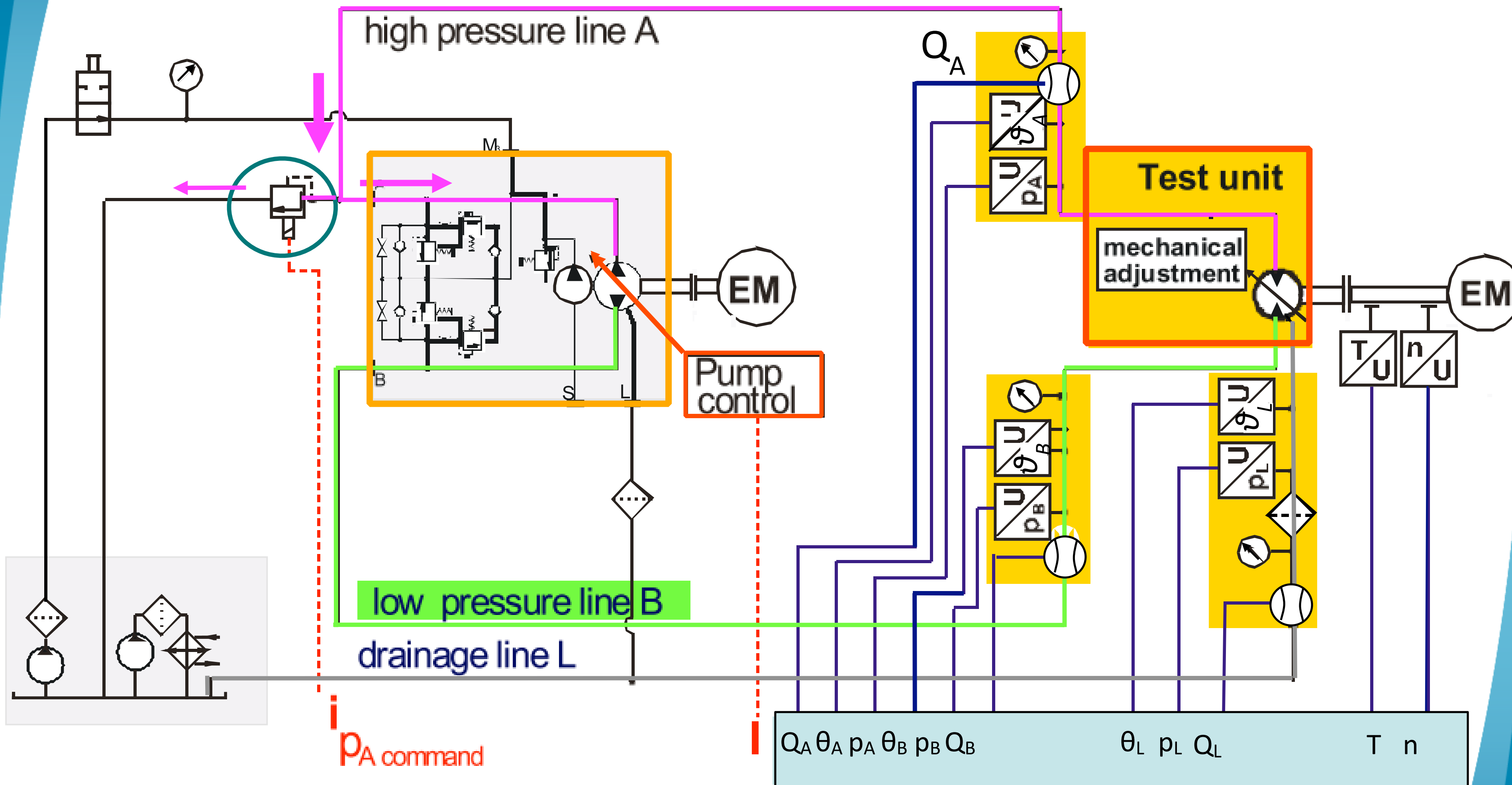
K ... bulk modulus
$ß_\theta$ ...thermal volumetric expansion coefficient

$K = 2\cdot10^9$ Pa
$ß_\theta = 0.65\cdot10^{-3}\ K^{-1}$



Μετρήσεις steady state

Measurement in pumping and motoring mode



⇨ Less temperature problems, because only a small amount of volume flow is throttled in pressure relief valve

Μετρήσεις steady state

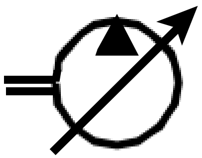
ISO measurement accuracy classes

Table 1: Permissible systematic errors of measuring instruments as determined during calibration

Parameter of measuring instrument	Permissible systematic errors for classes of measurement accuracy		
	A	B	C
Rotational frequency [%]	± 0.5	± 1.0	± 2.0
Torque [%]	± 0.5	± 1.0	± 2.0
Volume flow rate [%]	± 0.5	± 1.5	± 2.5
Pressure below 2 bar gauge [bar]	± 0.01	± 0.03	± 0.05
Pressure greater than or equal to 2 bar gauge [%]	± 0.5	± 1.5	± 2.5
Temperature [°C]	± 0.5	± 1.0	± 2.0

Permissible temperature variation

Accuracy class	A	B	C
Temperature variation [K]	±1.0	±2.0	±4.0



Μετρήσεις πίεσης

The different types of pressure differ only with respect to their reference point.

Types of pressure

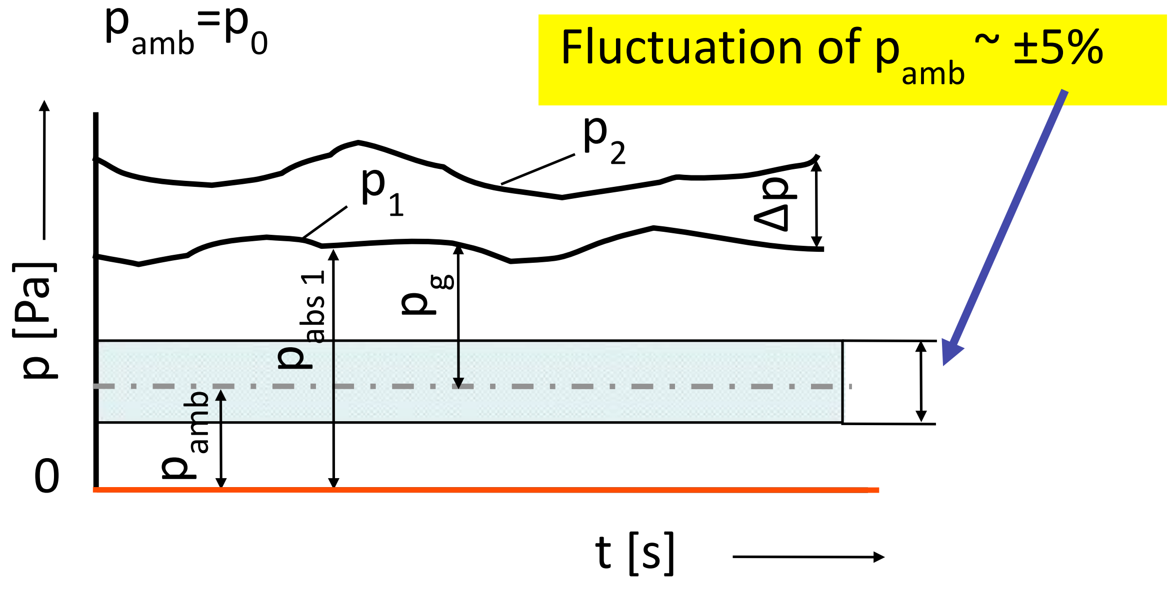
Absolute pressure p_{abs}

Atmospheric pressure p_{amb}

Differential pressure Δp

Gauge pressure p_g

$p_g = p_{measured} - p_{amb}$



Direct measuring pressure instruments

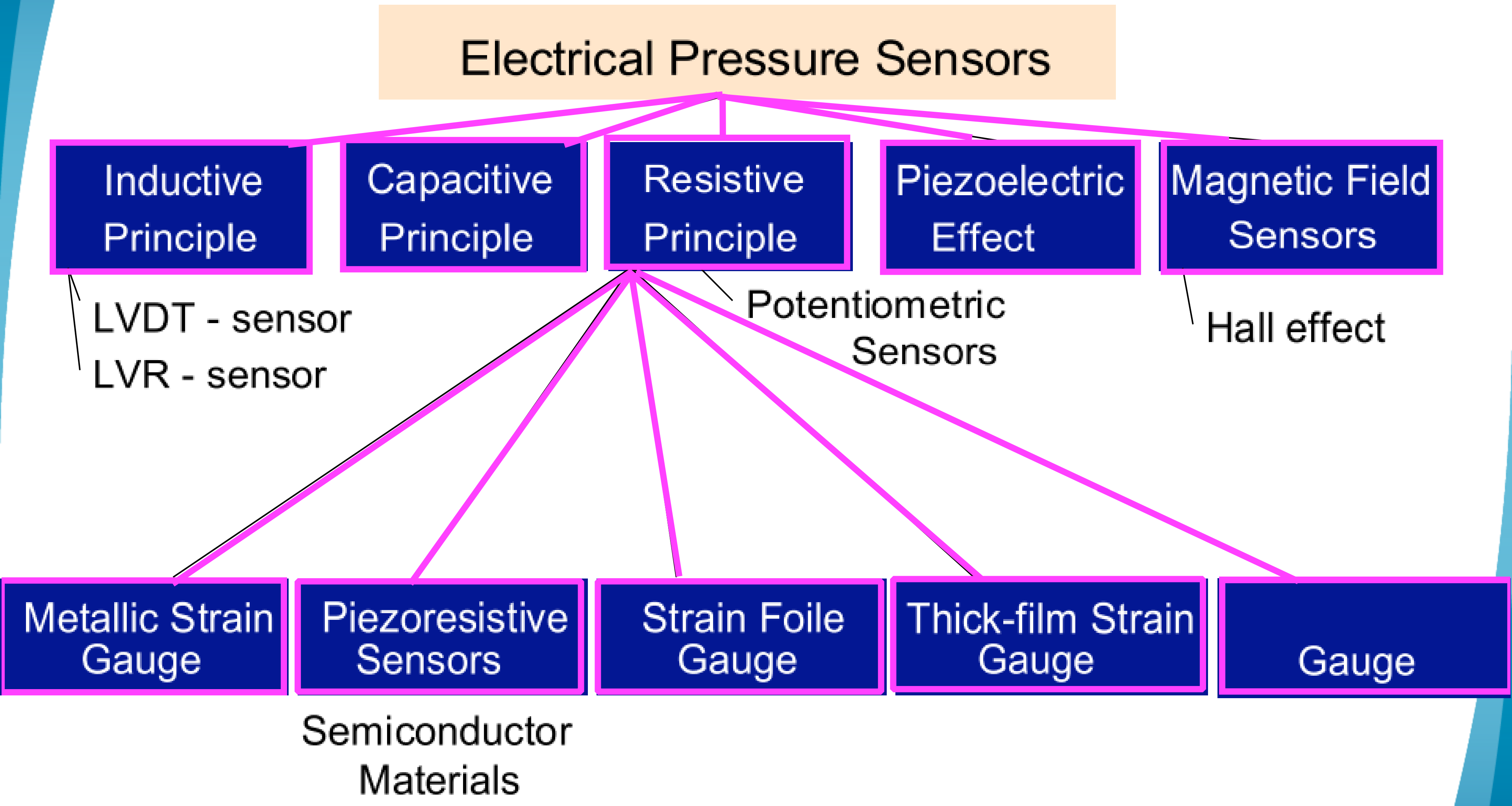
Using a liquid column

Indirect measuring pressure instruments

Using the effect of a pressure acting on a material or on bodies of a certain shape

Electrical pressure sensors

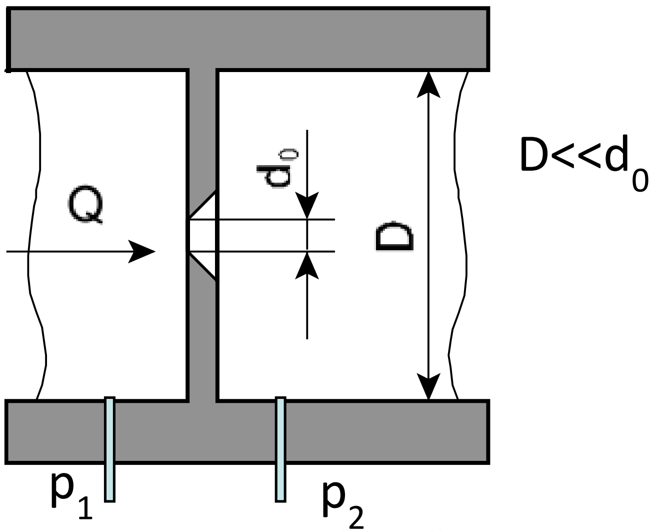
Μετρητές πίεσης



Μετρητές παροχής

Flow measuring instrument is defined as device which measures the flow rate of a fluid.

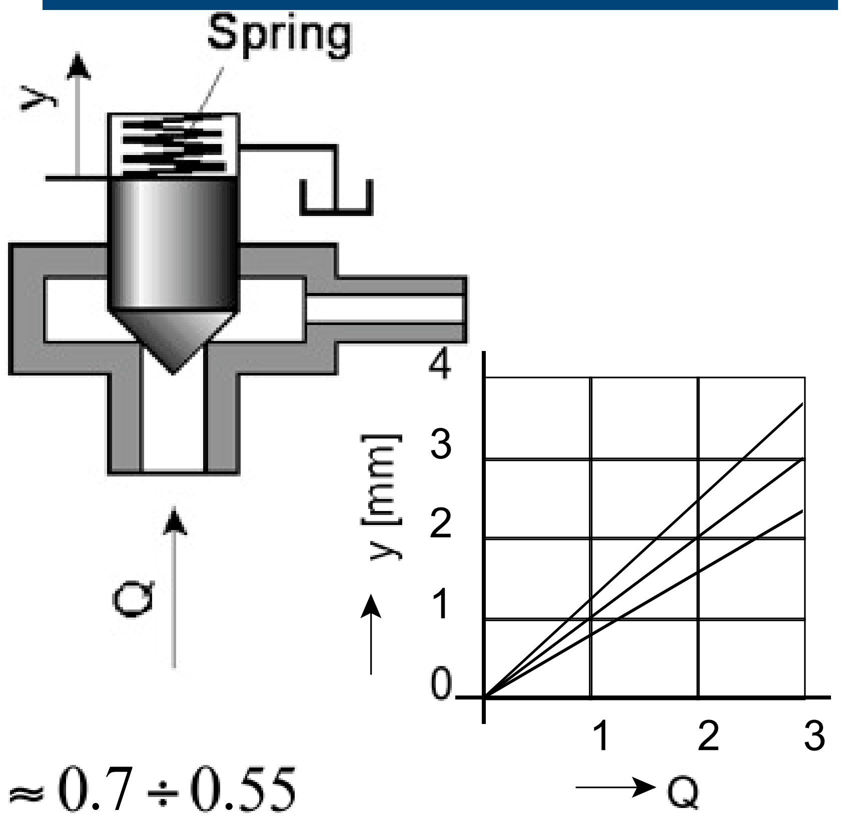
Pressure difference across an orifice



$$Q = \alpha_D \cdot \frac{\pi \cdot d^2}{4} \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}}$$

α_D...flow discharge coefficient

Displacement of a spring loaded floated element



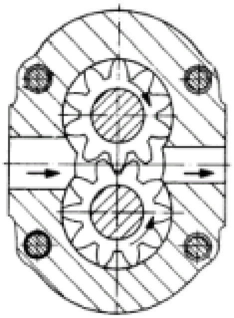
$$\alpha_D \approx 0.7 \div 0.55$$

Should not be used for determination of steady state characteristics!

Μετρητές παροχής

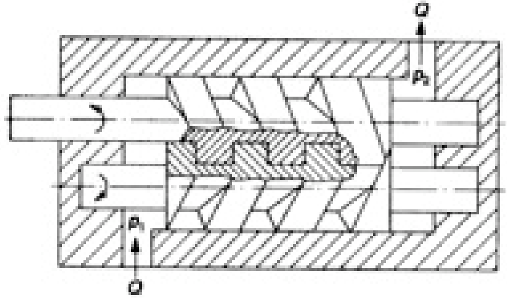
Flow meter – device which directly indicates the rate of flow of a fluid

Displacement principle



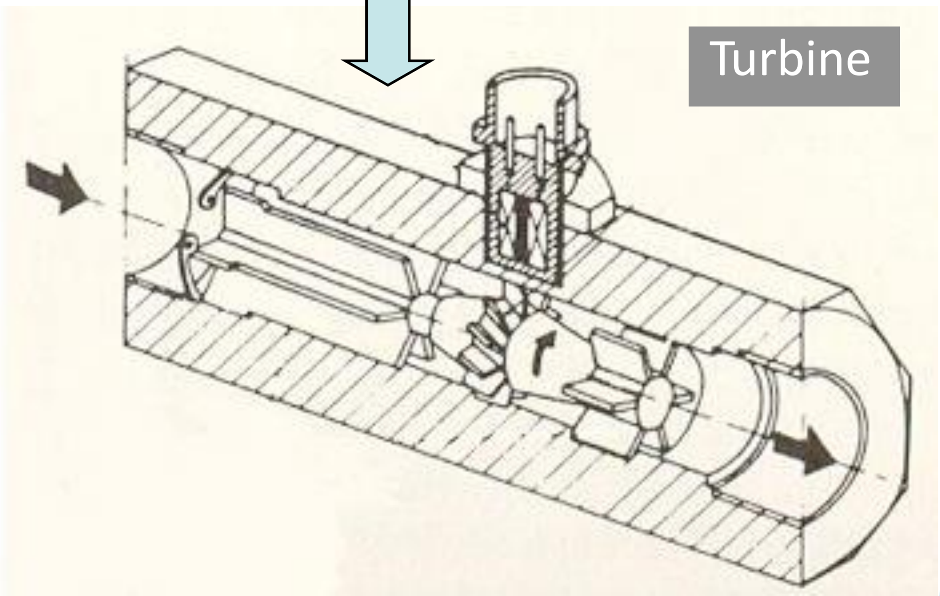
Gear flow meter

Screw flow meter



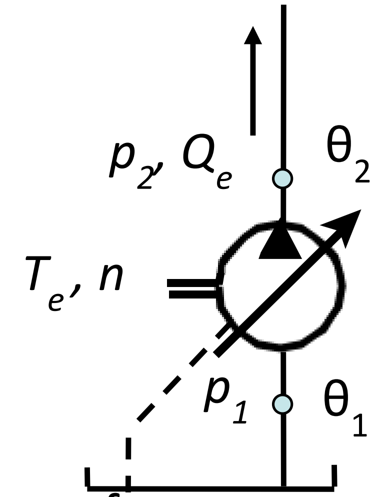
Hydrodynamic principle

Using kinetic energy of a fluid to drive a rotating system of blades (an impeller), whereas the rotational speed of the rotor is measured with an electric speed sensor (frequency measurement device)



Turbine

Διαδικάσία μέτρησης



1. Keep p_1 and θ_1 constant
2. Adjust different pressure levels at constant shaft speed
3. Record all measured values under steady state conditions

⇨ Repeat measurements for different speed settings

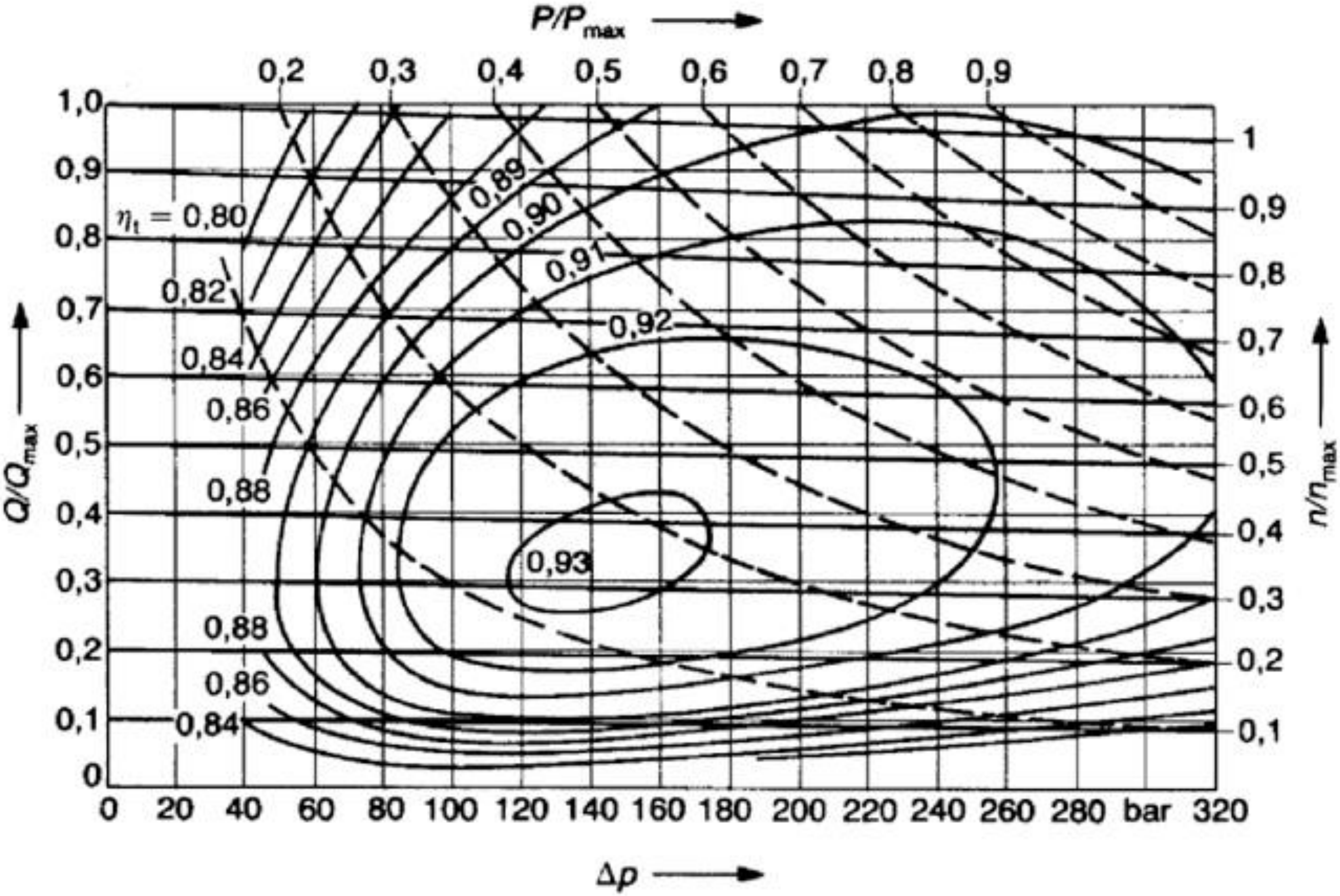
⇨ In case of variable displacement pump repeat measurements for 75%, 50% and 25% of V_{max}

⇨ Repeat measurements for different temperatures θ_1

Provide table with measurement results

No.	n [RPM]	T [Nm]	p_1 [bar]	θ_1 [°C]	p_2 [bar]	Q_2 [l/min]	θ_2 [°C]	p_{Se} [bar]	Q_{Se} [l/min]	θ_{Se} [°C]
1	2001.46	15.03	19.77	50.50	25.11	145.13	61.20	1.20	0.21	65.30
2	2001.63	68.78	19.79	50.30	69.65	144.02	60.30	1.25	0.43	66.00
3	2001.79	130.59	19.86	50.30	120.82	142.92	60.40	1.28	0.56	65.20

Χαρκτηριστικά steady state



Ογκομετρικές απώλειες

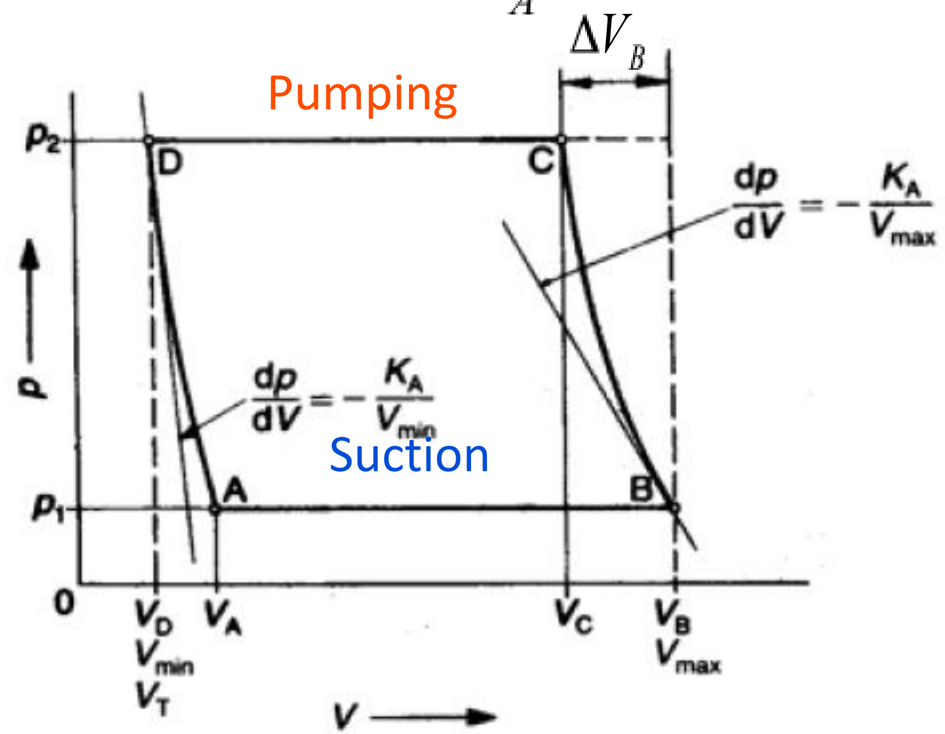
Effective volume flow rate is reduced due to compressibility of the fluid

$$\int_B^C \frac{dV}{V} = \int_B^C -\frac{1}{K_A}\,dp \quad \Rightarrow \quad \ln V_C - \ln V_B = -\frac{1}{K_A}\left(p_C - p_B\right)$$

$$\Delta V_B = V_B\left(1 - e^{-\frac{1}{K_A}(p_C - p_B)}\right)$$

simplified

$$\Delta V_B = V_B\,\frac{\Delta p}{K_A}$$



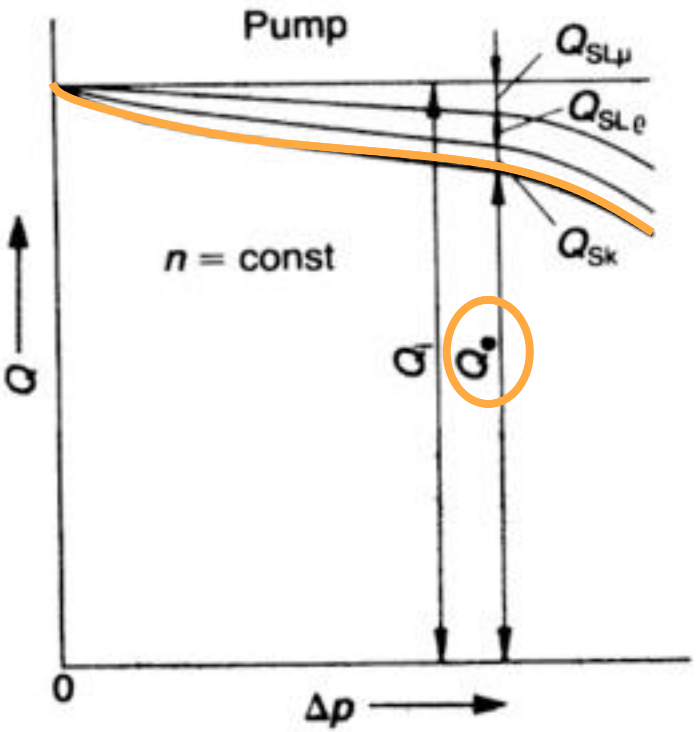
$$Q_{SK} = n\,\Delta V_B$$

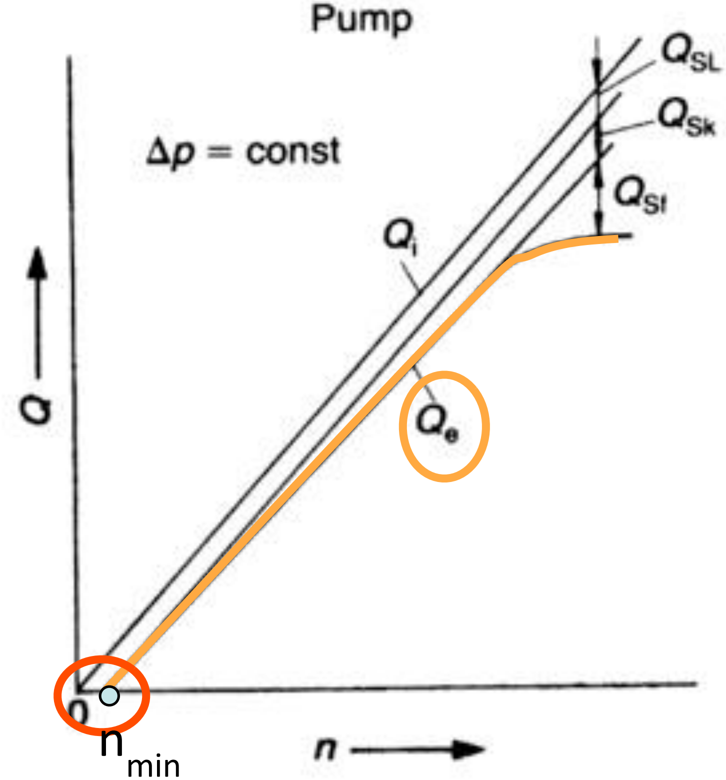
with n … pump speed

Συμπεριφορά steady state

of a real displacement machine $Q_i = V \; n = \alpha \; V_{max} \; n$

Effective volumetric flow rate $Q_e = Q_i - Q_S$





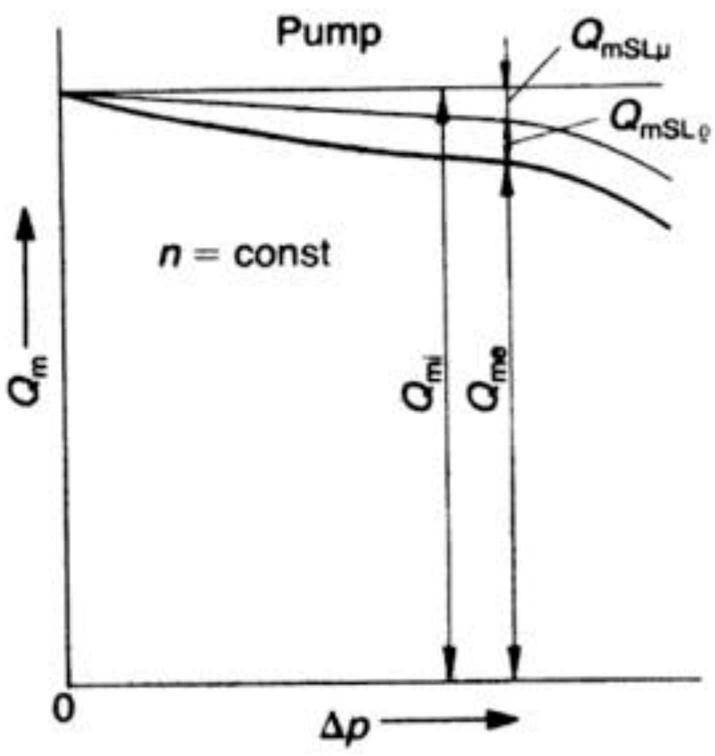
$$Q_S = f\left(\Delta p, n, V, \theta\right)$$

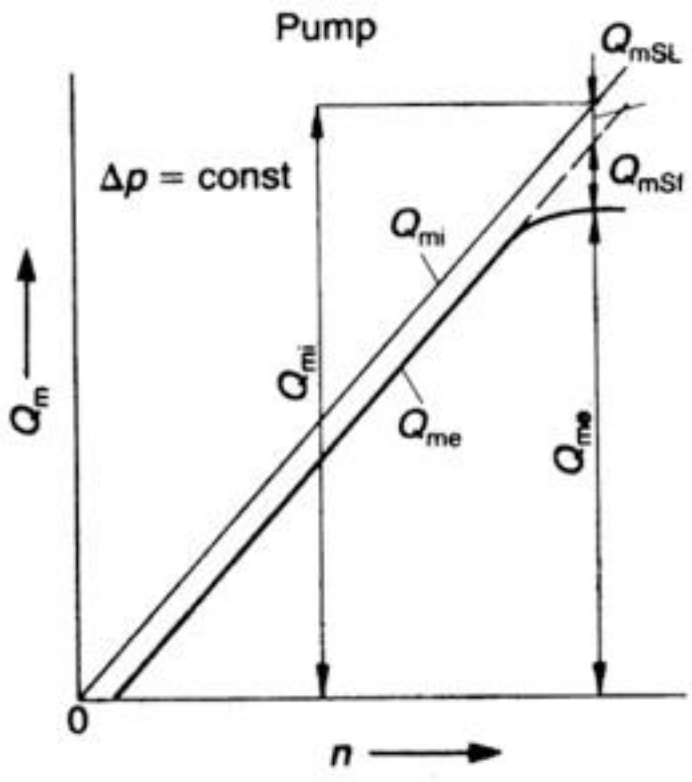
θ ...temperature

Συμπεριφορά steady state

Effective mass flow at pump outlet Q_{me}

Loss component due to compressibility does not occur!





Ευχαριστώ για την προσοχή σας

Email:
etsolakis@central.ntua.gr