MEC-E5003

FLUID POWER BASICS

Study Year 2018 - 2019

Pumps
Actuators
Accumulators

Lecture themes

Flow to the system – How?

Making use of the hydraulic power – How?

Storing energy in hydraulic system – Why and is that even possible?

Hydrostatic pumps
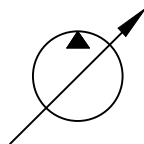
Convert mechanical power into hydraulic power

Hydrostatic pumps produce <u>flow</u>, not pressure
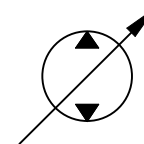
Unidirectional   Bidirectional

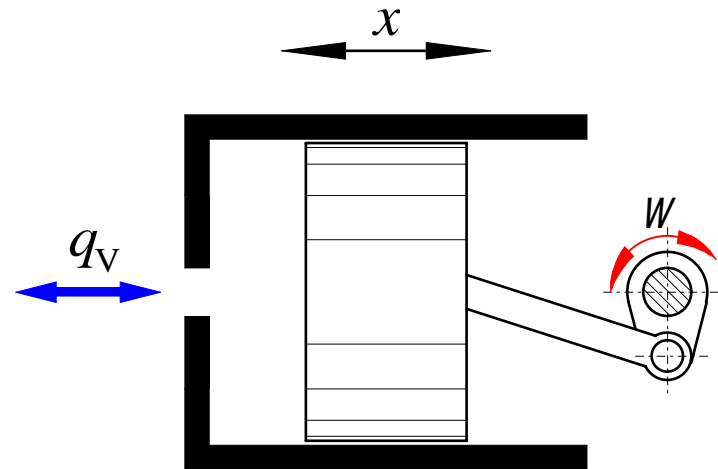Constant displacement

Variable displacement

Construction

Most common construction types:

- gear
- vane
- screw
- piston

All operate on positive displacement principle
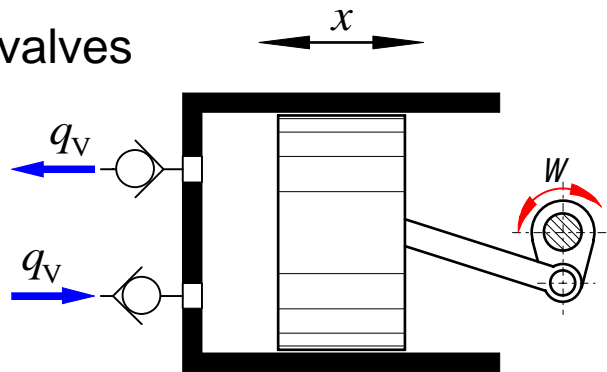
Positive displacement principle

x

q_V

W

Operating phases:
Fluid flows into transfer volume   – suction phase
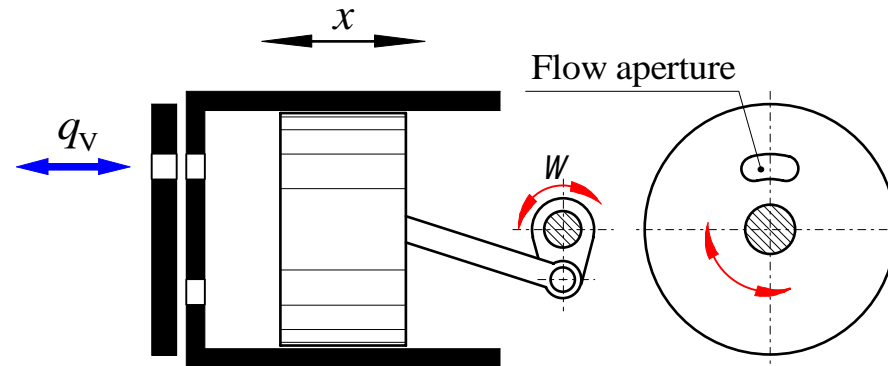Fluid flows out from volume   – pressure phase

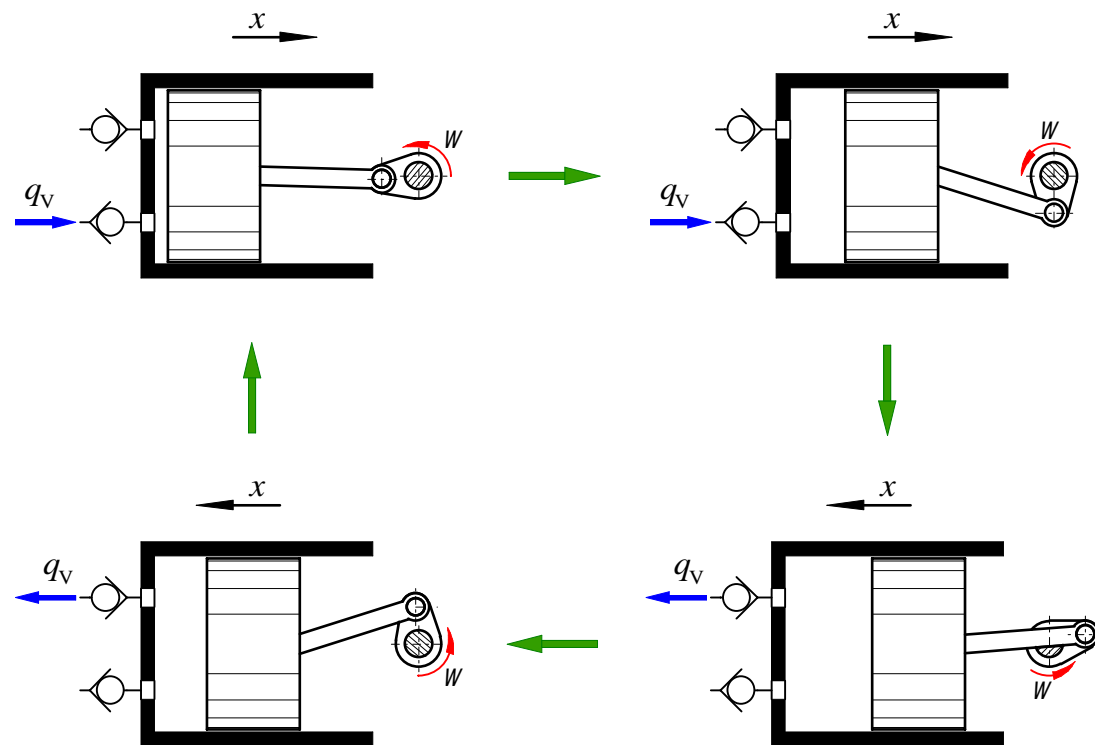# Control of flow direction

Check valves

x

q_V

q_V

W

¬ Pressure control
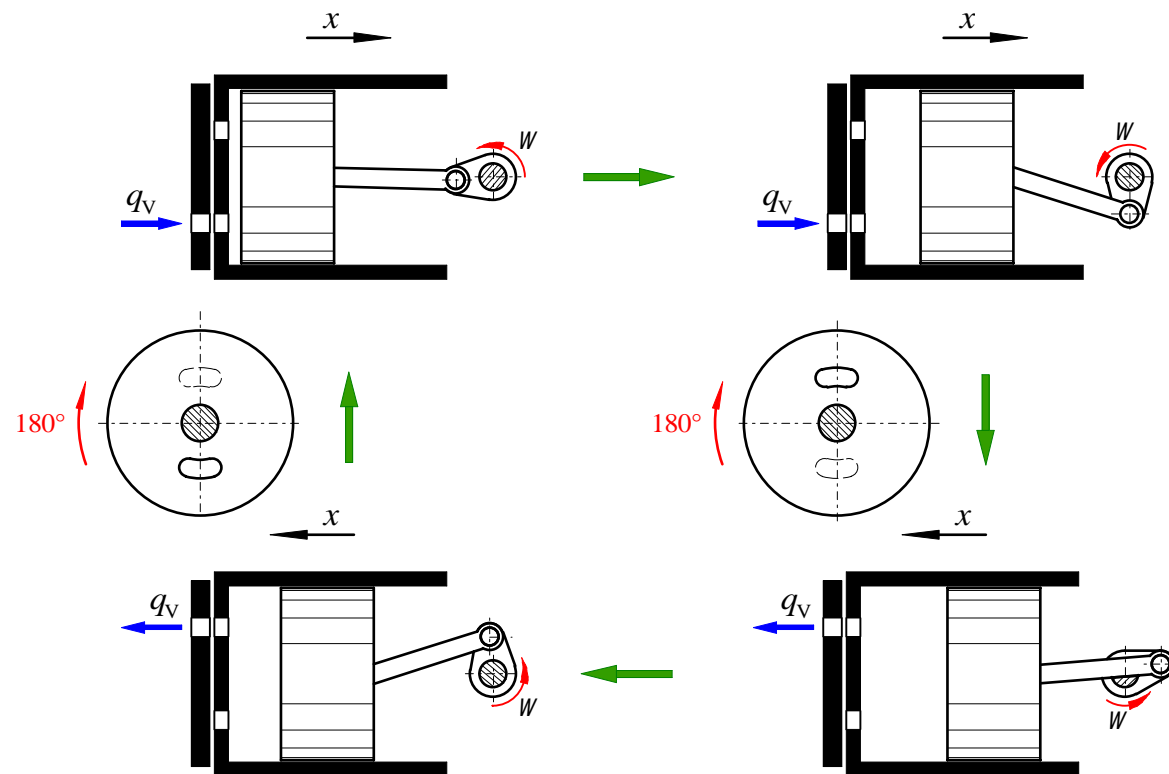aka valve control

x

Flow aperture

q_V

W

Forced control ®

Pressure control aka valve control
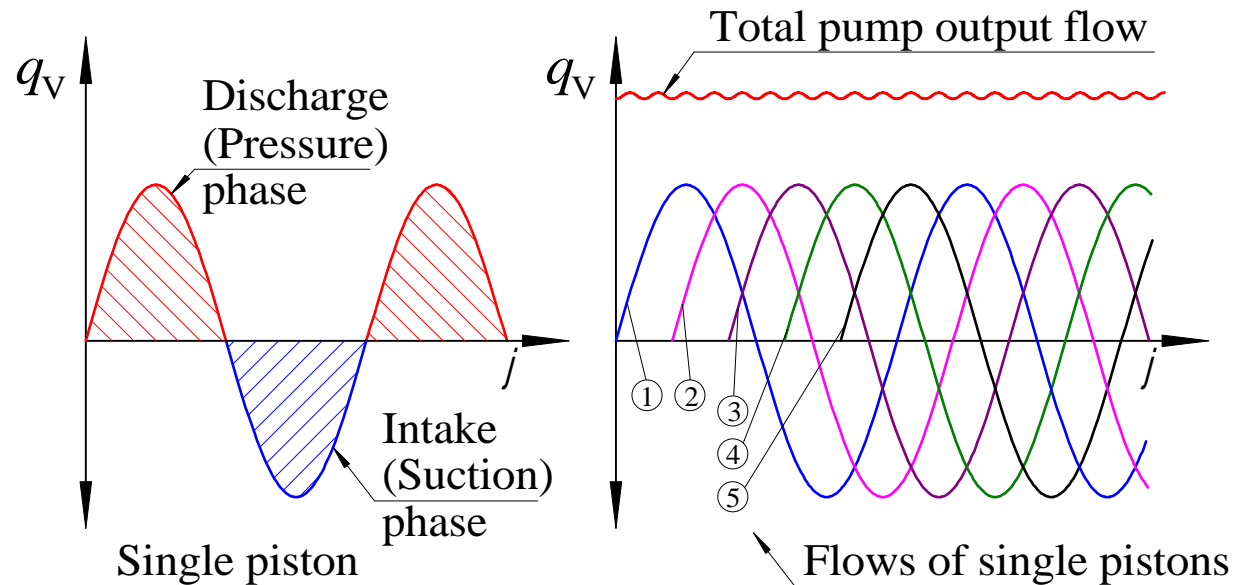
Forced control

Output flow variation ->
Internal pressure variation (depending
on the system impedance) ->

Flow pulsation External (air) pressure variation -> Noise

Total pump output flow

q_V

Discharge
(Pressure)
phase

Intake
(Suction)
phase

Single piston

q_V

① ② ③ ④ ⑤

Flows of single pistons

Flow pulsation is due to intermittent
nature of positive displacement principle

Cavitation in pumps

Cause: Friction losses in inlet channel of pump

Pressure in fluid decreases to vapour pressure of the fluid

® fluid starts to vaporize (also size of air bubbles increases)

® vaporized fluid is pressurized in pump

® vapour bubbles collapse rapidly ("implosion")

® pressure shocks

® material damages, noise, decreased output flow

Suction lines of pumps are
- short
- straight
- wide (large diameter) and the pump can be placed lower than the reservoir (tank) surface to avoid cavitation.

Pump characteristics

Theoretical output flow $\qquad q_{V,theor} = n \times V_g$

Swept volume V_g [m³/r] $\qquad\qquad\qquad$ cm³/r = 10⁻⁶ m³/r

Rotation speed n [r/s] $\qquad\qquad\qquad$ r/min = 1/60 r/s

$q_{V,theor} = w \times V_{rad}$

$w = 2p \cdot n$ $\qquad\qquad$ Angular velocity ω [rad/s]

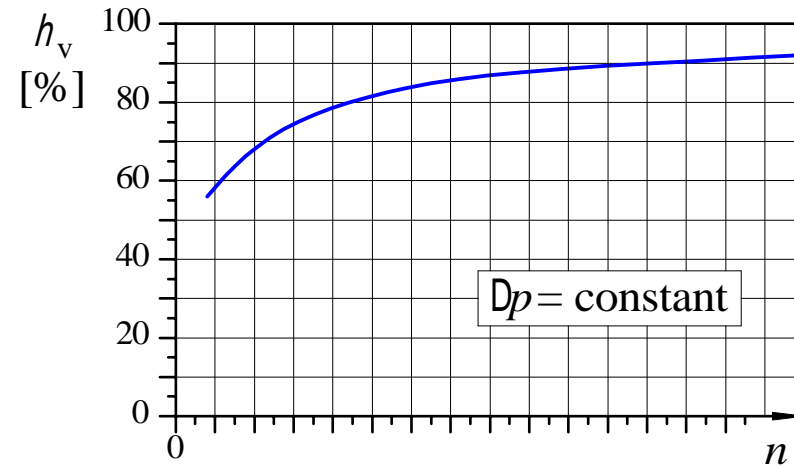$V_{rad} = \dfrac{V_g}{2p}$ $\qquad\qquad$ Swept volume per radian V_{rad} [m³/rad]
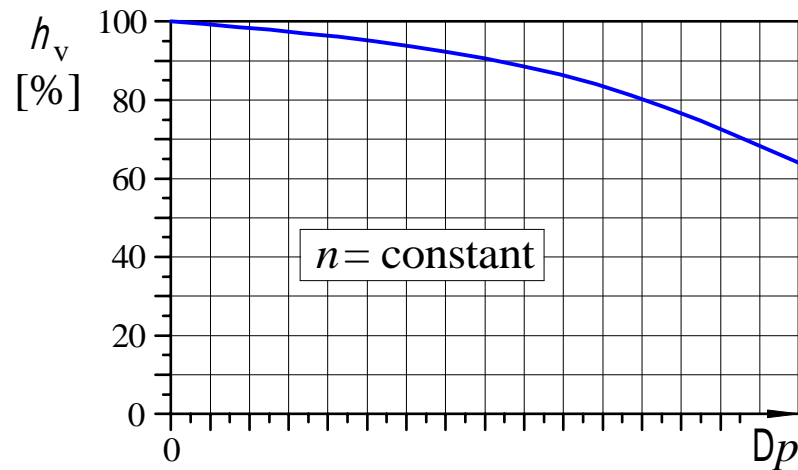
Effective output flow

$$q_{V,real} = n \times V_g \times h_v$$

Leakage – volumetric efficiency h_v

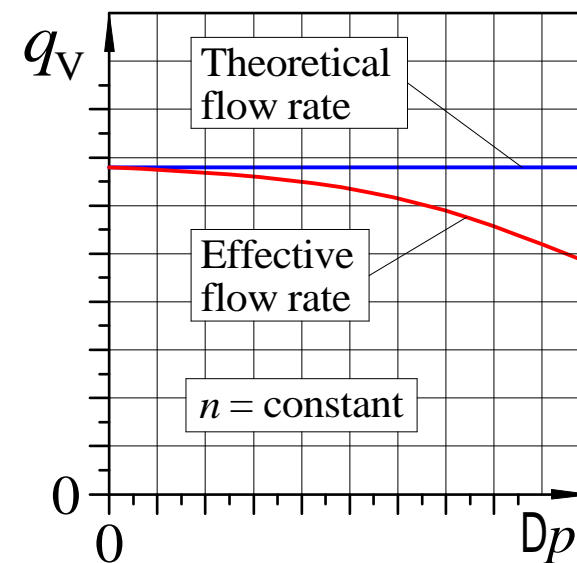$$q_{V2} = \varepsilon V_i n - C_s \frac{V_i \Delta p}{2\pi\nu\rho}$$

h_v [%]

$n = $ constant

Δp

h_v [%]

$\Delta p = $ constant

n

$$q_{\mathrm{V2}} = \varepsilon V_{\mathrm{i}} n - C_{\mathrm{s}} \frac{V_{\mathrm{i}} \Delta p}{2\pi\nu\rho}$$

Theoretical output flow – Effective output flow
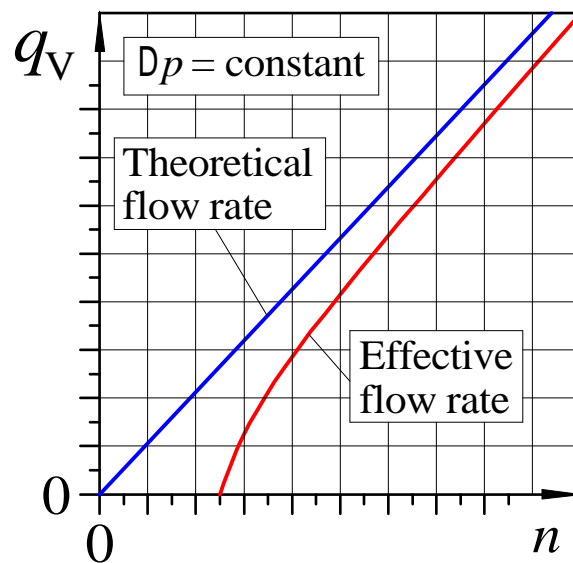
ε	pump angle set value (0 - 1)
V_{i}	max. displacement (per revolution)
n	rotational speed (1/s)
C_{s}	laminar flow loss coefficient
Dp	pressure difference over pump
ν	fluid kinematic viscosity
ρ	fluid density

q_{V}

Dp = constant

Theoretical flow rate

Effective flow rate

0

0   n

q_{V}

Theoretical flow rate

Effective flow rate

n = constant

0

0   Dp

Leakage flows in pumps
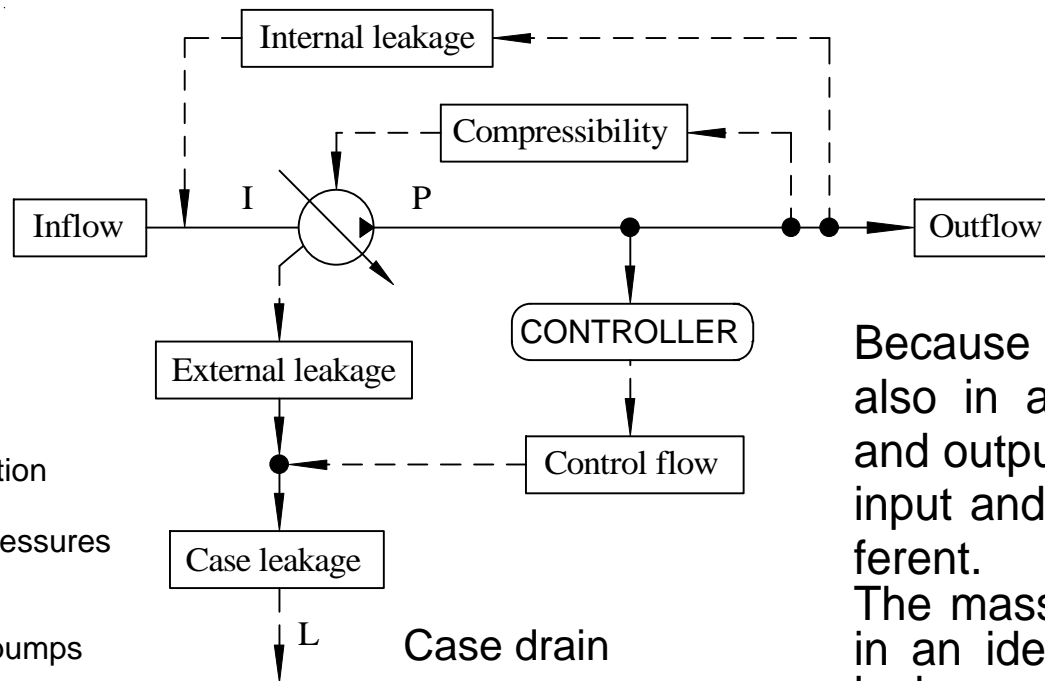
Internal leakage
- All pumps

External leakage
- With case drain connection
- To protect shaft seal
- To prevent high case pressures

Controller drain
- If pump has a controller
- Variable displacement pumps

Because of the fluid compressibility also in an **ideal pump** the input and output flow rates are different if input and output pressures are different.
The mass flow rates are the same in an ideal pump without external leakage.

Theoretic drive torque $\qquad T_{\text{theor}} = \dfrac{\Delta p \cdot V_{\text{g}}}{2 \cdot \pi}$

Swept volume V_{g} [m^3/r]

Pressure difference Δp [N/m^2]

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Performance of pumps and motors

PUMP
Wilson's model
Flow rate (output)

$$q_{V2} = \varepsilon V_i n - C_s \frac{V_i \Delta p}{2\pi \nu \rho}$$

Pump torque (input)

$$T = \varepsilon \frac{V_i \Delta p}{2\pi} + C_f \frac{V_i \Delta p}{2\pi} + C_v V_i n \nu \rho + T_c$$

ε	Pump angle set value (0 - 1)
V_i	displacement (per revolution)
n	rotational speed (1/s)
C_s	laminar flow loss coefficient
Δp	pressure difference over pump
ν	fluid kinematic viscosity
ρ	fluid density
C_f	Coulomb friction coefficient
C_v	viscous friction coefficient
T_c	constant torque loss

Ideal pump and motor

Wilson's model is very simplistic and it can't explain all the phenomena in pumps (and motors).
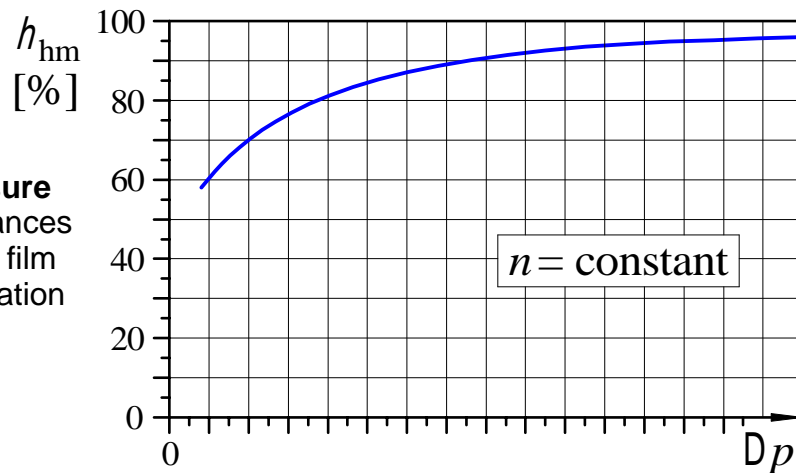
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Effective drive torque

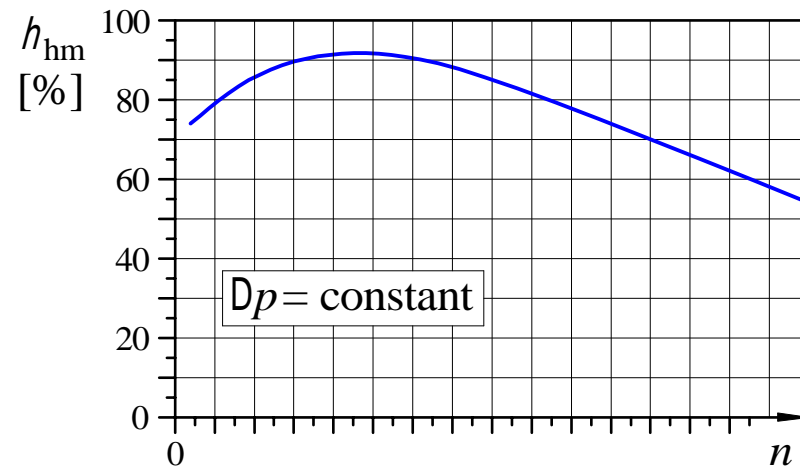$$T_{\text{real}} = \frac{\Delta p \cdot V_g}{2 \times \pi \times h_{\text{hm}}}$$

Wilson's pump model does not explain the phenomena seen in the figures below well

Friction – hydromechanical efficiency h_{hm}

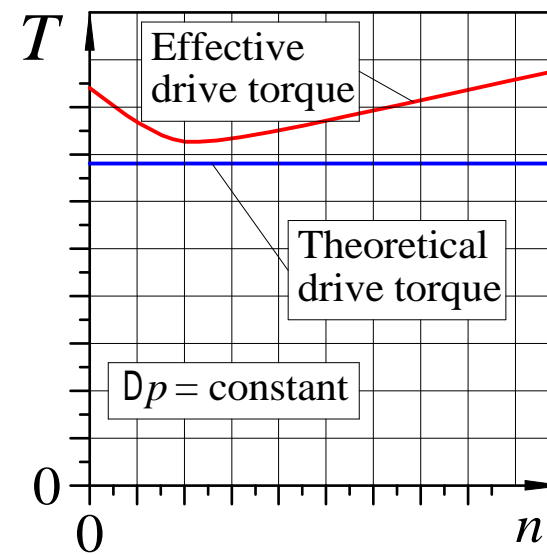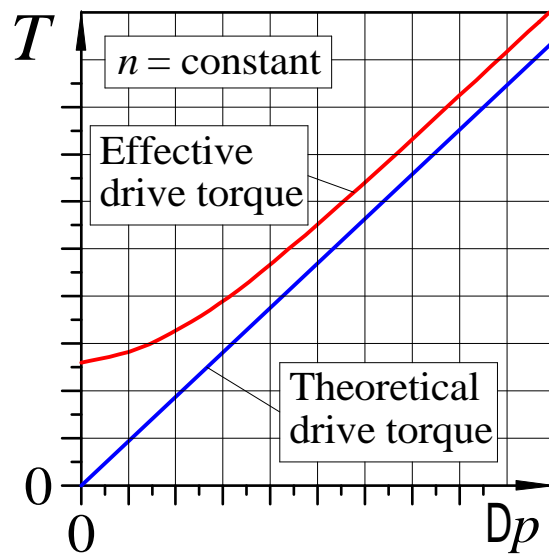$$T = \varepsilon \frac{V_i \Delta p}{2\pi} + C_f \frac{V_i \Delta p}{2\pi} + C_v V_i n \nu \rho + T_c$$

Higher pressure
- Wider clearances
- Thicker fluid film
- Better lubrication

h_{hm} [%] $n = $ constant, axis Δp

h_{hm} [%] $\Delta p = $ constant, axis n

Effect of n
- At first Coulomb friction dom-inates
- Then bet-ter lubrica-tion
- Finally hy-draulic losses in-crease

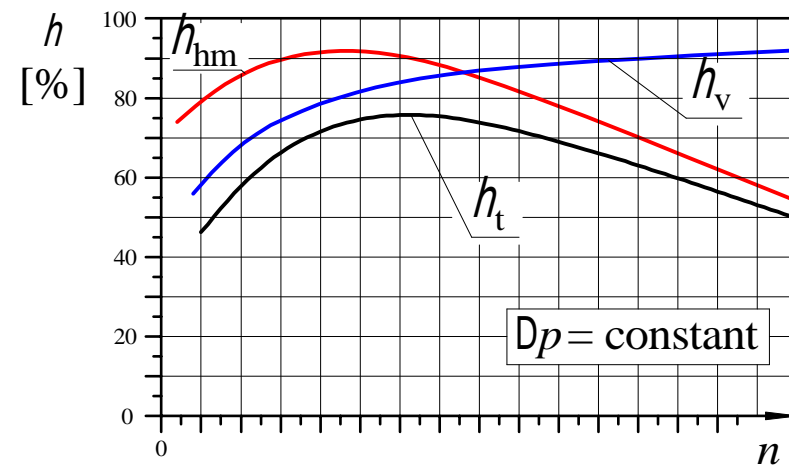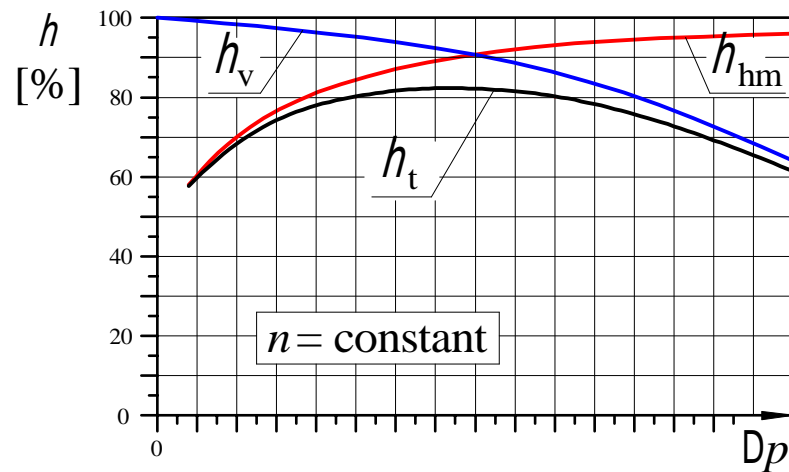# Theoretical drive torque – Effective drive torque
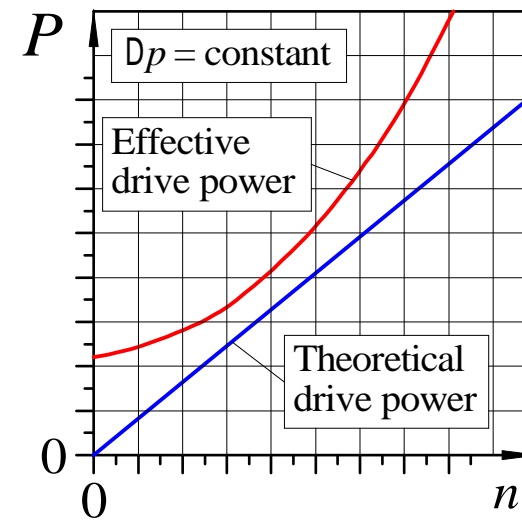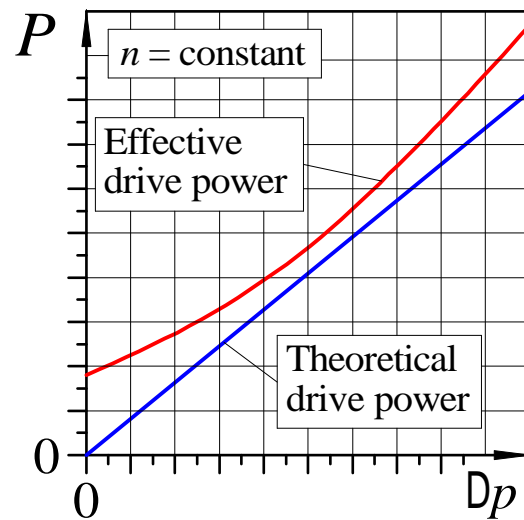
Theoretic drive power $\qquad P_{\text{theor}} = q_{\text{V}} \cdot \mathrm{D}p$

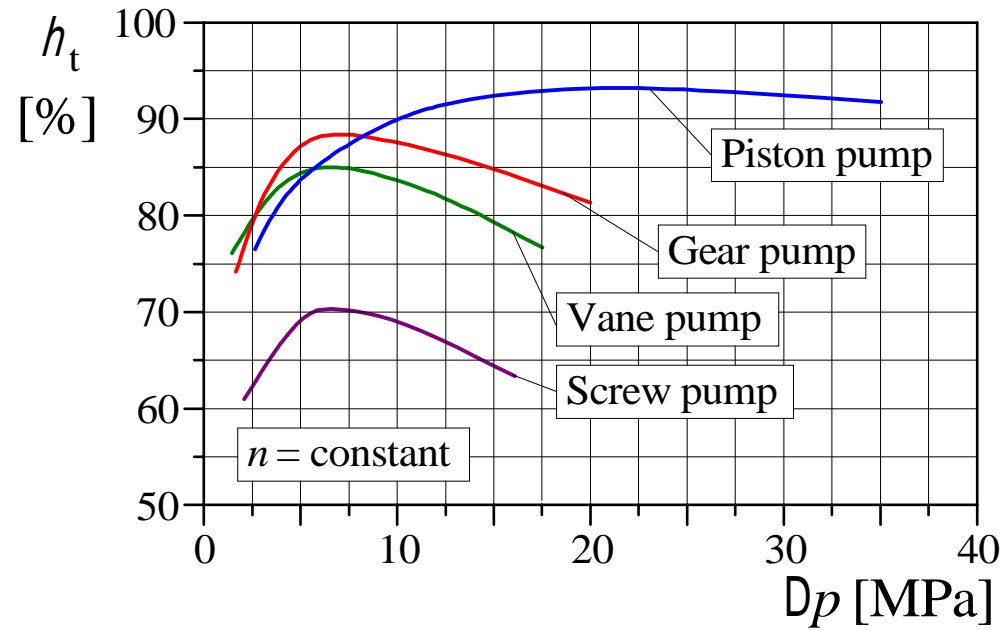Effective drive power $\qquad P_{\text{real}} = \dfrac{q_{\text{V}} \cdot \mathrm{D}p}{h_{\text{t}}} \qquad\qquad h_{\text{t}} = h_{\text{v}} \cdot h_{\text{hm}}$
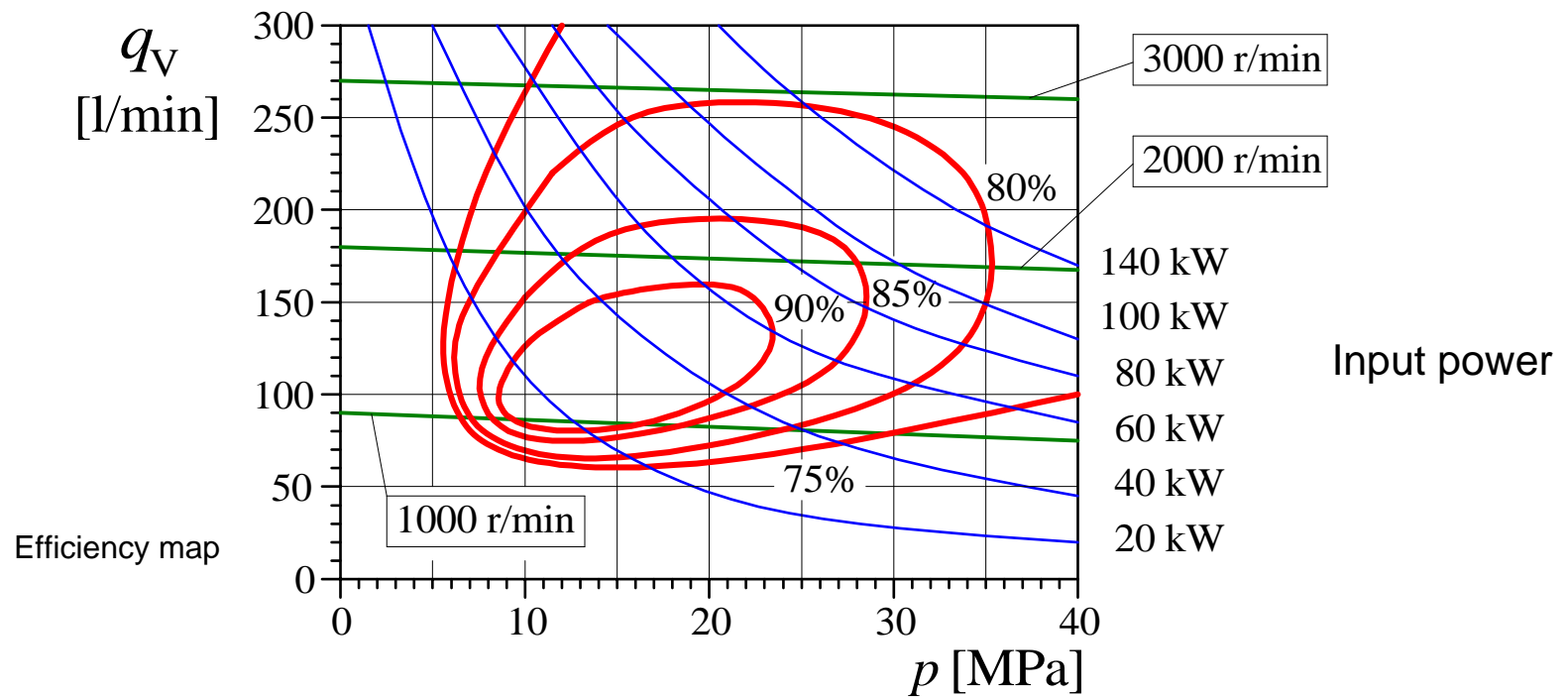
Theoretical drive power – Effective drive power

Comparison of structure types

h_t [%]

Piston pump

Gear pump

Vane pump

Screw pump

$n = $ constant

Δp [MPa]
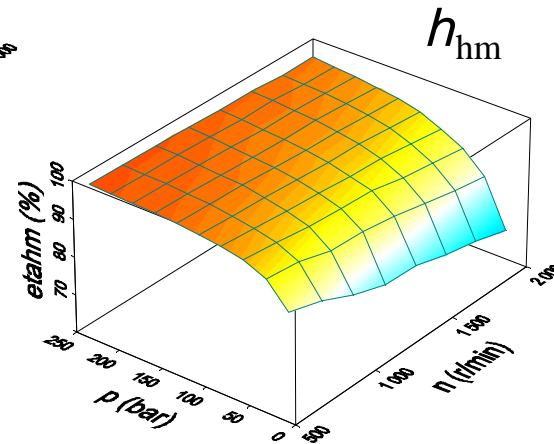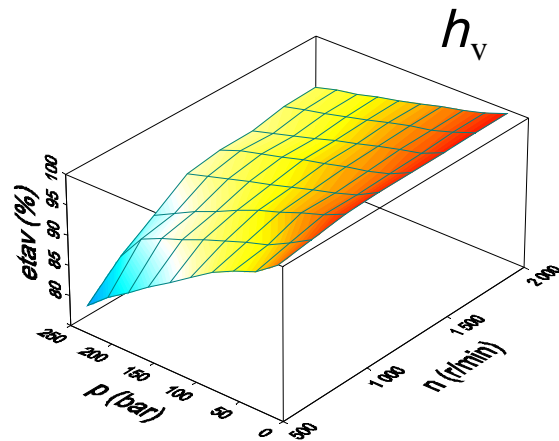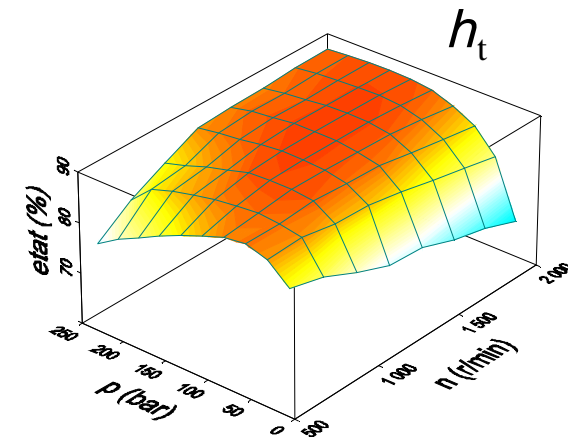
Characteristic curves of pump
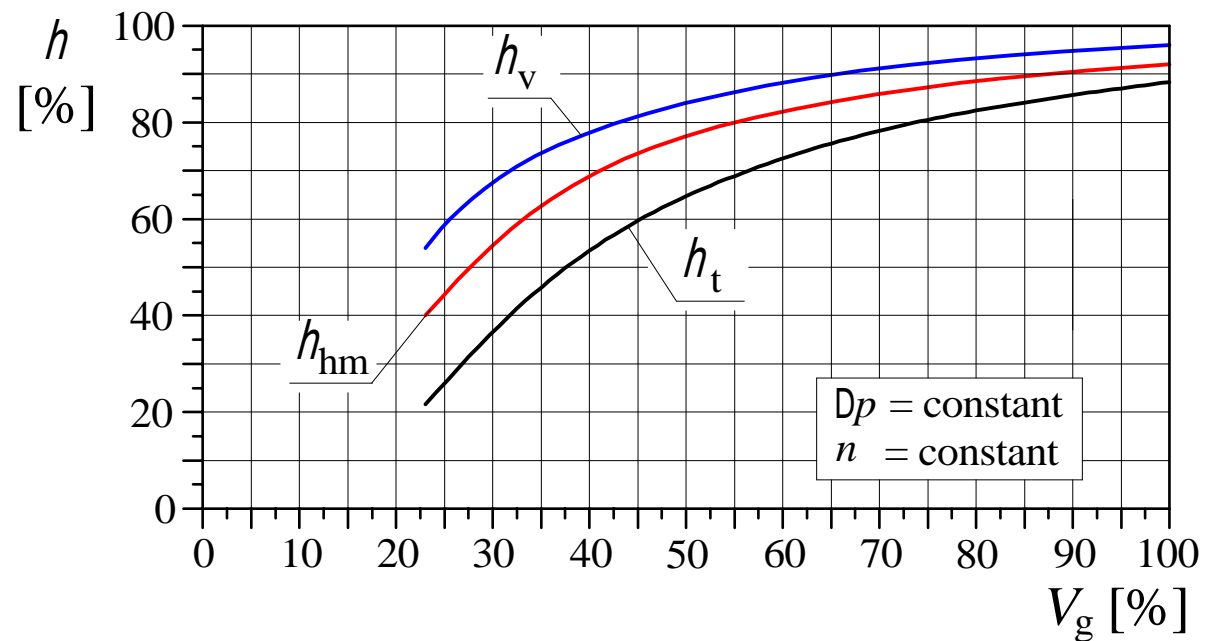
Example:
Pressure-rotational speed-dependency of axial piston pump

$$T = \varepsilon \frac{V_i \Delta p}{2\pi} + C_f \frac{V_i \Delta p}{2\pi} + C_v V_i n \nu \rho + T_c$$

⇐ Wilson's model (check the effect of decreasing ε)
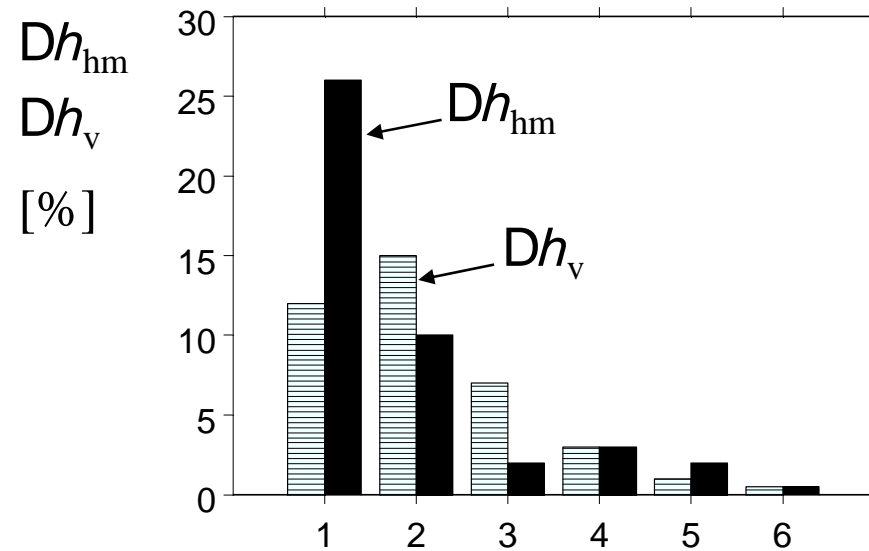
ε Pump angle set value (0 - 1) (axial piston pumps)

$$q_{V2} = \varepsilon V_i n - C_s \frac{V_i \Delta p}{2\pi \nu \rho}$$

Effect of displacement setting value to the efficiencies in variable displacement pumps

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Factors affecting efficiency

Dh_{hm}
Dh_{v}

[%]

1: pressure   2: swept volume   3: rotational speed
4: temperature   5: pump specimen   6: fluid

Pump types

Gear pumps

- external gear

- internal gear

Screw pumps

Vane pumps

- vanes in rotor
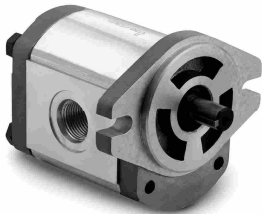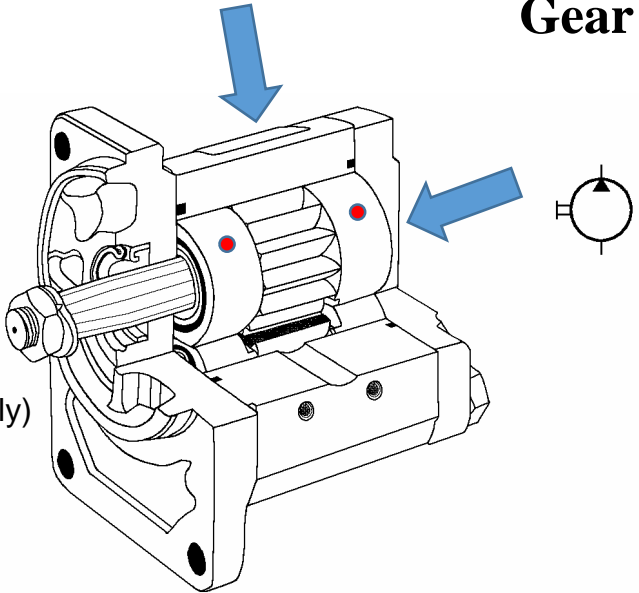
- vanes in stator

Piston pumps

- line piston pumps

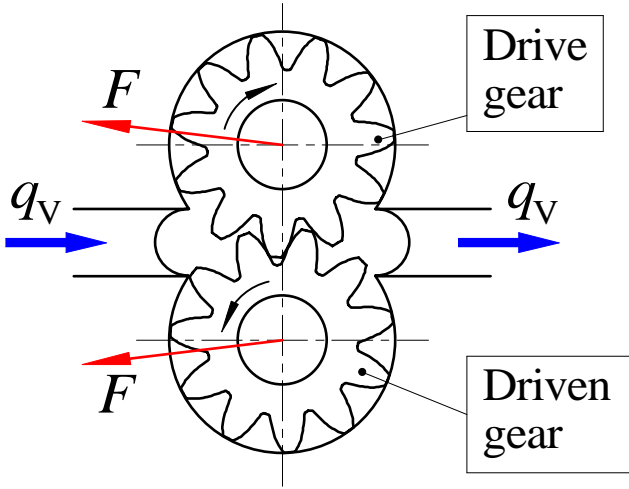- radial piston pumps

- axial piston pumps

Gear pumps

Reduction of leakage by
- axial gap compensation
- radial gap compensation
in some models
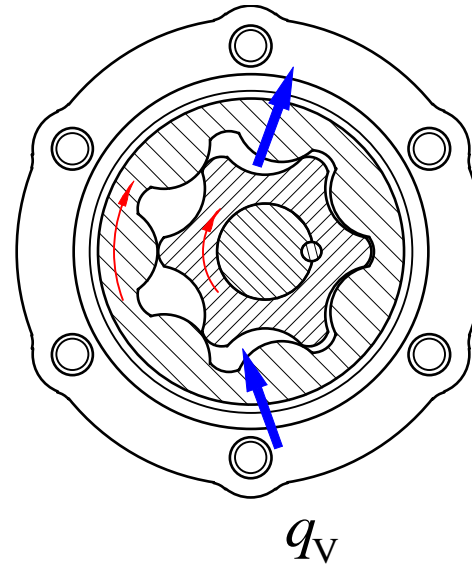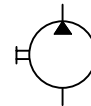
- Hydrodynamic bearings (typically)
- Roller bearings

External gear
- two gear
- multi gear

F

q_V q_V

Drive
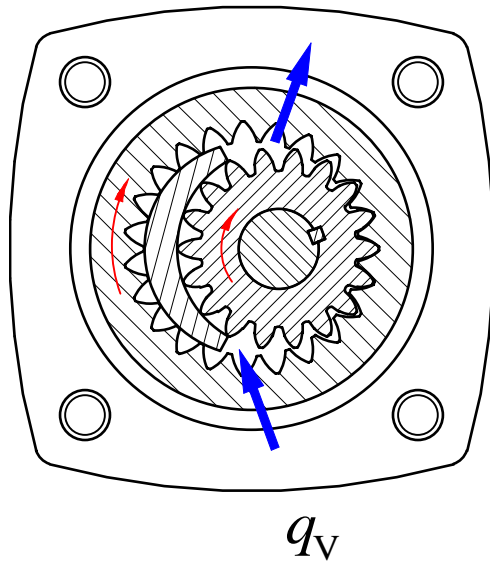gear

F

Driven
gear

Internal rotor
External rotor

Axial compensation
Radial compensation
possible

- Low noise
- Even flow

q_V

Internal rotor
External rotor

Also "roller rotor" design

q_V

Internal gear
- crescent (segment pump)
- gerotor   (ring pump)

Performance characteristics of gear pumps

Total efficiency max. h_t » 0.8 – 0.93
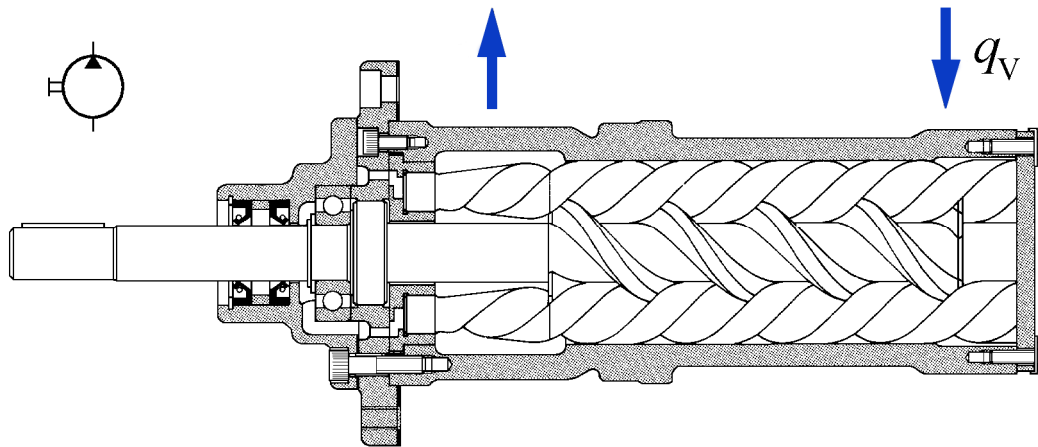
Rotational speed range n » 500 – 5000 r/min

Operating pressure max. p » 14 – 21 (– 32) MPa

- depends on compensation of leakage
  and radial forces

Screw crests roll against screw roots and seal fluid chambers

Screw pumps

q_V

Fluid volumes do not change during movement
- Even flow
- Low noise
- High rotational speeds possible

Number of screws
- one
- two
- three

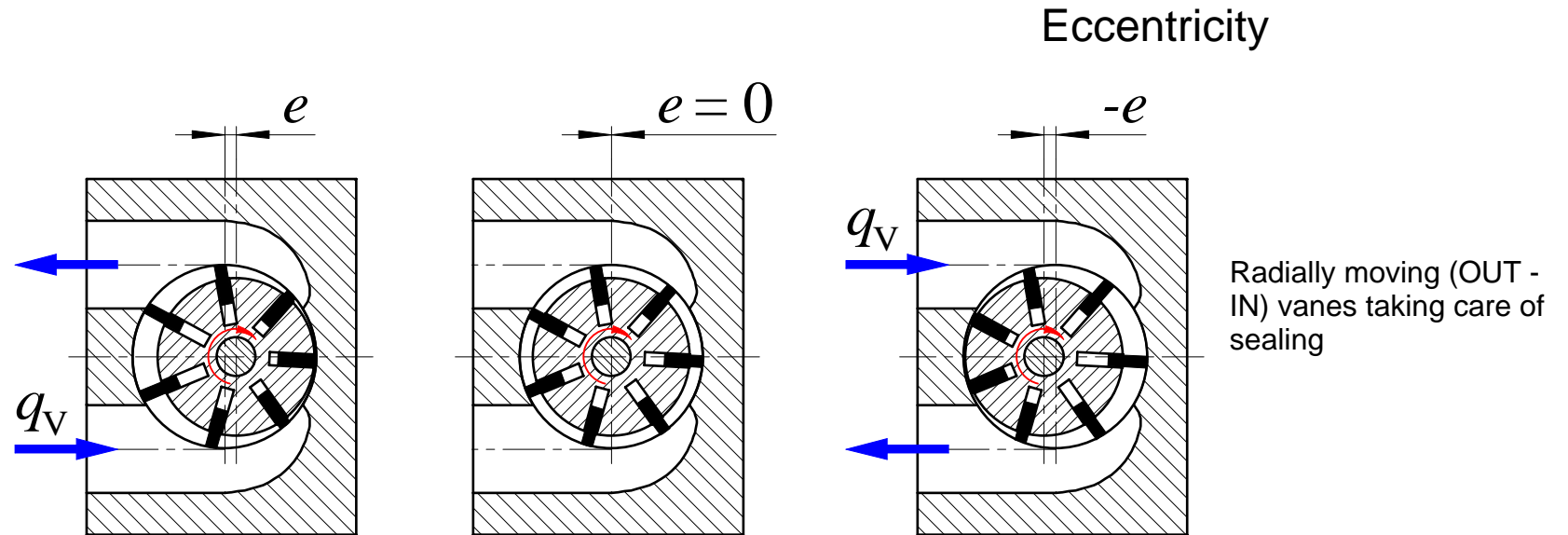Total efficiency max. h_t » 0.7 – 0.8

Rotational speed max. n » 30000 r/min

Operating pressure max. p » 14 – 20 MPa

By altering eccentricity of rotor the displacement and even the flow direction can be changed.

Vane pumps

Eccentricity

e $e = 0$ $-e$

q_V

q_V

q_V

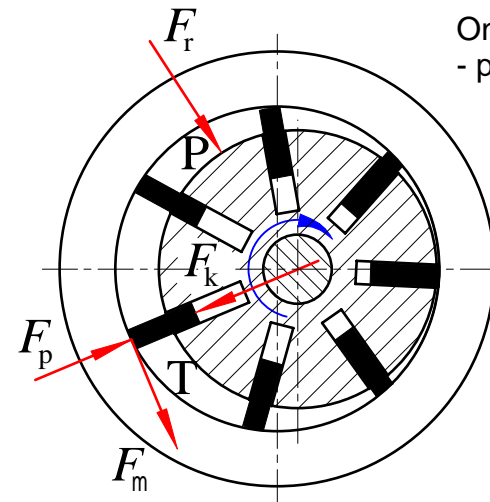Radially moving (OUT - IN) vanes taking care of sealing

Vanes are pushed outwards to make a contact with the pump body
- with extra force behind vane (spring or pressure)
- "centrifugal force" (not a real force)
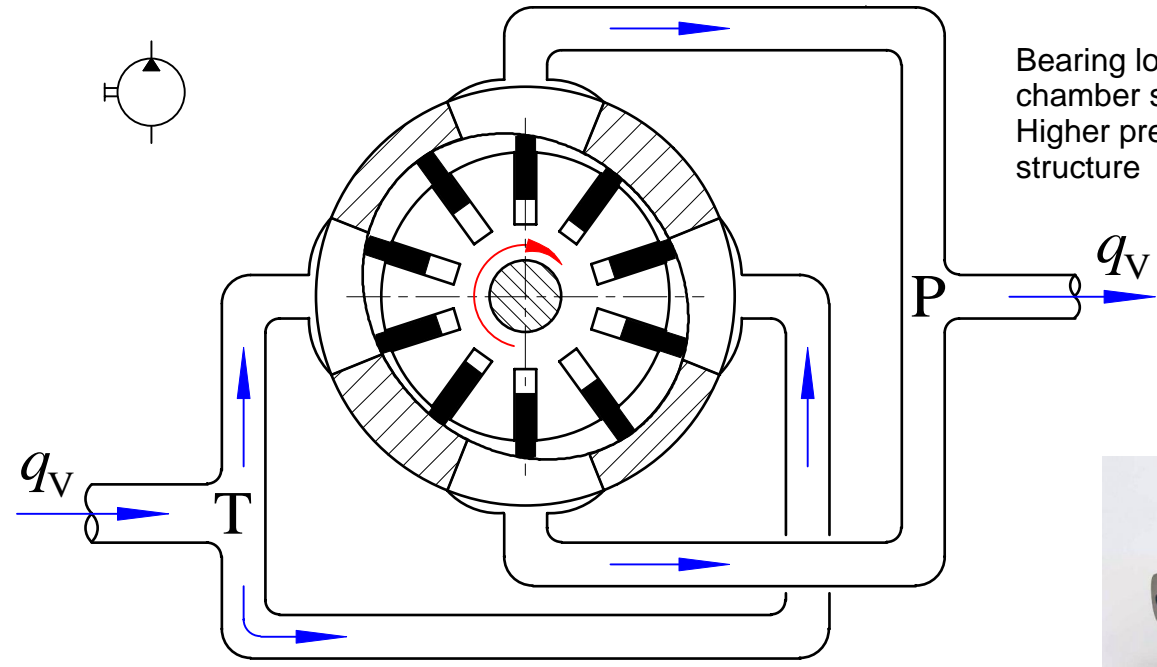
Variable displacement operation

Control piston

One chamber structure
- pressure forces acting on the shaft

F_r

P

F_k

F_p

T

F_m
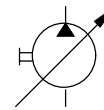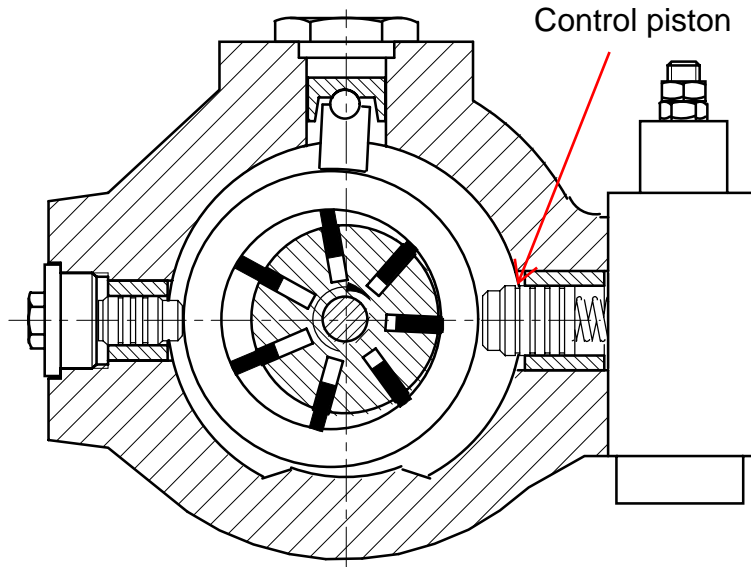
Even flow
- low noise

Vanes in rotor
- one chamber

Two chamber structure doubles the flow rate
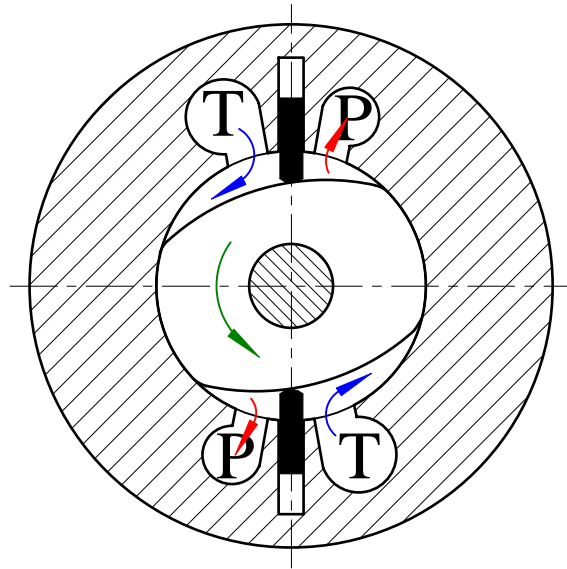(compared with one chamber models)

Bearing load forces are smaller than in one
chamber structure
Higher pressures possible than in one chamber
structure

q_V

P

Variable displacement operation
possible with two pump rings in
series

q_V

T

Vanes in rotor
- two chamber

Flow rate varies slightly more than in other structures

Vanes in stator

Performance characteristics of vane pumps
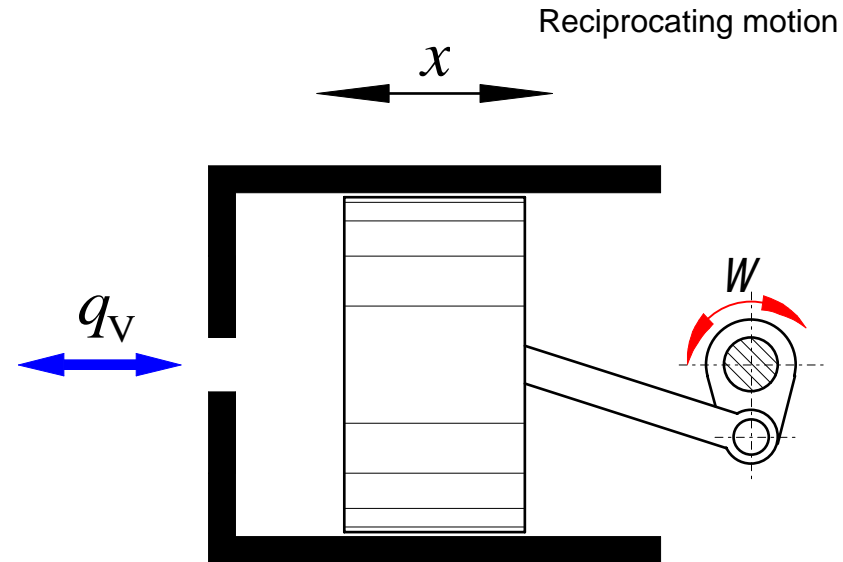
Total efficiency max. h_t » 0.8 – 0.92

Rotational speed range n » 600 – 2500 r/min

Operating pressure max. p » 7 – 14 (– 18) (– 21– 28) MPa

- depends on compensation of leakage,
  radial forces and number of chambers
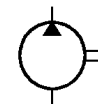
Piston pumps

Reciprocating motion

$$x$$

$$W$$

$$q_V$$
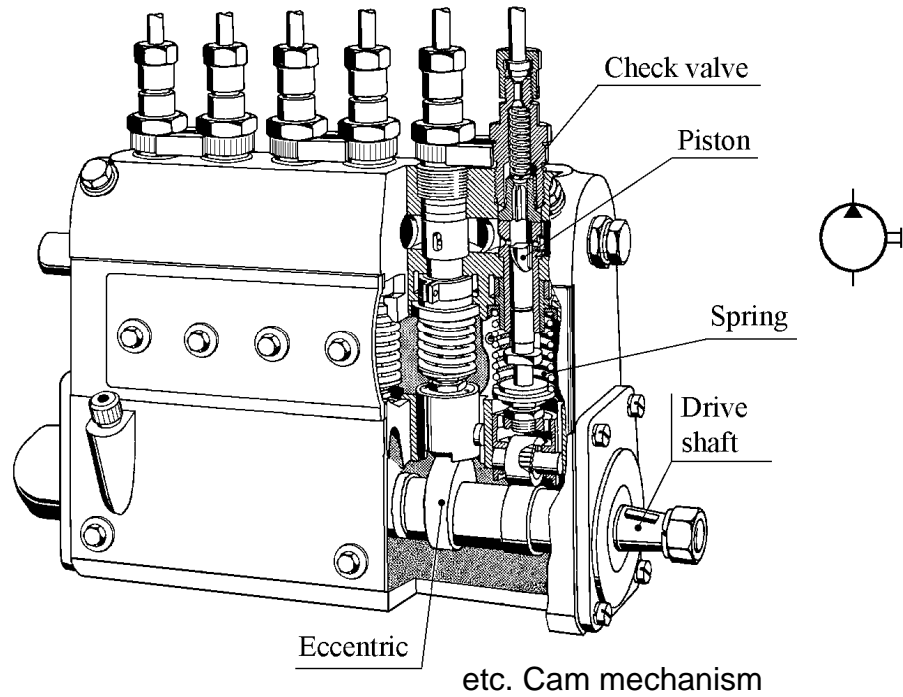
Piston pumps
- line piston pumps
- radial piston pumps
- axial piston pumps

Small clearances $\Rightarrow$ Small leakages $\Rightarrow$ Good volumetric efficiency

Check valve

Piston

Mainly for high very pressures
$\Rightarrow$ 1200 bar $\Rightarrow$ 2500 bar

Spring

Drive
shaft

Eccentric

etc. Cam mechanism

Line piston pumps

Hydrostatic bearings between pistons and pump ring

Case drain

Variable displacement design

Movable pump ring

Eccentricity $\Longrightarrow$ stroke

Cylinder flows through hollow shaft

Controller
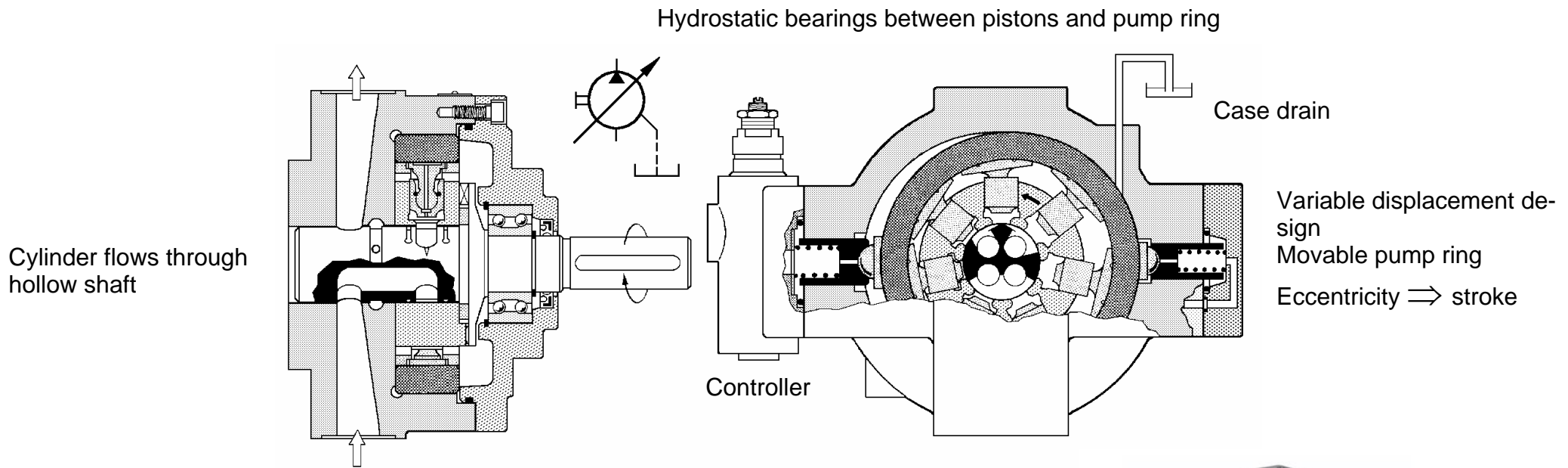
Radial piston pumps
- internal flow channels
- external flow channels

High pressures possible up to 450 bar

Springs for contact during suction

Cam operated

Check valves integrated in pistons
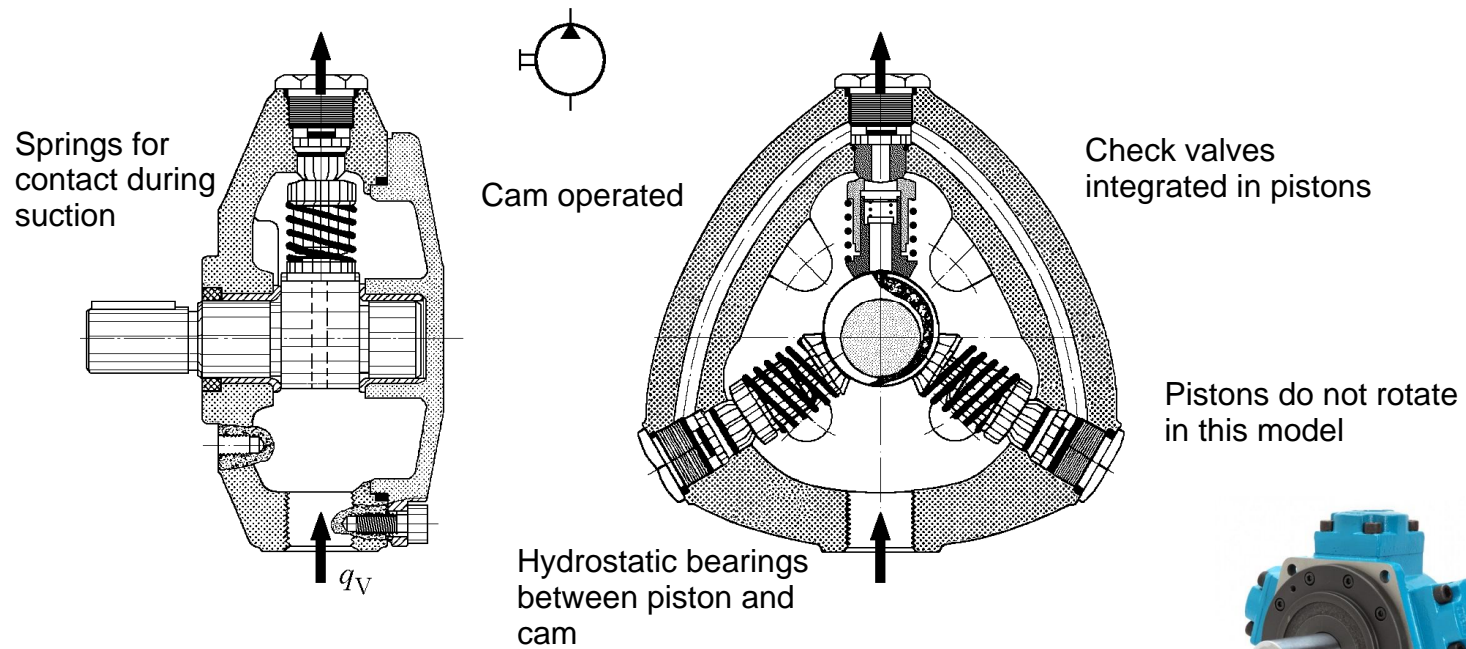
Pistons do not rotate in this model

q_V

Hydrostatic bearings between piston and cam

Radial piston pumps
- internal flow channels
- external flow channels

High pressures up to 600 - 700 bar possible
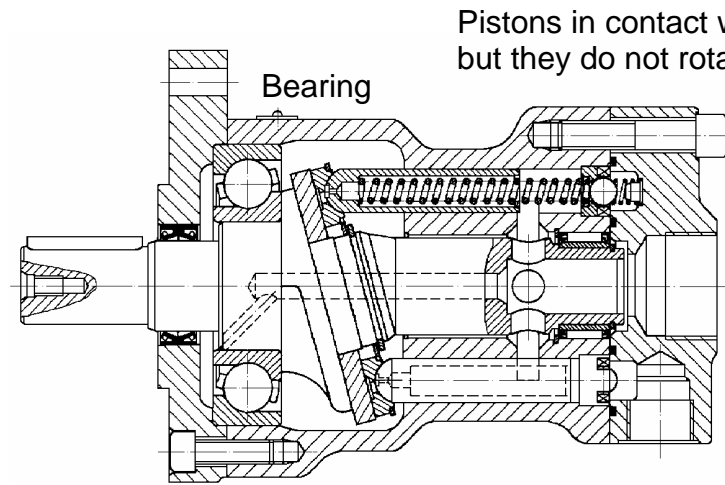
Pistons in contact with the plate,
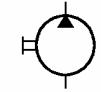but they do not rotate

Bearing

Possibility to transform this type to
"digital" by disconnecting some of
the pistons

The rotating masses are not in
balance which limits the rotational
speed

Without special arrangements the contact
forces between piston and wobble plate
cause radial forces to piston

Pistons in parallel with the shaft

Wobble plate rotates with shaft
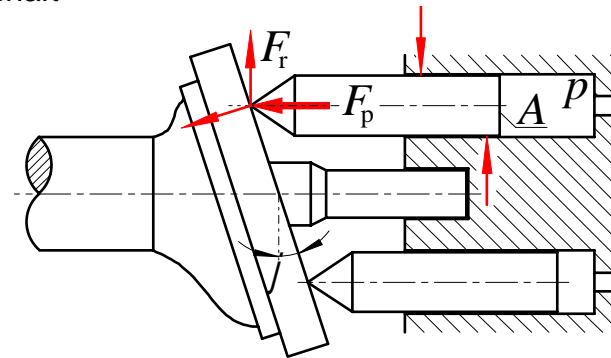
F_r

F_p

A p

Axial piston pumps
- wobble plate pumps
- swash plate pumps
- bent axis pumps

The direction of flow can be changed in some models

(-18° ®) 0° ® +18°

Contact forces between pistons and swash plate limit
the control angle
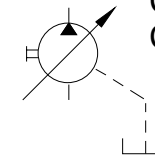Rotational speeds can be 1500 - 3000 rpm

Cylinder block rotates with the shaft
Swash plate does not rotate, it can
turn to control the piston stroke and
pump displacement

Counter piston for swash plate control

Hydrostatic bearing between
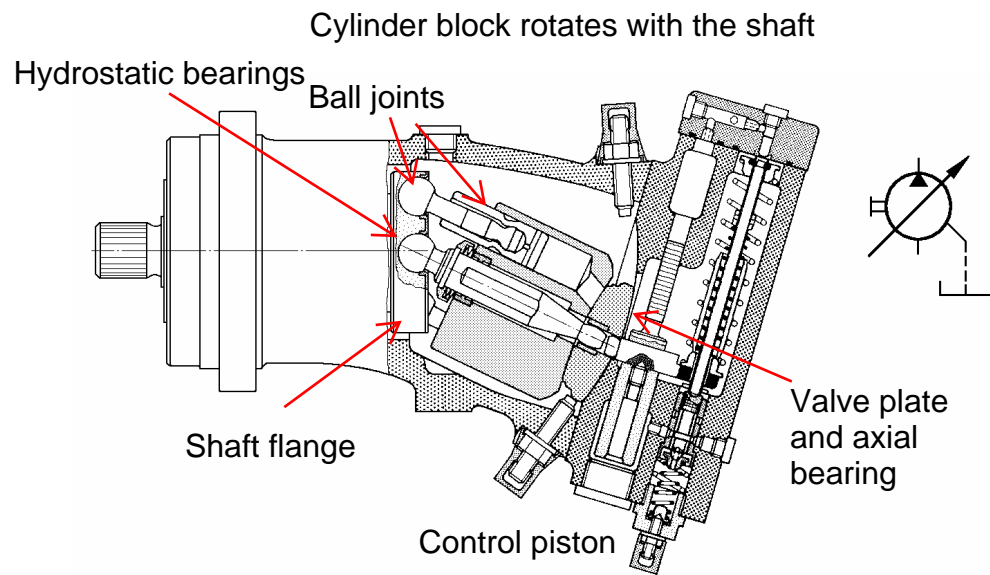pistons and swash plate
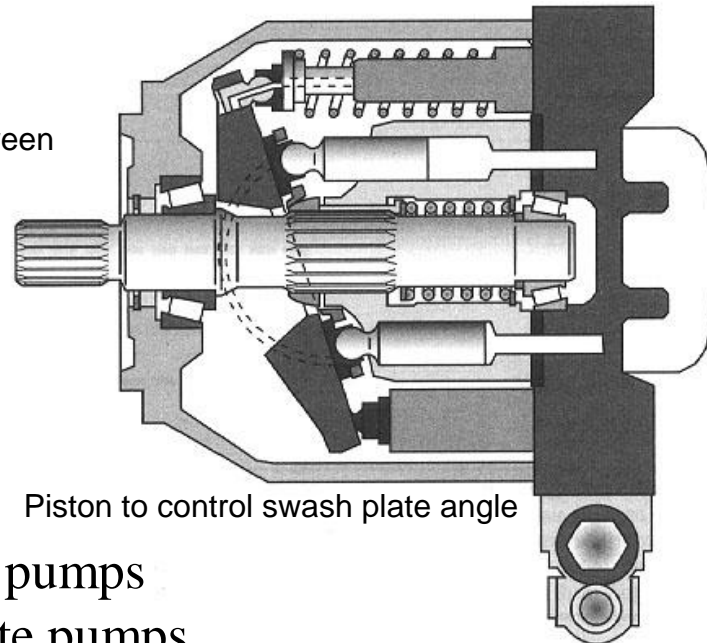
Piston to control swash plate angle

Case drain because of hydrostatic bearings
Case pressure must be kept small!

Axial piston pumps
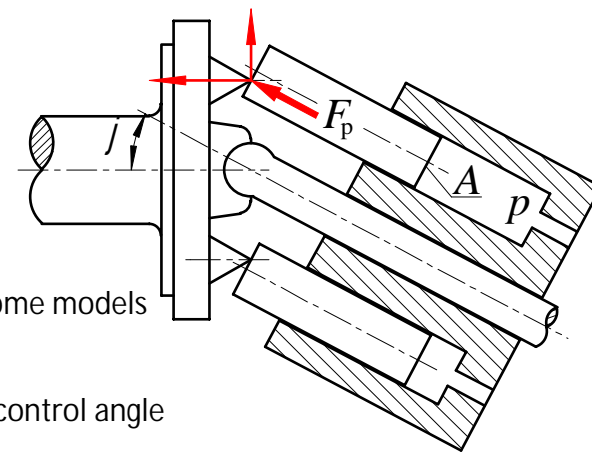- wobble plate pumps
- swash plate pumps
- bent axis pumps

swashplate pump - inline piston pump

Size is typically large especially with controllers

Cylinder block rotates with the shaft

Hydrostatic bearings

Ball joints

Shaft flange

Valve plate and axial bearing

Control piston

Rotation transmitted with
- Cardan shaft
- Bevel gear
- Pistons

Radial forces on piston are small

F_p

A p

Axial piston pumps
- wobble plate pumps
- swash plate pumps
- bent axis pumps

The direction of flow can be changed in some models

(-25° ®) 0° ® +25°

Contact forces of pistons do not limit the control angle as much in this model
Rotational speeds can be 800 - 8000 rpm

Performance characteristics of piston pumps

Total efficiency max. h_t » 0.8 – 0.9 even higher

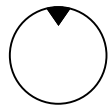Rotational speed range n » 300 – 8000 r/min

Operating pressure max. p » 20 – 35 (– 45) (– 70) (– 250) MPa

- depends on structure type
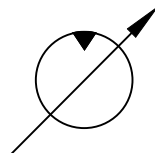
Hydraulic motors
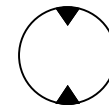
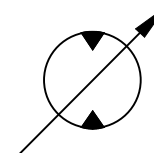Convert hydraulic power into mechanical power

Unidirectional   Bidirectional

Constant displacement

Variable displacement

Speed ranges and structures

Speed range	r/min
Slow	1- 150
Middle	10- 750
High	300- 5000

09 Series Hydraulic Motor

Parker Oildyne 09 gear motor

Most common construction types:
- gear
- vane
- piston

All operate on positive displacement principle

Pump-motors
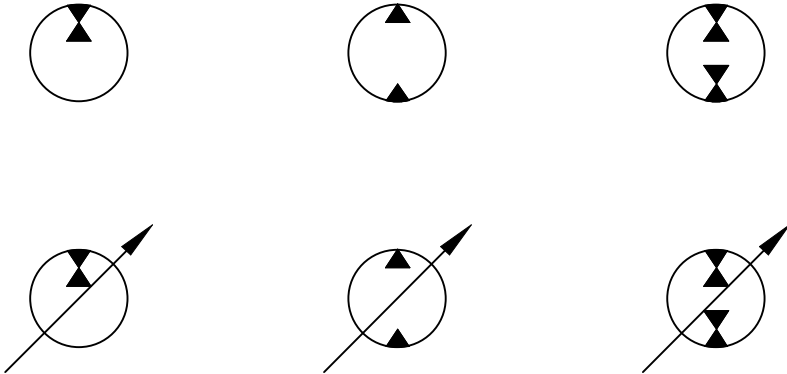
Control of flow direction
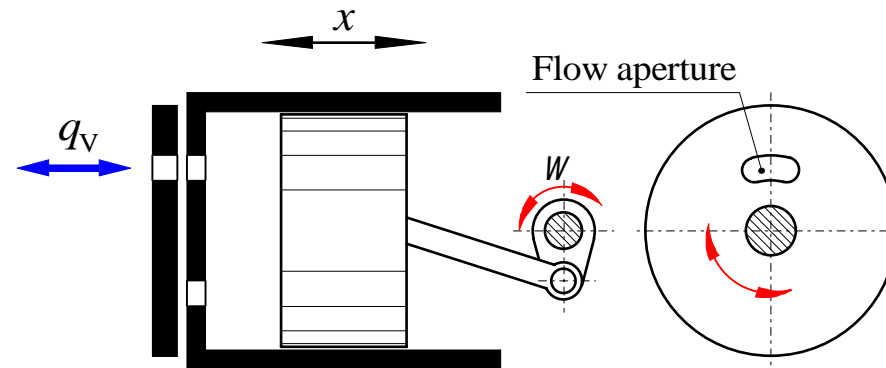
Only forced control is applicable

Operating phases:
Fluid flows into transfer volume   – work phase
Fluid flows out from volume   – free phase

Forced control

Motor characteristics

Starting characteristics

Especially at low rotational speeds
the operation can be uneven
a) rotational speed
b) torque
- displacement
- leakage
- friction (bearing type)
change

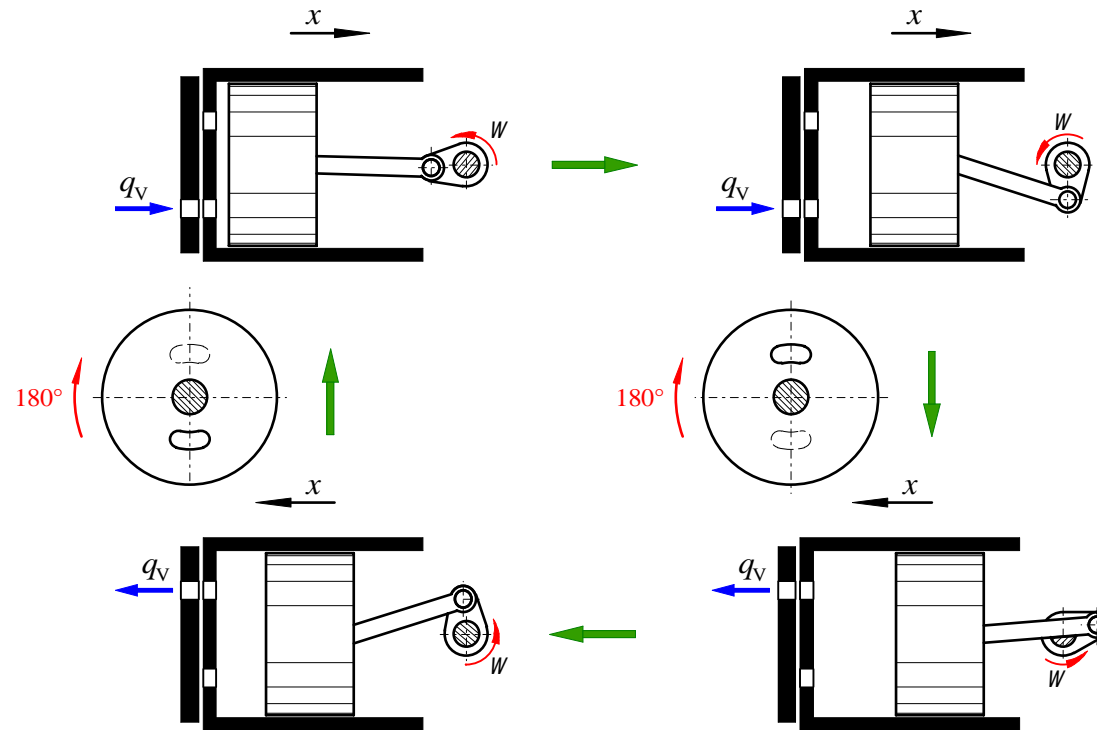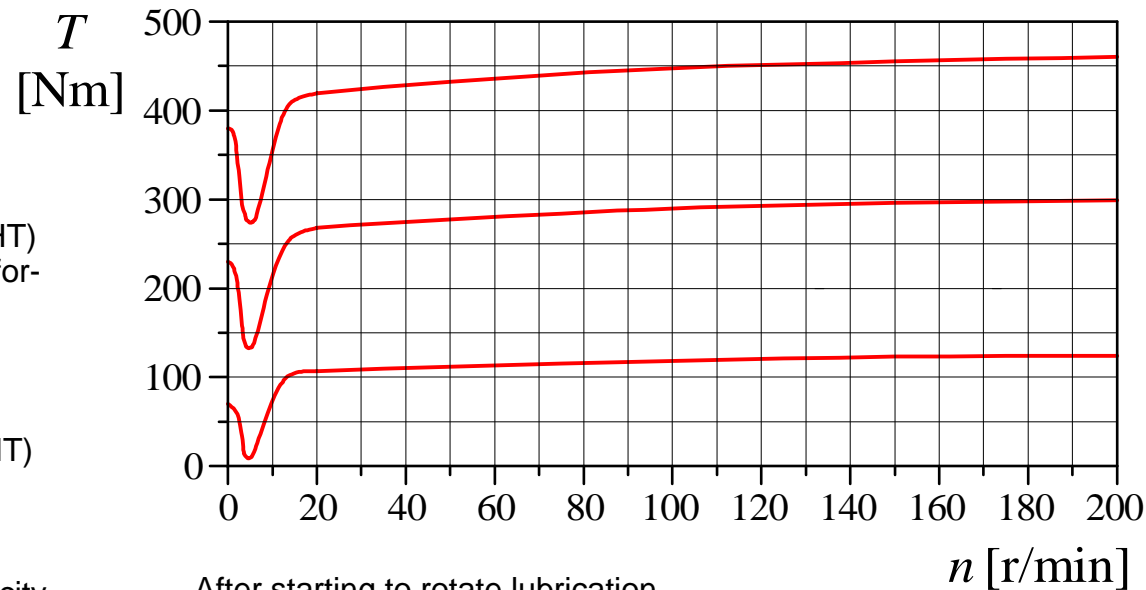Low Speed High Torque (LSHT)
motors have good quality perfor-
mance at low speeds!
Bearings
hydrostatic - hydrodynamic
Loading of motor?
Full load at slow speeds (LSHT)

Higher speeds:
- less (relative) leakage
- inertia effects -> steady velocity

After starting to rotate lubrication
problems with low speeds ->
torque diminishes

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Running characteristics

60 r/min

600 r/min

1500 r/min

Inertia effects help at high rotational speeds
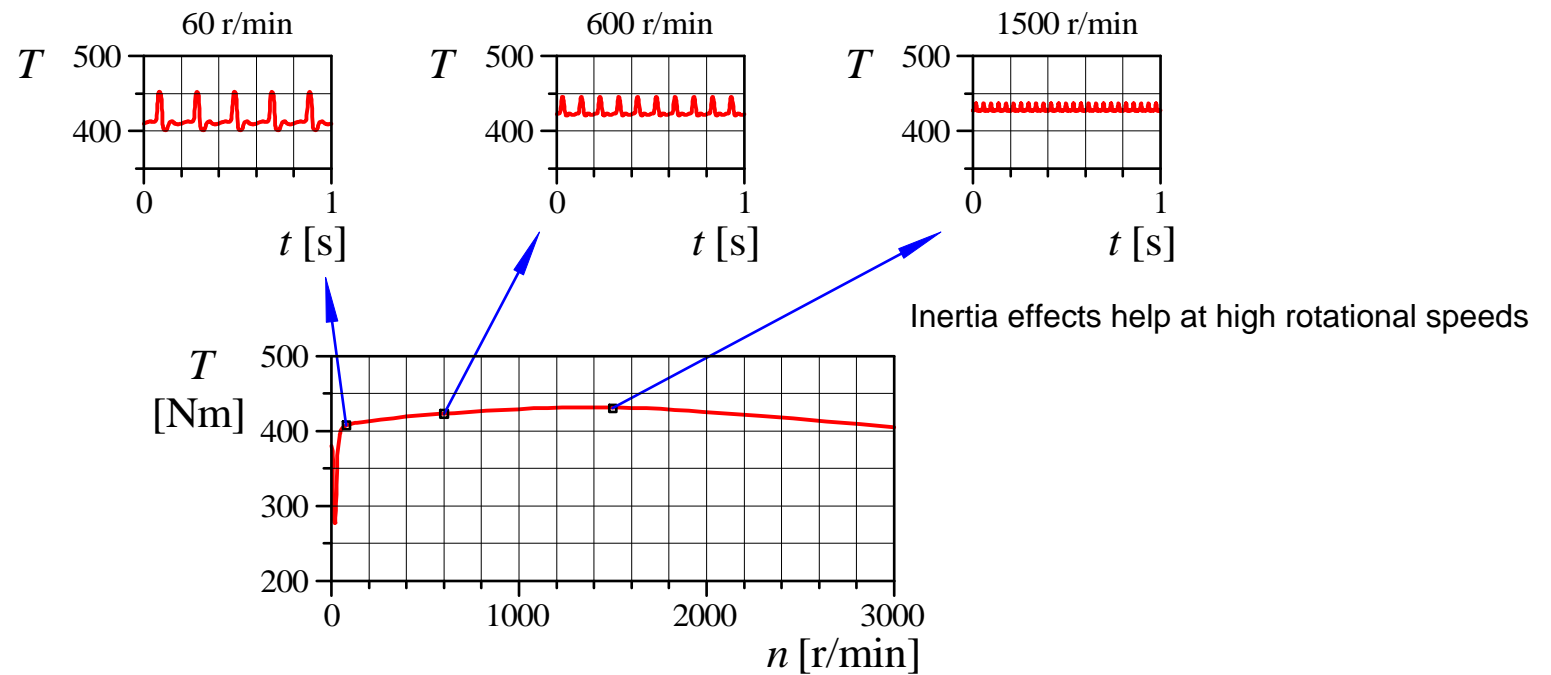
Theoretical flow demand $\qquad q_{\mathrm{V,theor}} = n \cdot V_{\mathrm{g}}$

Swept volume V_{g} [m^3/r] $\qquad\qquad\qquad$ cm^3/r = 10^{-6} m^3/r

Rotation speed n [r/s] $\qquad\qquad\qquad\qquad$ r/min = 1/60 r/s

$$q_{\mathrm{V,theor}} = \omega \cdot V_{\mathrm{rad}}$$

$$\omega = 2\pi \cdot n \qquad\qquad \text{Angular velocity } \omega \text{ [rad/s]}$$

$$V_{\mathrm{rad}} = \frac{V_{\mathrm{g}}}{2\pi} \qquad\qquad \text{Swept volume per radian } V_{\mathrm{rad}} \text{ [m}^3\text{/rad]}$$

Effective flow demand $\qquad q_{V,real} = \dfrac{n \cdot V_g}{h_v}$

Leakage – volumetric efficiency h_v

Leakage flows in motors

Internal leakage

Expansion

p_1　p_2

Inflow

Outflow

CONTROLLER

External leakage

In an ideal motor the flow rate out can be
higher than the flow rate in
(not the mass flow rate)

Control flow

Case leakage

L

Keep case pressure low!
Leakage flow straight to tank at
least if the outflow is pressurized!
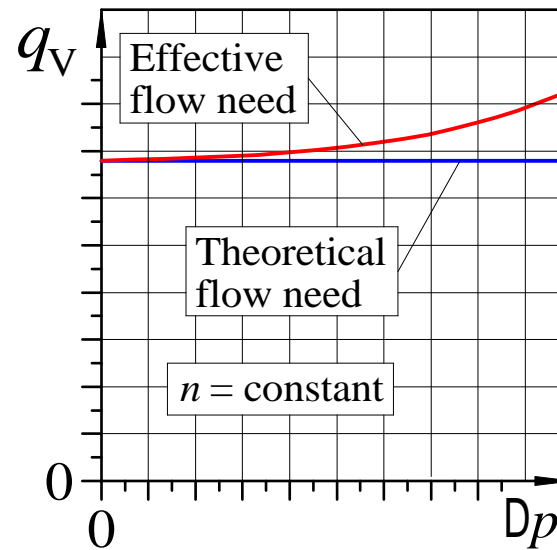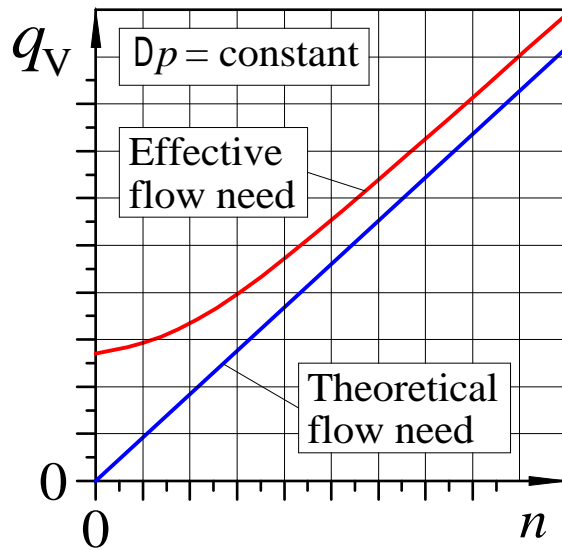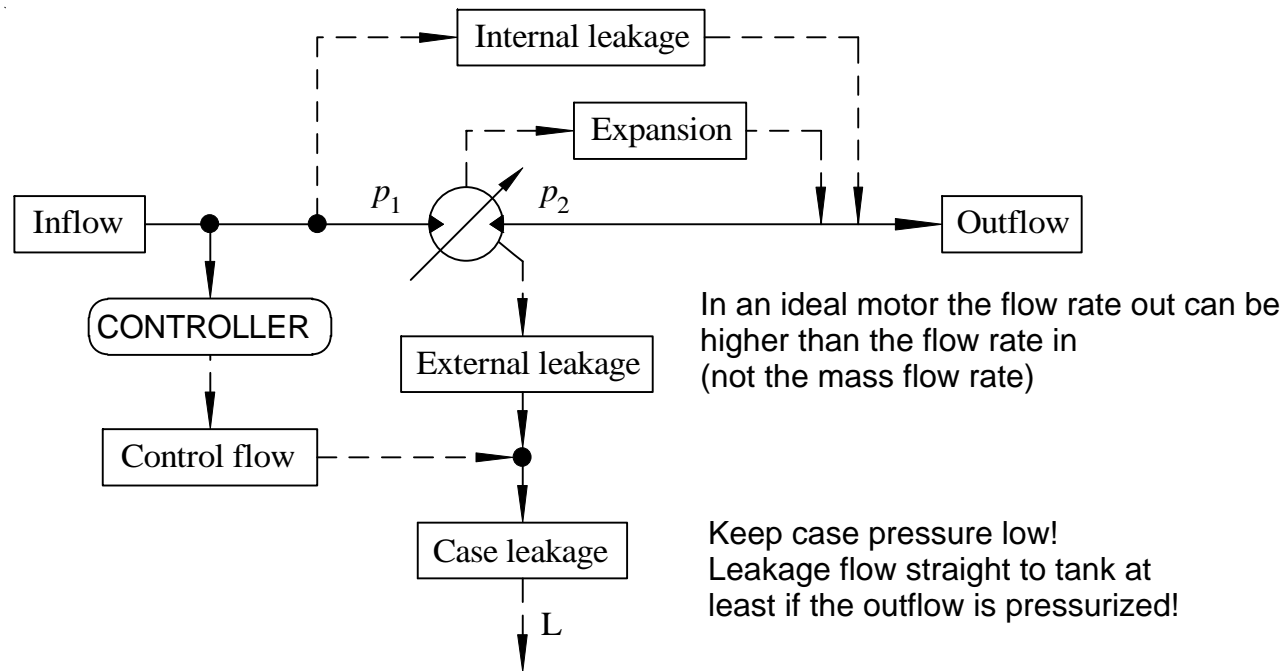
Theoretic pressure demand

$$\mathrm{D}p_{\text{theor}} = \frac{2 \cdot p \cdot T}{V_{\text{g}}}$$

Effective pressure demand

$$\mathrm{D}p_{\text{real}} = \frac{2 \cdot p \cdot T}{V_{\text{g}} \cdot h_{\text{hm}}}$$

Friction – hydro-mechanical efficiency h_{hm}
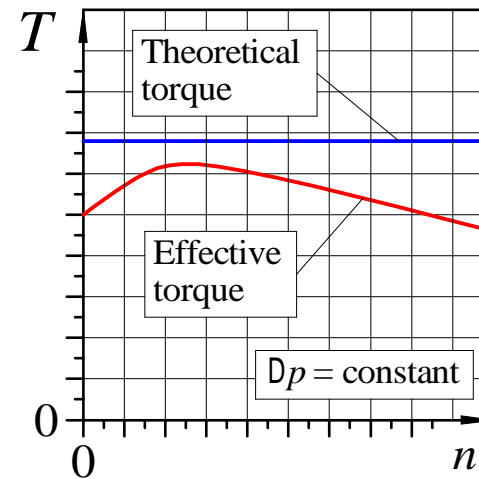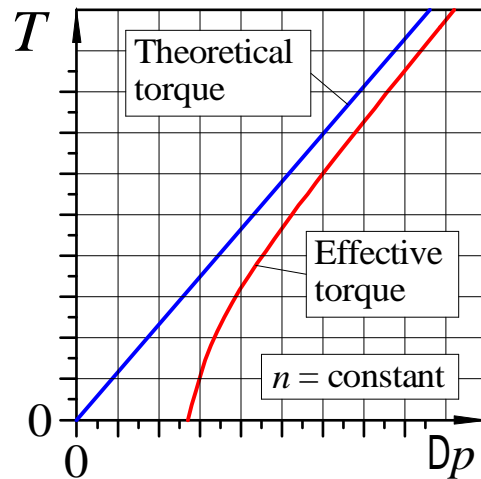
Theoretic power demand $\qquad P_{\text{theor}} = q_V \cdot \Delta p = T \cdot \omega$

Effective power demand $\qquad P_{\text{real}} = q_V \times \Delta p = \dfrac{T \cdot \omega}{\eta_t} \qquad \eta_t = \eta_v \times \eta_{\text{hm}}$

n = constant

Theoretical power

Effective power

Δp = constant

Theoretical power

Effective power
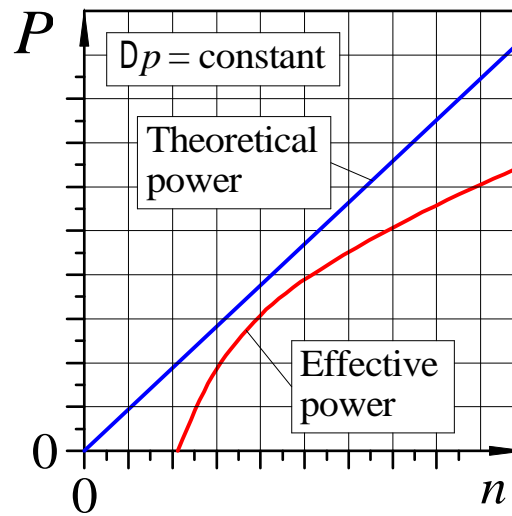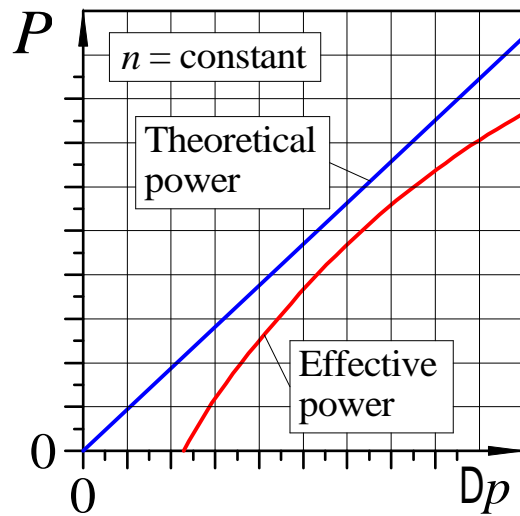
Power demand of load

$P_{\text{mech}} = T \times \omega = 2 \times p \times n \times T$

Characteristic curves of motor
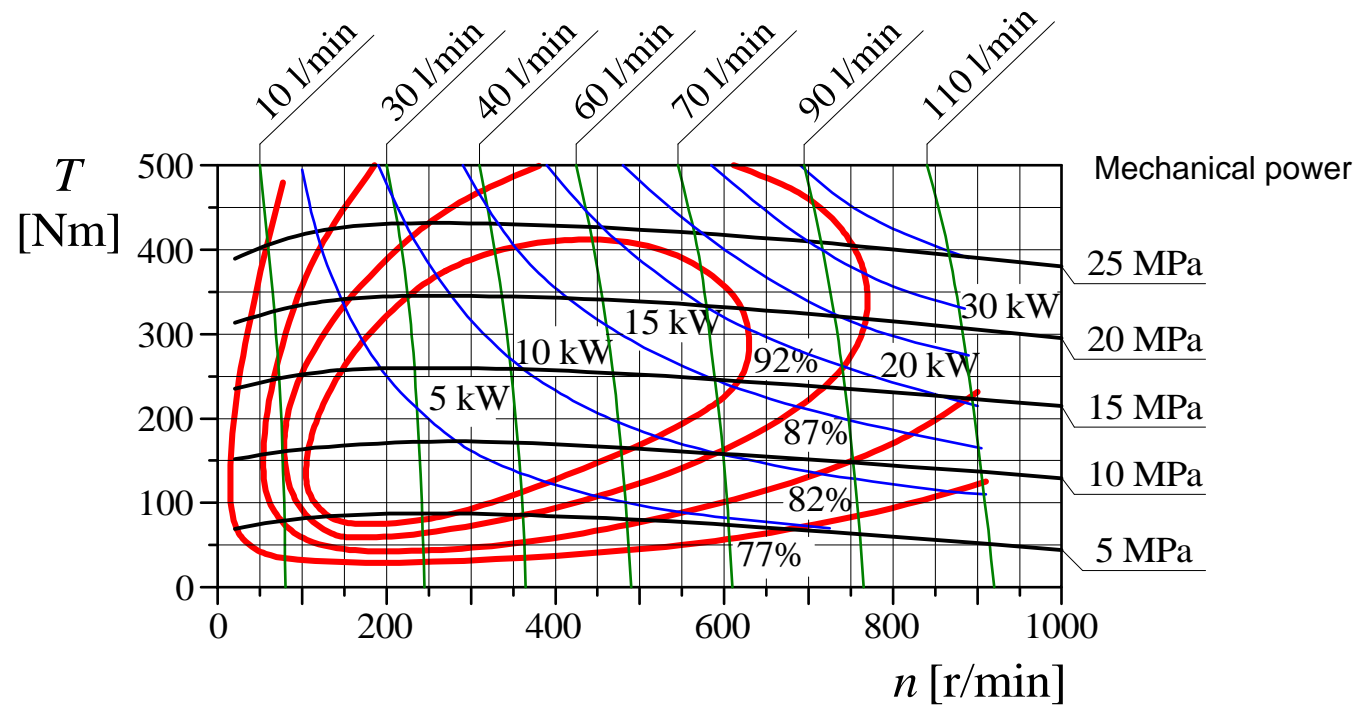
Low speed high torque motors (LSHT)

Large swept volume
- large displacement area of working elements
- several work stages per one rotation of the axle
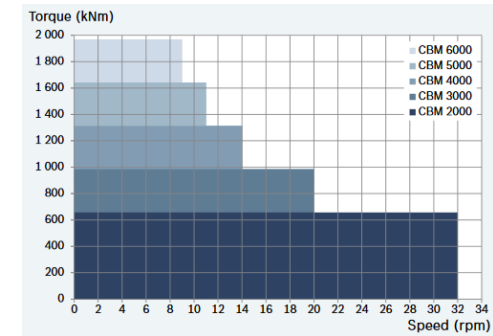
Radial piston motors
Vane motors with several chambers
Orbital motors

Total efficiency max. $h_t » 0.8 – 0.92$

Rotational speed range $n » 1 – 500 (– 2400)$ r/min

Torque max. $T » 1000 – 20000 (– 125000)$ Nm

Maximum torque 1.97 MNm

Bosch - Rexroth Hägglunds
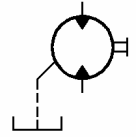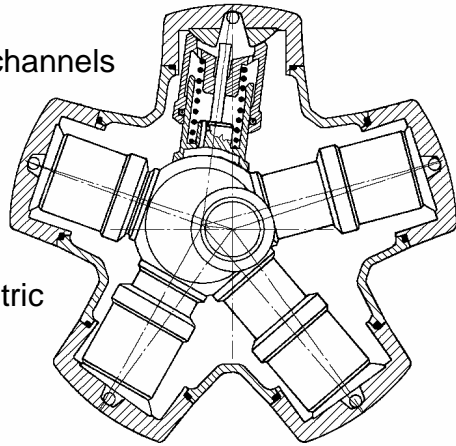
CBm radial piston motor

Radial piston motors
- external flow channels
- internal flow channels

Hydrostatic bearings

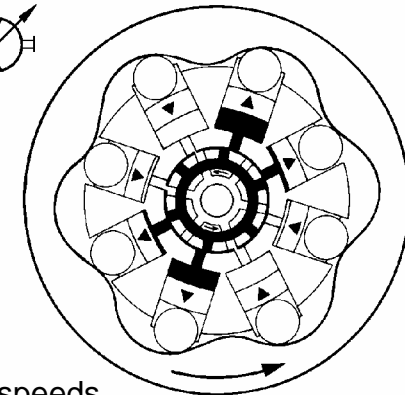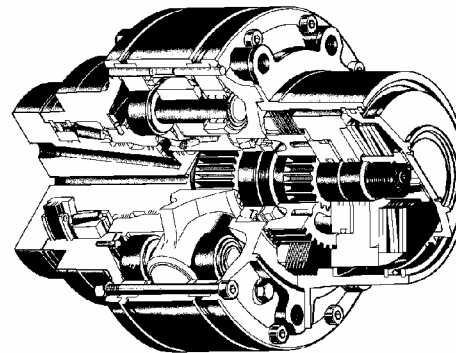External flow channels

Eccentric

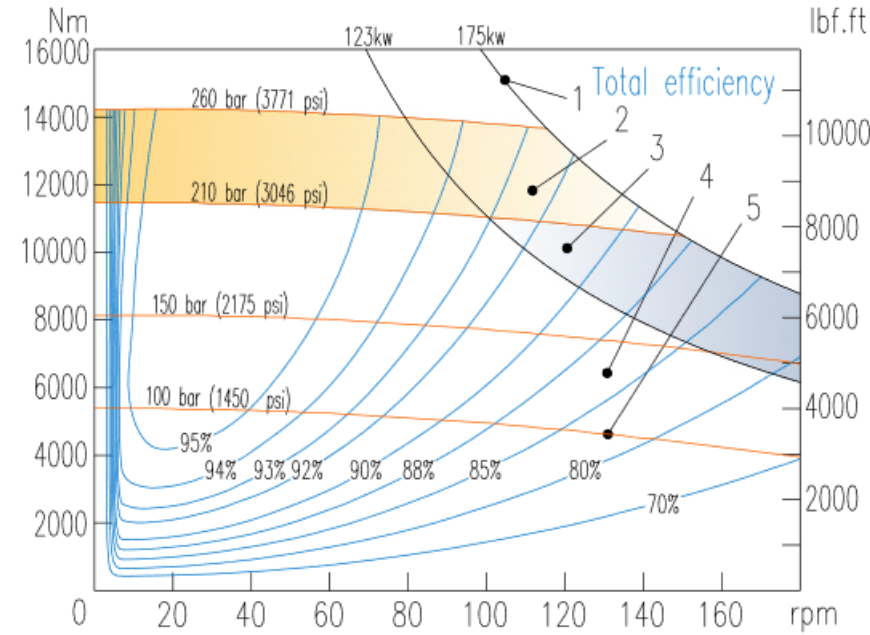Case drain

Up to 420 bar

Internal flow channels

Cam mechanism, several cams for one revolution

Rotating housing

Opposite cylinder forces cancel internal bearing loadings

Can be freewheeled (springs) during on-road operation (trailer), 2 speeds

https://www.blackbruin.com/

Stepwise change of displacement

Three displacements: 1, 2 and 1+2
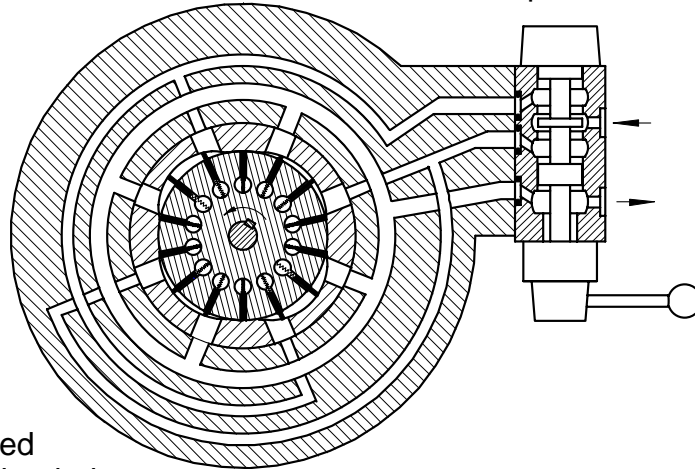
¬ Multi chamber vane motor

4 chamber model
Radial forces cancelled
Spring and pressure loaded vanes

http://files.danfoss.com/documents/520l0262.pdf

Drum distributor valve (or disc valve)

Orbital motor ®
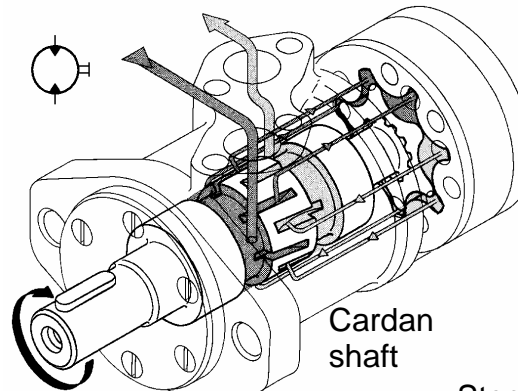
External gear
Stator, 7 cams
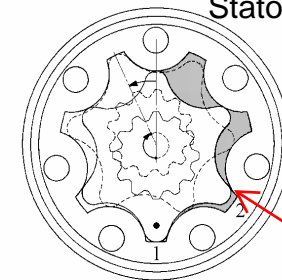
For one rotation 42 tooth volumes have to be filled

Cardan shaft

Rotor, 6 cams
Internal gear

Models with rollers also

Stepwise displacement changes possible

Middle speed range motors

Gerotor motors (ring motors)
Wobble plate motors

Total efficiency max. h_t **»** 0.8 – 0.88

Rotational speed range n **»** 200 – 1000 (– 1500) r/min

Torque max. T **»** 20 – 200 (– 1200) Nm

Distributor valve to control the cylinder flows

¬ Wobble plate motors
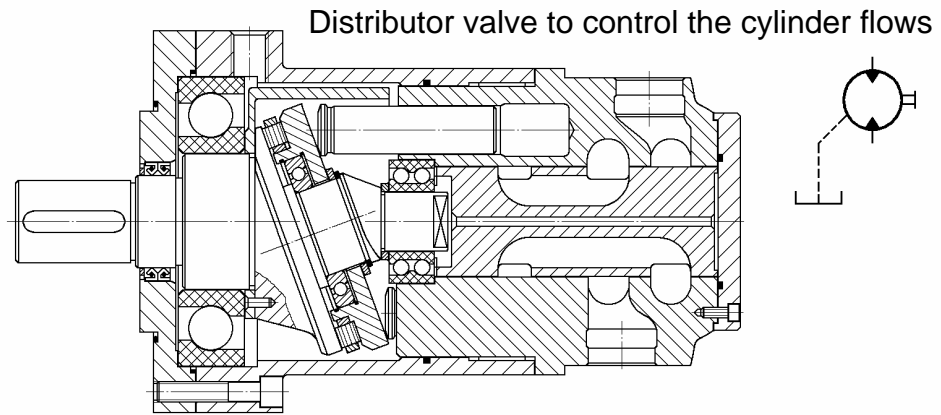
Constant displacement
Medium speed because of rotor's mass imbalance

Gerotor motor (ring motor) ®

q_V

q_V
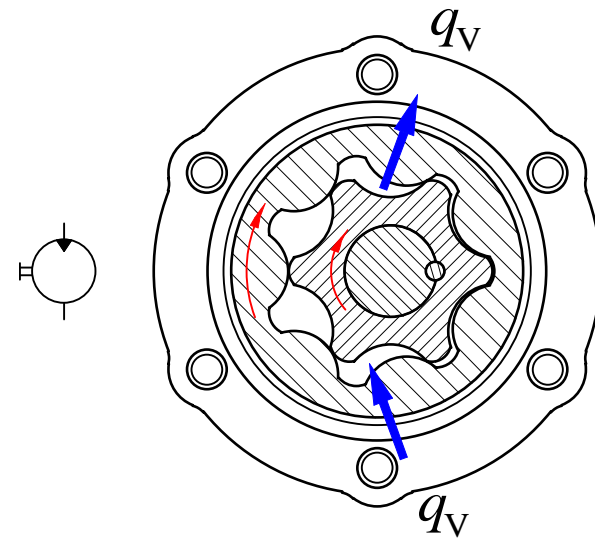
High speed range motors

External gear motors
Vane motors
Axial piston motors

Total efficiency max. h_t » 0.82 – 0.9

Rotational speed range n » 100 – 3000 (– 6000) r/min

Torque max. T » 10 – 700 (– 3000) Nm

Rotational speed may change frequently -> accelerations

Medium pressure (180 bar) unless radial compensation (250 bar)
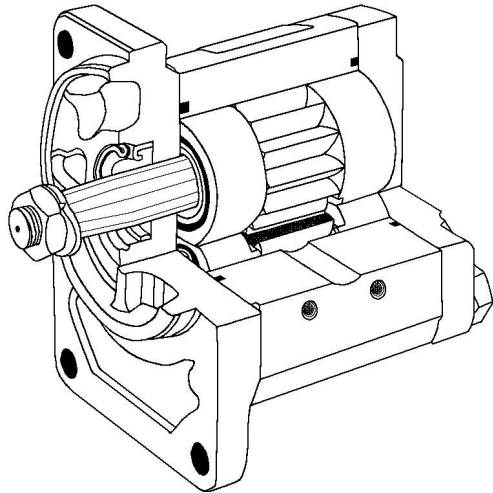
External leakage con-
nection needed if outlet
flow is pressurized

¬ External
gear motor

Medium pressure (up to 210 bar)

Constant or variable displacement

Some models with needle bearings for better
performance at low speeds

Vane motor ®

Hydraulic forces in
balance

External leakage connection for pressurizing of both
flow channels

Distributor valve   Static swash plate

Up to 350 bar
High efficiency

Constant or variable displacement
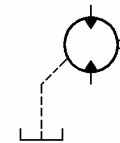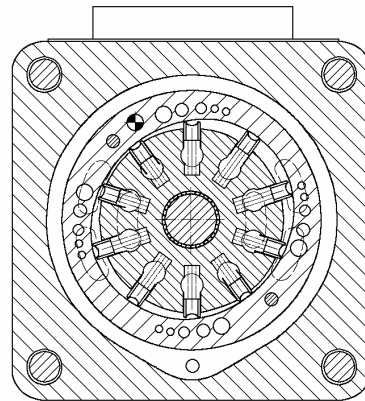In variable models typically 7°-18° control angle

Limited by radial forces (large) and and weakening
of hydromechanical efficiency (small)

Bent axis   ®
piston motor

Parker F12 bent axis motor

Volumetric efficiency.

Mechanical efficiency.

¬ Swash plate
piston motor

Distributor valve   Constant or variable displacement
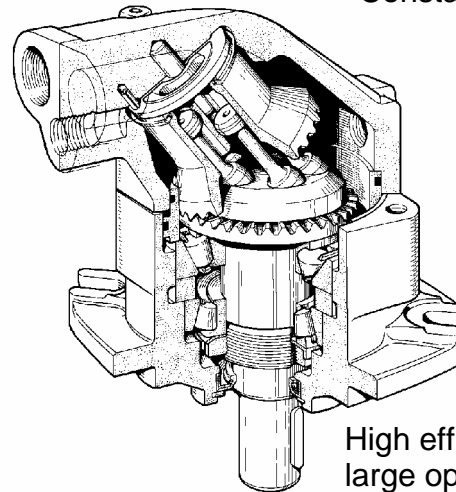
In variable models typically large 8°-25°
control angle (up to 40°)

Up to 400 bar

High efficiency at
large operational
area

https://www.parker.com/literature/Literature%20Files/hydraulicpump/
cat/english/F11-F12_HY17-8249-US.pdf

Hydraulic cylinders

Convert hydraulic power into mechanical power

Construction types:

- single acting
- double acting

Both operate on positive displacement principle

Terminology

plus direction   minus direction

A chamber   plus chamber   minus chamber   B chamber

stroke length

A_1 A_3 A_2

② ③ ④ ⑥ ⑤   ① ⑦   ⑧
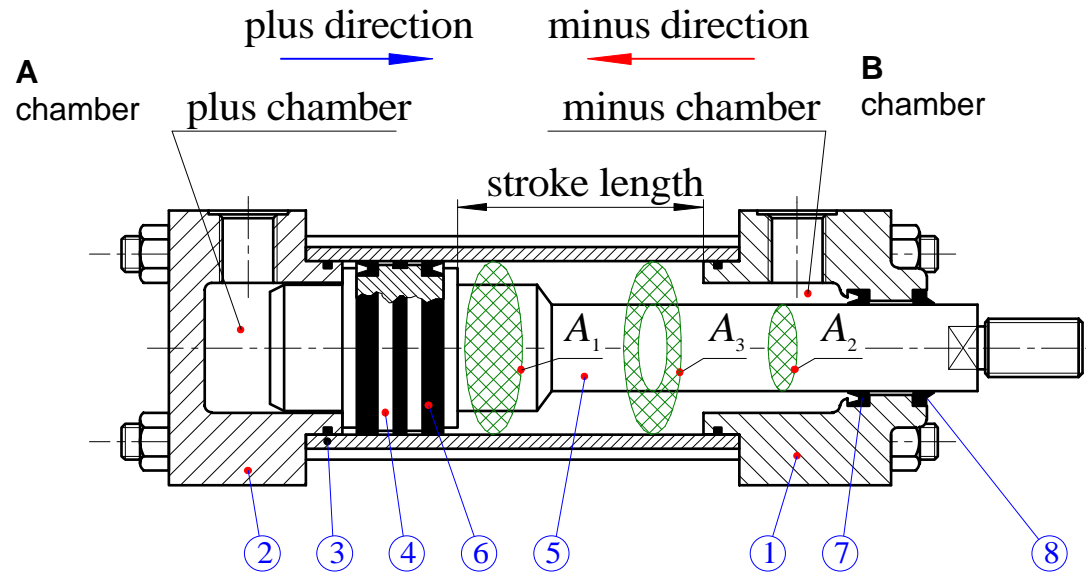
1. Head with inlet   2. Cap with inlet   3. Cylinder tube

4. Piston   5. Piston rod

6. Piston seals   7. Piston rod seal   8. Piston rod wiper seal

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Single acting cylinders

Operate hydraulically to only one direction

Vent port - atmospheric pressure

¬ Piston type cylinder

http://www.rocla.com/en/products/
roclas-humanic2-reach-trucks
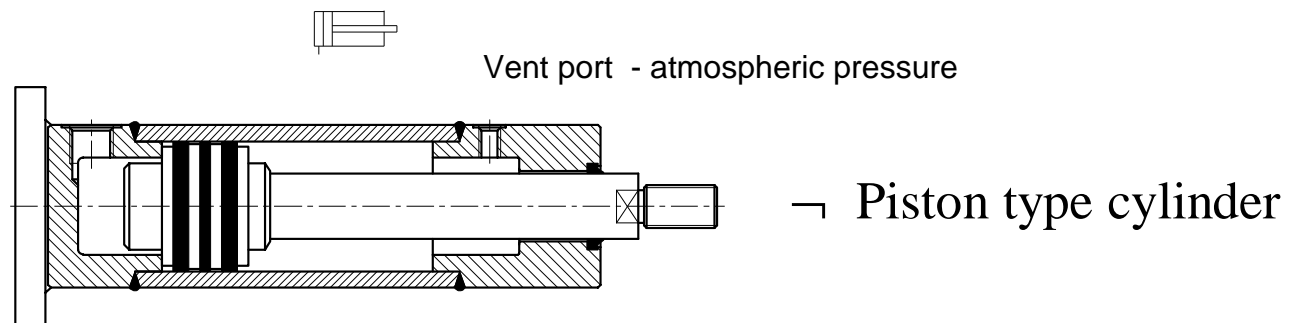
Plunger
cylinder   ®
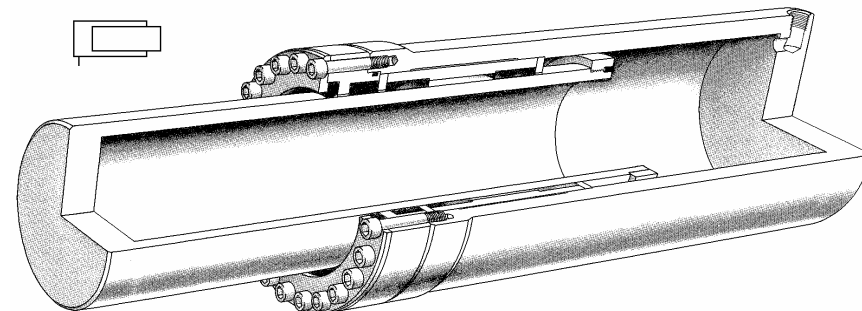
Telescopic cylinder

Variable speed
- the largest piston starts
(the least pressure need)
- the smallest piston is the last
(the highest pressure need)

Used in dump trailers etc.
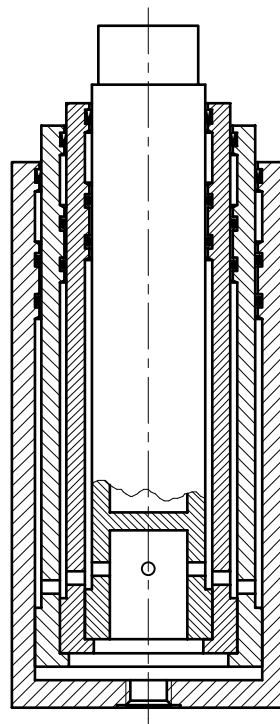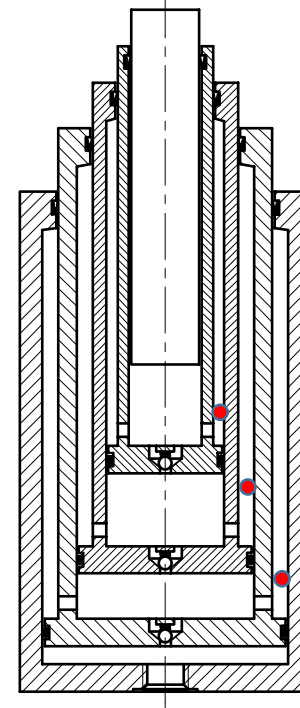
Constant speed telescopic cylinder
All the pistons move at the same time
The movement of a piston forces the fluid from the space between the adjacent cylinders out to lift the next piston

a

b

http://www.nurmihydro.fi/products_akipit.html

Double acting cylinders
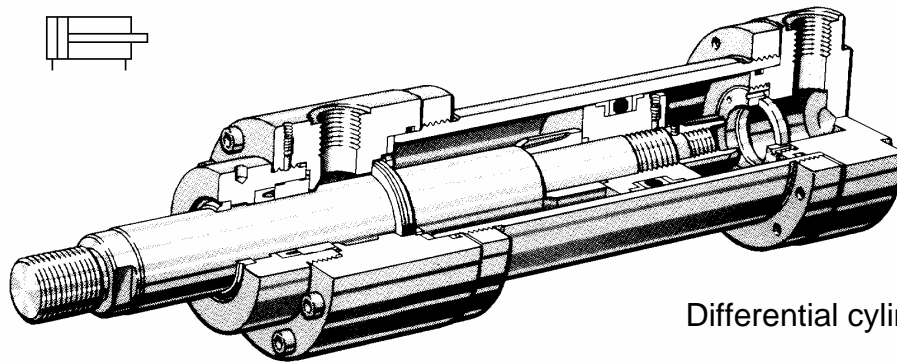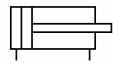
Differential cylinder

¬ Single piston rod

Double piston rod

Symmetric cylinder ®

Telescopic cylinder

Double acting

1st inlet in cap

2nd inlet in rod

Hollow rod

hydraulicpower.fi

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71
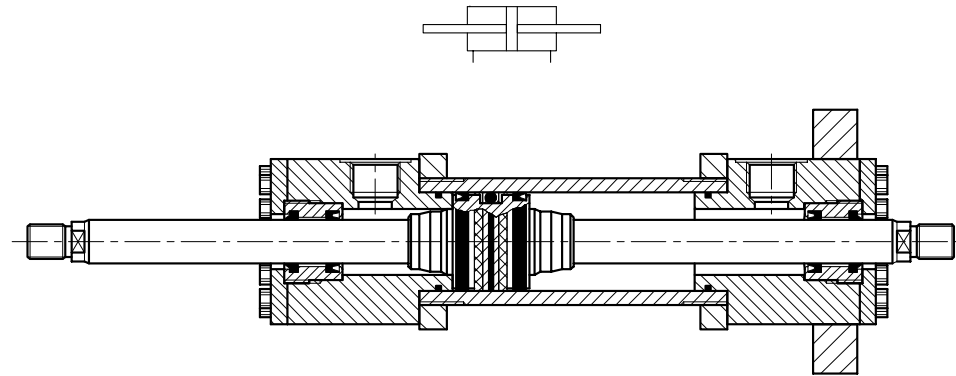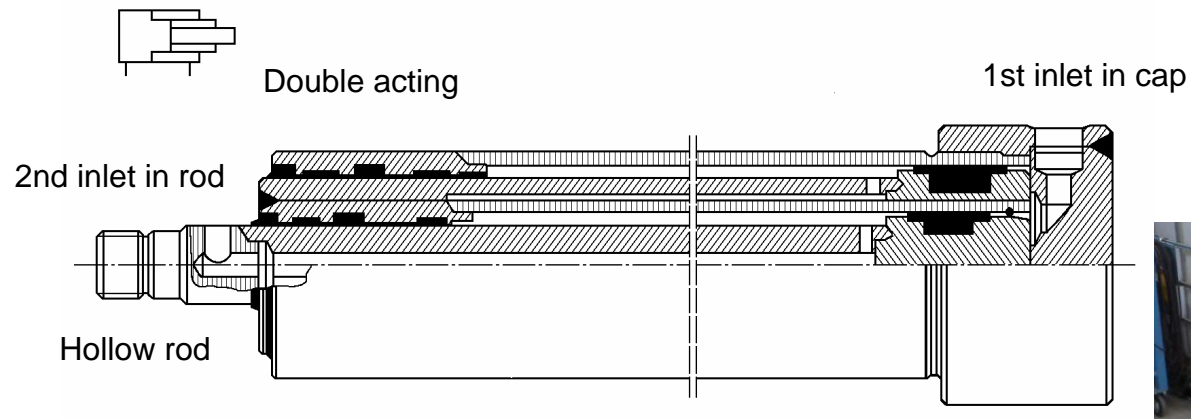
Characteristics of cylinders

Theoretic pressure demand $\qquad p_{\text{theor, in}} \times A_{\text{in}} = F + p_{\text{out}} \times A_{\text{out}}$

Theoretic flow demand $\qquad q_{\text{V,in,theor}} = A_{\text{in}} \times v$

Quasi-static process
Hydraulic variables do not change

A_1 $\qquad$ A_3 $\qquad$ A_2
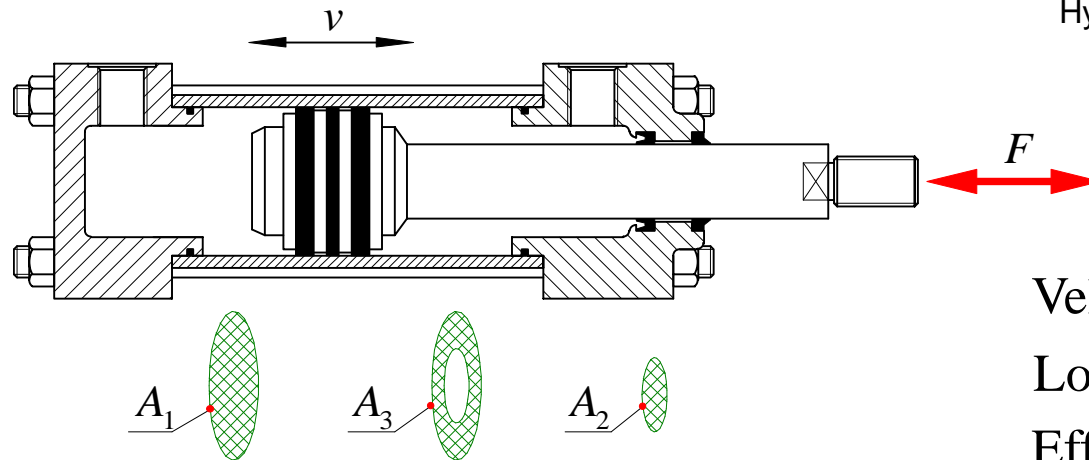
Velocity v [m/s]

Load force F [N]

Effective piston area A [m^2]

Reality

Small minor losses (pressure losses) in inlets
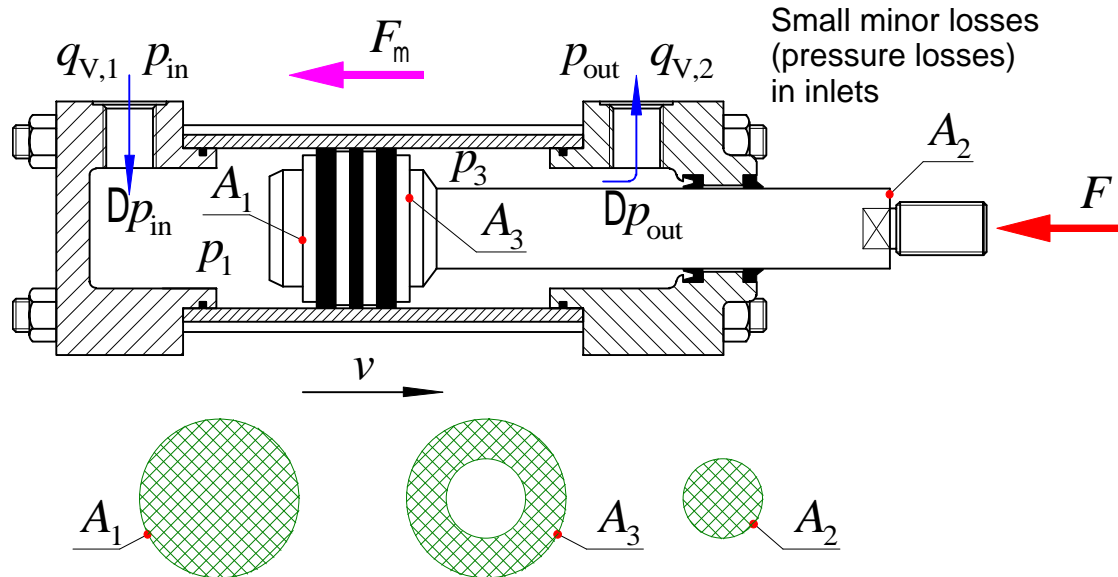
$q_{V,1}$ p_{in} F_m p_{out} $q_{V,2}$ Small minor losses (pressure losses) in inlets

A_2

p_3

Dp_{in} A_1 A_3 Dp_{out} F

p_1

v

A_1 A_3 A_2

Example case:
plus-direction movement, opposing force

Friction – hydromechanical efficiency h_{hm}

Leakage – volumetric efficiency h_v

Efficiencies in cylinder case are quite unclear
It would be better to use in calculations
- Friction forces
- Leakage flow rates
instead

Pressure

Example case:
plus-direction movement,
opposing force
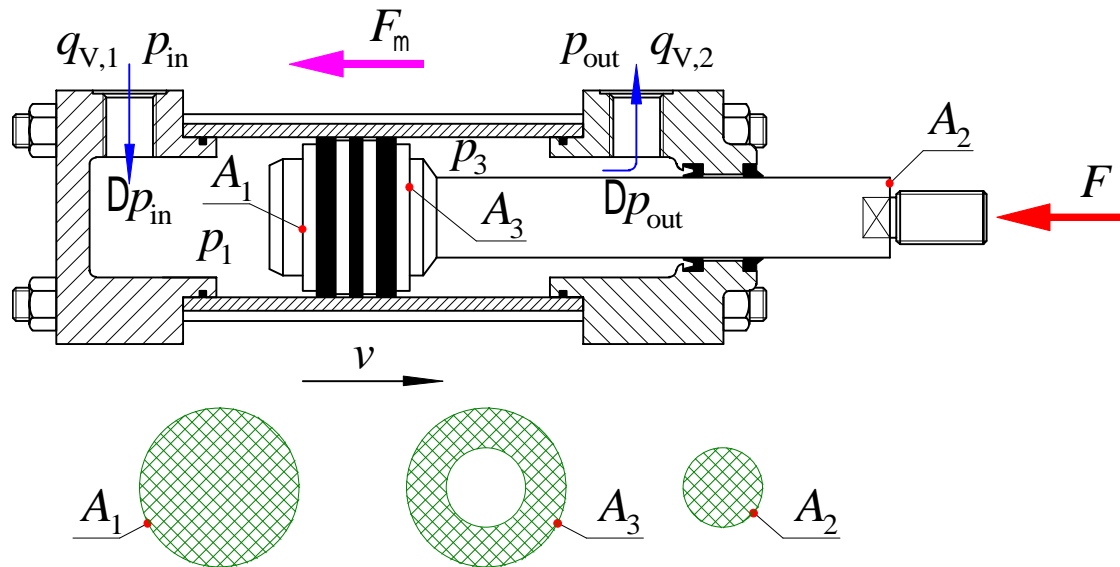
Actual A chamber pressure $\Rightarrow$ force   Actual B chamber pressure $\Rightarrow$ force

Force equation $p_{\text{in}} \times A_1 - Dp_{\text{in}} \times A_1 = p_{\text{out}} \times A_3 + Dp_{\text{out}} \times A_3 + F + F_\mu$ Friction force (opposing movement)

Load force

Force equation in general form

Flow OUT increases the actual chamber pressure because of minor loss

$$p_{\text{real, in}} \times A_{\text{in}} - \text{D}p_{\text{in}} \times A_{\text{in}} = F + F_\mu + p_{\text{out}} \times A_{\text{out}} + \text{D}p_{\text{out}} \times A_{\text{out}}$$

Flow IN decreases the actual chamber pressure because of minor loss

Actual pressure demand

$$p_{\text{real, in}} = \frac{F + F_\mu + p_{\text{out}} \rangle A_{\text{out}} + \text{D}p_{\text{out}} \rangle A_{\text{out}} + \text{D}p_{\text{in}} \rangle A_{\text{in}}}{A_{\text{in}}}$$

In efficiency form

$$p_{\text{real, in}} = \frac{F}{A_{\text{in}} \times h_{\text{hm}}} + p_{\text{out}} \times \frac{A_{\text{out}}}{A_{\text{in}}}$$

Correct or not?
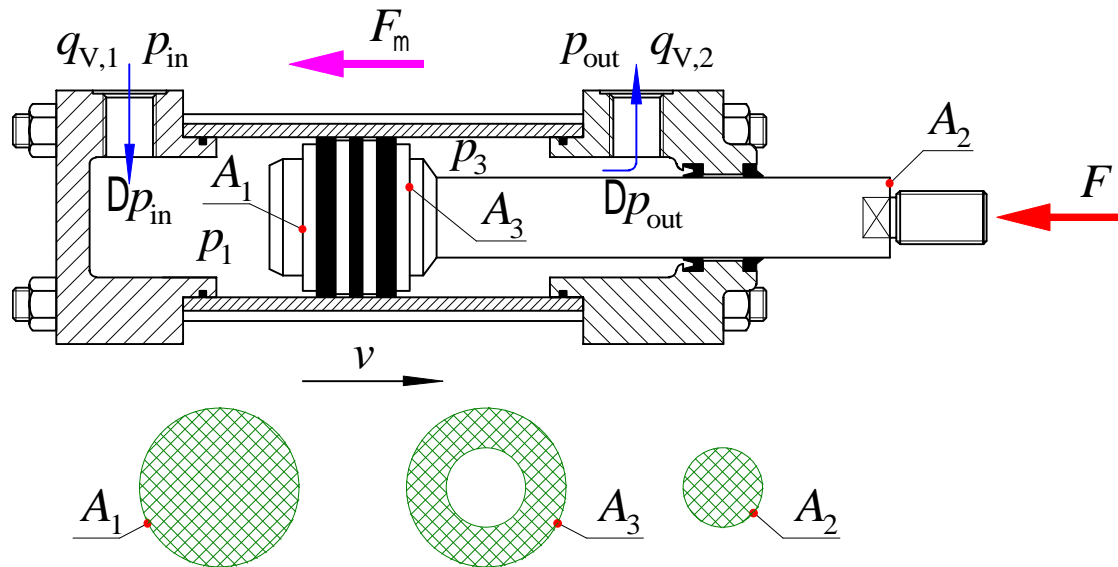Hydromechanical efficiency depends on what?

Flow
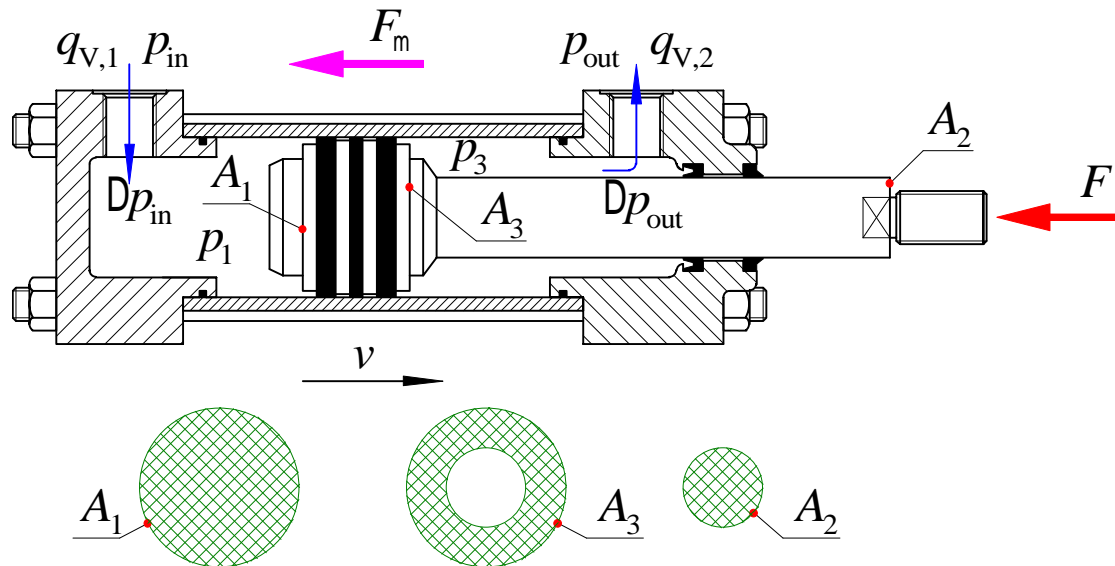
Example case:
plus-direction movement,
opposing force

Actual flow demand $\qquad q_{V,in,real} = \dfrac{A_{in} \cdot v}{h_v}$

Volumetric efficiency depends on the leakages
What if pressure p_3 is higher than p_1 ?
Unclear!

Power

Example case:
plus-direction movement,
opposing force

Power demand of external force $\quad P_{mech} = F \cdot v$
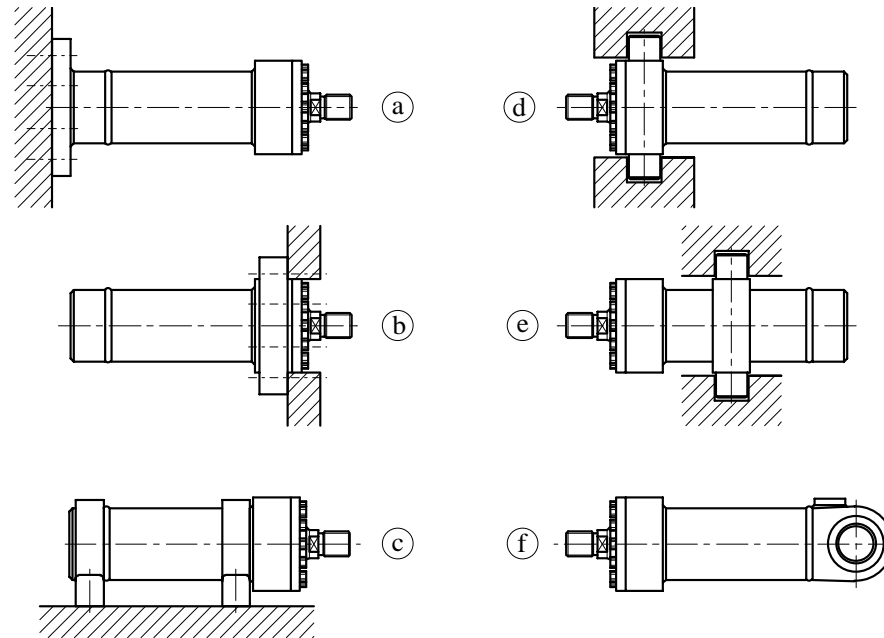
Theoretic power demand

$$P_{\text{theor}} = q_{V,\text{in}} \times \left(p_{\text{in}} - \frac{A_{\text{out}}}{A_{\text{in}}} \times p_{\text{out}} \right) = F \times v$$

Actual power demand

$$P_{\text{real}} = q_{V,\text{in}} \times \left(p_{\text{in}} - \frac{A_{\text{out}}}{A_{\text{in}}} \times p_{\text{out}} \right) = \frac{F \times v}{h_t}$$
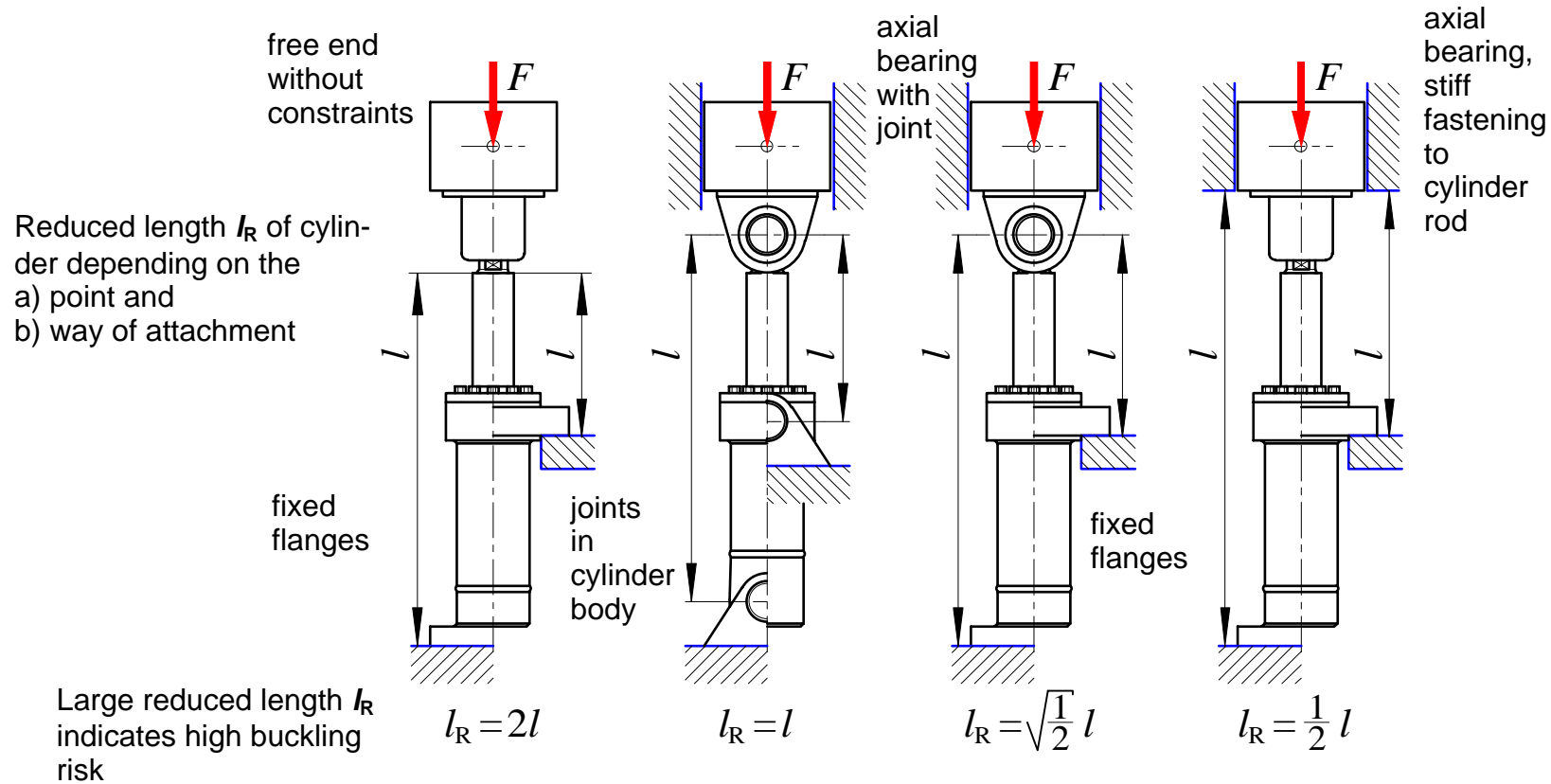
If there is meaningful leakage, both the volumetric efficiency and total efficiency depend heavily on the velocity.
Normally leakage is very small and total efficiency depends merely on seal friction(s) and minor losses.

Loading and buckling of cylinders

Loading only in parallel to the piston rod!

Mounting and buckling length

free end
without
constraints

F

axial
bearing
with
joint

F

F

axial
bearing,
stiff
fastening
to
cylinder
rod

F

Reduced length I_R of cylin-
der depending on the
a) point and
b) way of attachment

l l l l l l l l

fixed
flanges

joints
in
cylinder
body

fixed
flanges

Large reduced length I_R
indicates high buckling
risk

$$l_R = 2l$$

$$l_R = l$$

$$l_R = \sqrt{\tfrac{1}{2}}\, l$$

$$l_R = \tfrac{1}{2}\, l$$

Buckling diagram

Safety factor typically 4

$$F = \frac{p^2 \times E_{\mathrm{m}} \times I}{C_{\mathrm{n}} \times l_{\mathrm{R}}^2}$$

Young's modulus ∼ 210 GPa for steel

E_{m}	modulus of elasticity
I	area moment of inertia $\quad p\, d^4 / 64$
C_{n}	safety factor
l_{R}	effective length

l_{R} [mm]
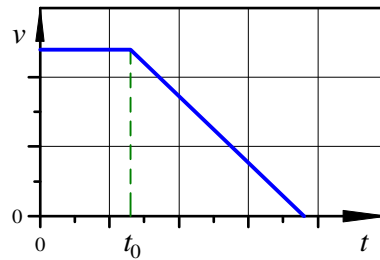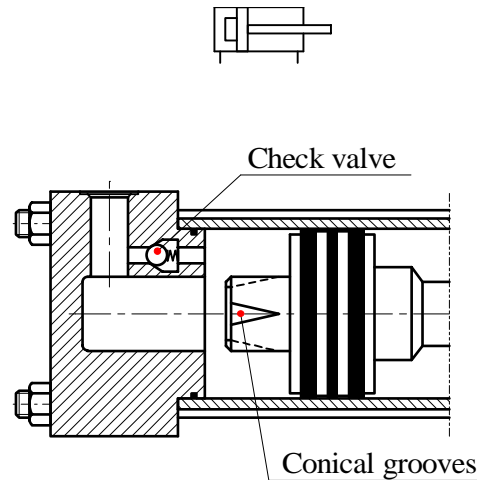
Piston rod diameter
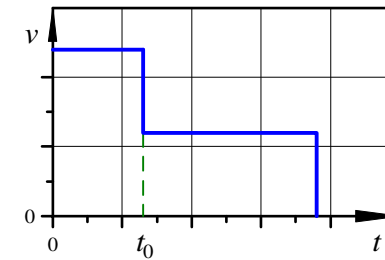
220
180
160
140
125
110
90
70
56
45
36
28

p [MPa]
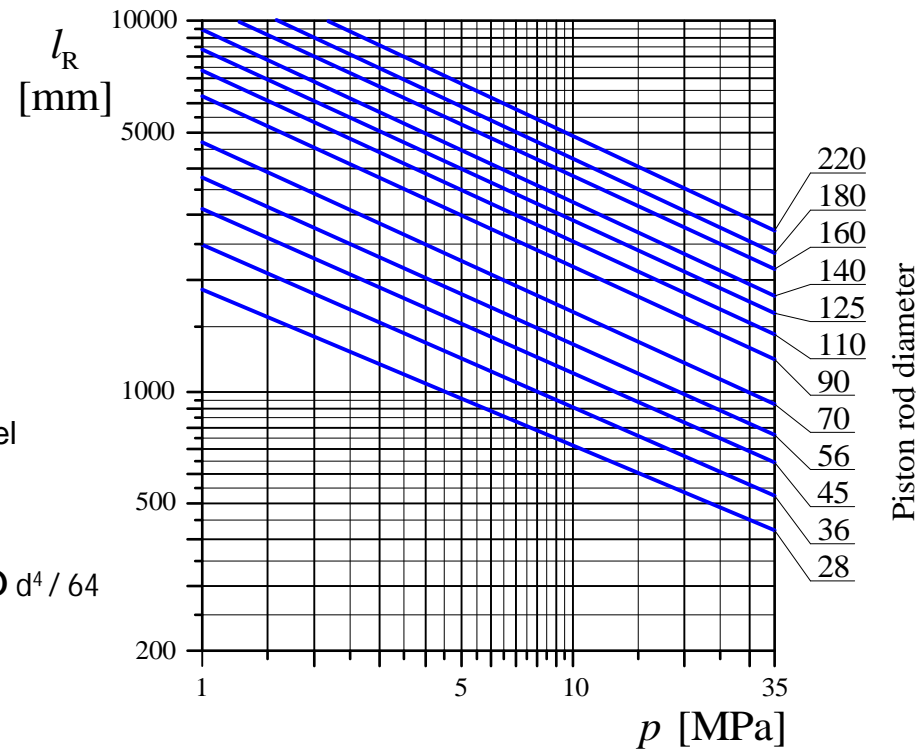
End cushioning of cylinders

At velocities > 0.1 m/s

Flow area diminishes flow resistance increases the deeper the rod end is in the flow channel

Check valve

Check valve

Conical grooves

Throttle
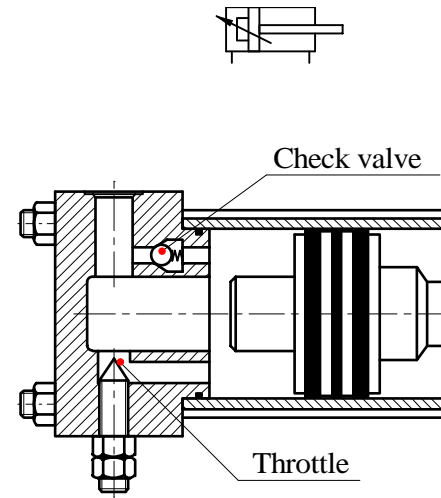
Flow area and flow resistance remains the same as the rod end gets into the flow channel
Flow resistance is adjustable

Torque motors

Convert hydraulic power into mechanical power

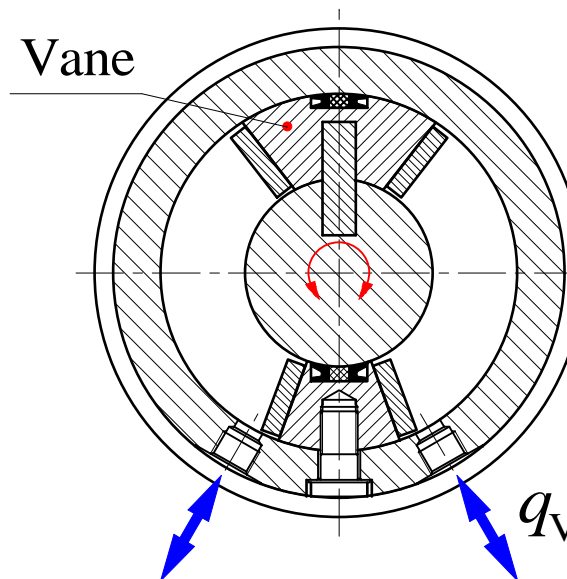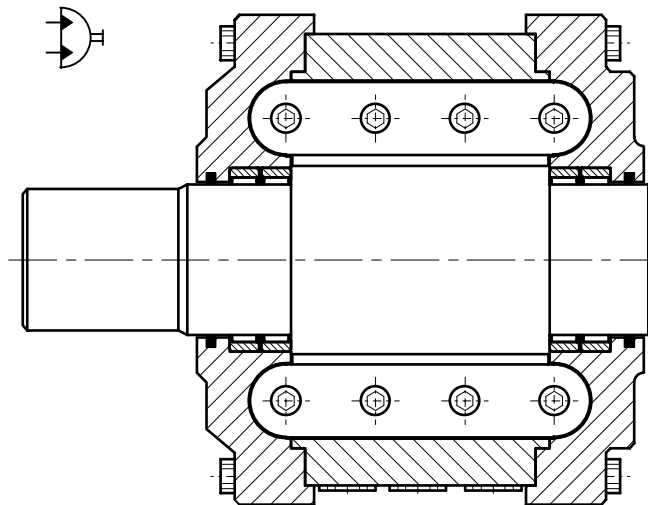Rotation angle restricted, generally $< 360°$

Construction types:
  - vane
  - piston

Both operate on positive displacement principle

Total efficiency max. $h_t » 0.6 – 0.86$

Torque max. $T » 10000 – 20000 (– 300000)$ Nm

Vane type

Vane

q_V

Maximum rotation angle $\sim\!320^{\mathrm{o}}$

With end cushioning $\sim\!240^{\mathrm{o}}$

Uneven pressure distribution causes high bearing loads

With 2 vane structure higher torques and pressure forces can be

compensated $\Longrightarrow$

smaller turning angle $\sim\!170^{\mathrm{o}}$

Constant torque
Leakage reduces total efficiency

Maximum $\sim\!72\%$

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Piston type

Gear type:

Screw gear
Crank mechanism
Cogwheel gear

Linear movement produced by piston
is transformed into rotational by using
two (2) screw gear mechanisms
- On the outer surface of piston
- On the inner surface of piston
The rotation angle can be up to 720°

Maximum pressure is about 100 bars
Efficiency is limited by leakages and
friction in the gear
Constant torque
Maximum efficiency is about 62%
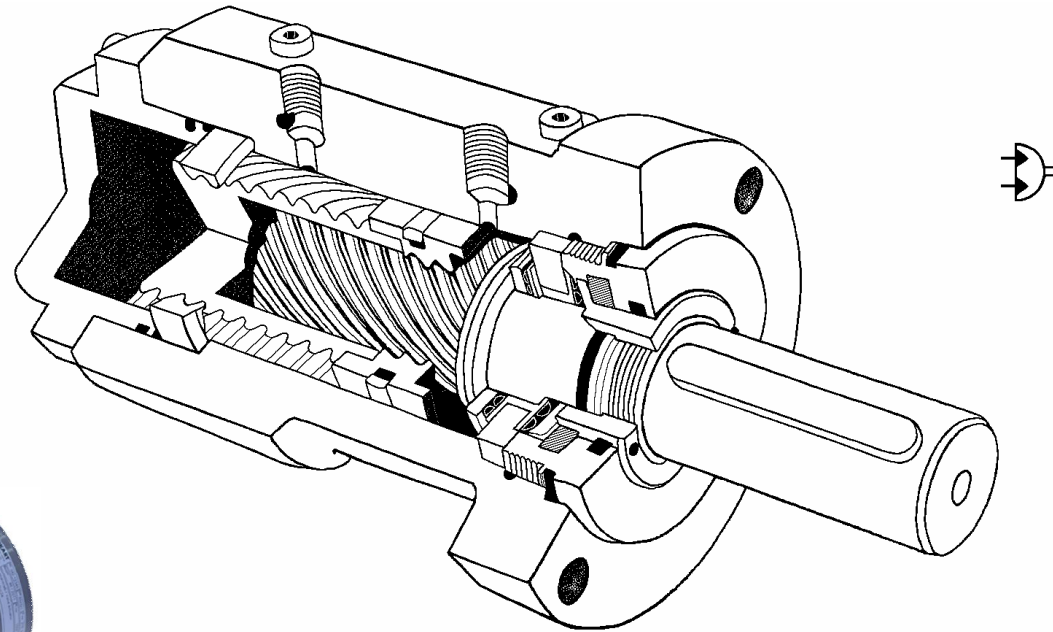
Screw gear
Crank mechanism
Cogwheel gear

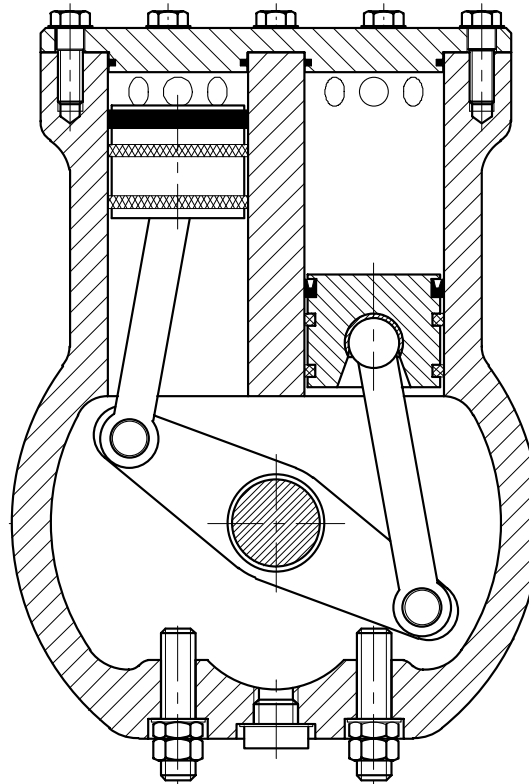Two pistons in parallel to have operation in two directions
Torque is not constant
- maximum in the middle
- only about 2/3 of the maximum near the ends
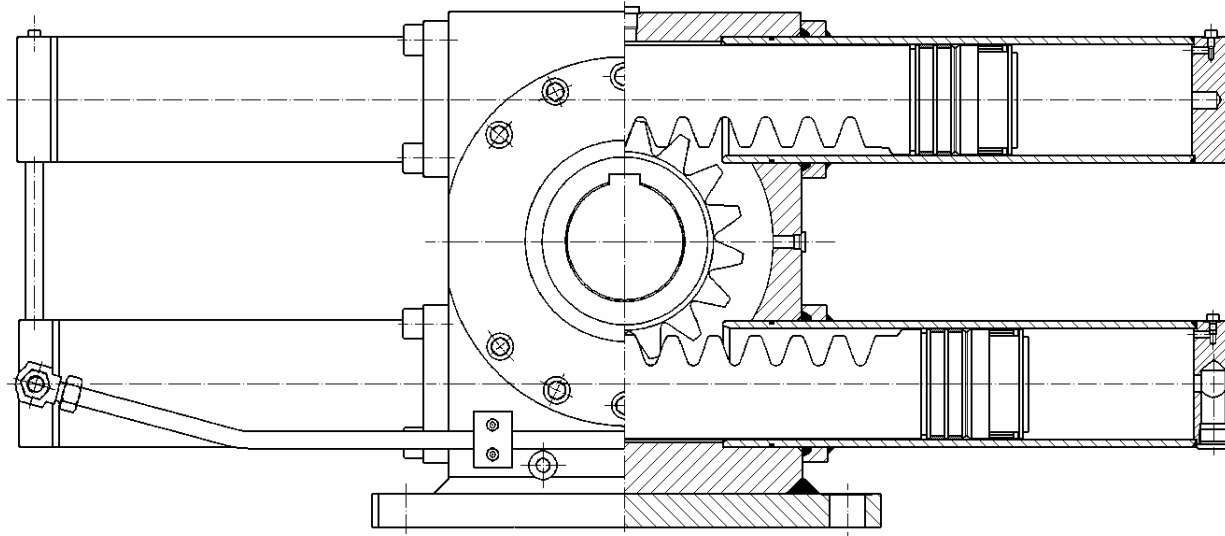
Turning angle about 100°

Maximum pressures 100 - 150 bar

Torque motors

Screw gear
Crank mechanism
Cogwheel gear

Linear movement is converted into rotational with cogwheel mechanism
The rotation angles are typically 90° - 720°
Maximum pressure typically 160 bar
Efficiency can be about 86%
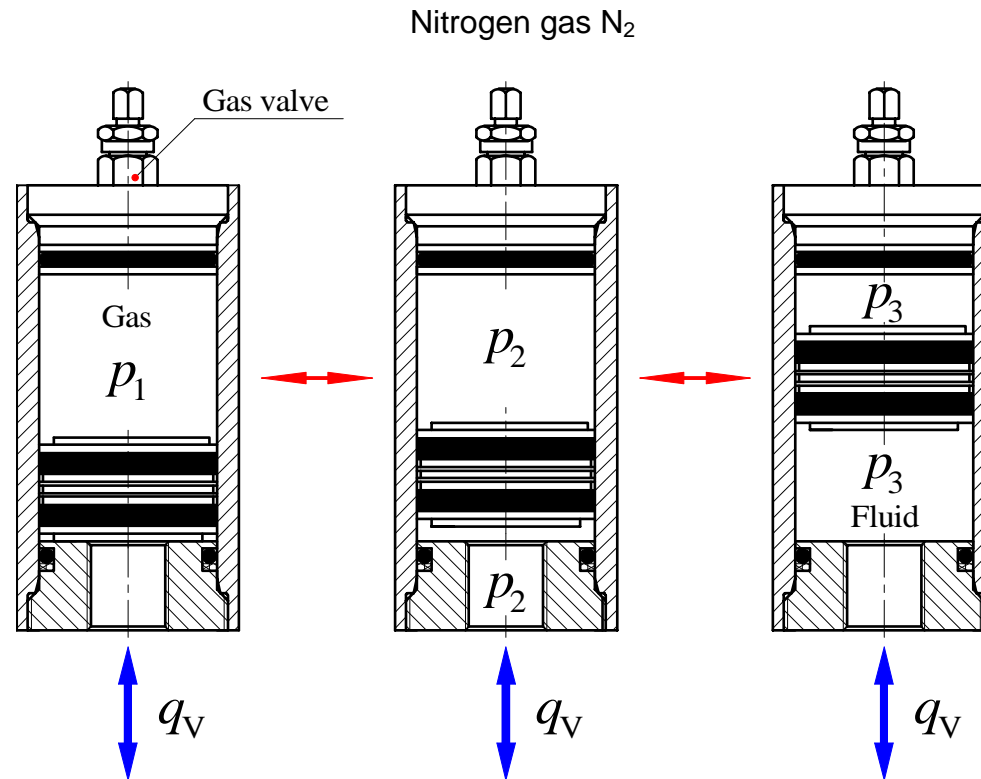
Rack and pinion structure

Torque motors

https://www.fiellberg.fi/products/torque-motor/

Screw gear
Crank mechanism
Cogwheel gear

Pressure accumulators

Nitrogen gas N_2

Store hydraulic energy
by utilizing the
compressibility of gas
(nitrogen)

Piston accumulator

To activate the accumulator the system
pressure has to exceed accumulator's
precharge pressure (p_1 in the figure))

Gas valve

Gas

p_1

p_2

p_2

p_3

p_3

Fluid

q_V

q_V

q_V

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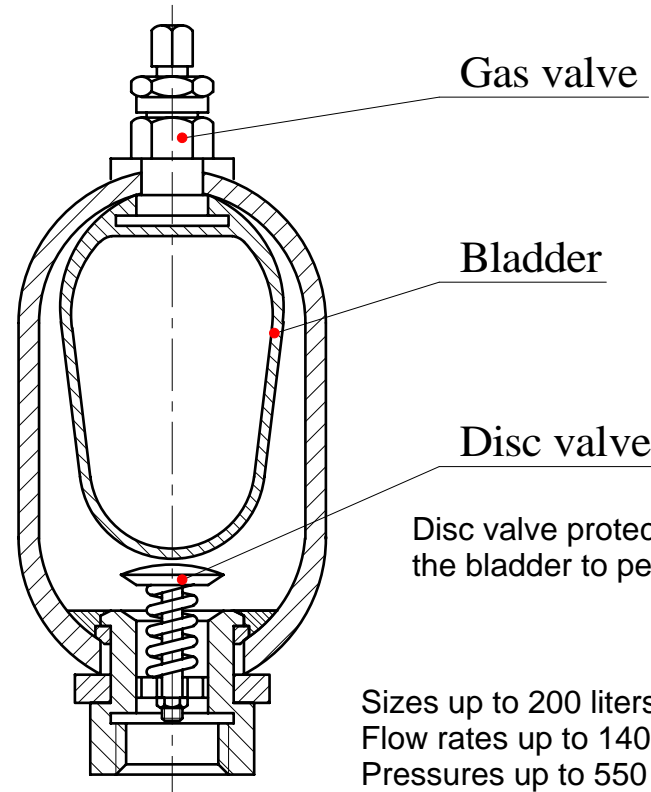# Construction and characteristics

Construction types:

- bladder
- diagphram
- piston

Nominal volumes $V \approx 0.1 - 600 \times 10^{-3}$ m^3

Charging and de-charging flows max. $q_V \approx 120 - 140 \times 10^{-3}$ m^3/s

Gas inside the bladder
Fluid outside

Gas valve

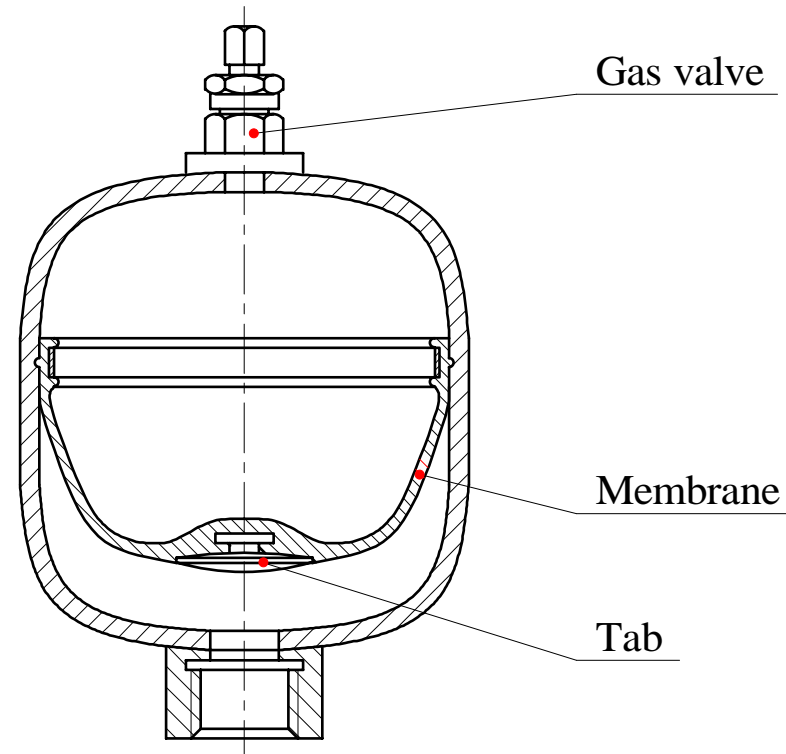Bladder

Disc valve

Disc valve protects the bladder and prevents
the bladder to penetrate out

Bladder
Diagphram
Piston

Sizes up to 200 liters
Flow rates up to 140 l/s (!)
Pressures up to 550 bar
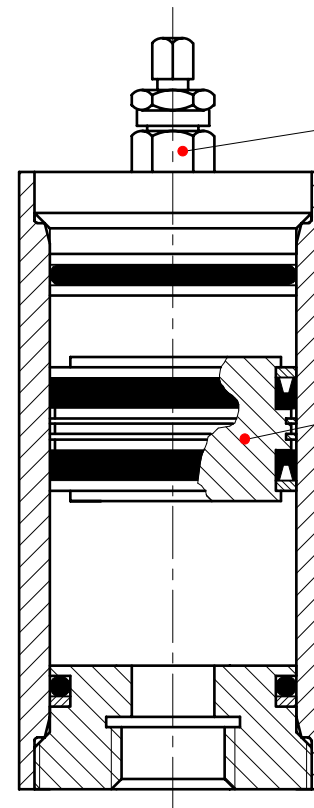
Membrane accumulators are
often used for damping of
hydraulic oscillations
Sizes up to 5 liters
Flow rates up to 140 l/s (!)
Pressures up to 500 bar

Bladder
Diagphram (aka Membrane)
Piston

Gas valve

Membrane

Tab

https://www.hydroll.com/fi/

Bladder
Diagphram
Piston

Gas valve

Piston

Sizes up to 600 liters
Flow rates up to 120 l/s (!)
Pressures up to 400 bar

Seal friction may decrease the
efficiency (at low pressures)

Gas volume can be increased by
connecting additional gas bottles to
accumulators

Application examples

Flow source

Upkeep of pressure

Levelling of flow fluctuation

Suppression of pressure shocks

Energy storage for exceptional situations

Storing of external energy

Levelling of volume changes
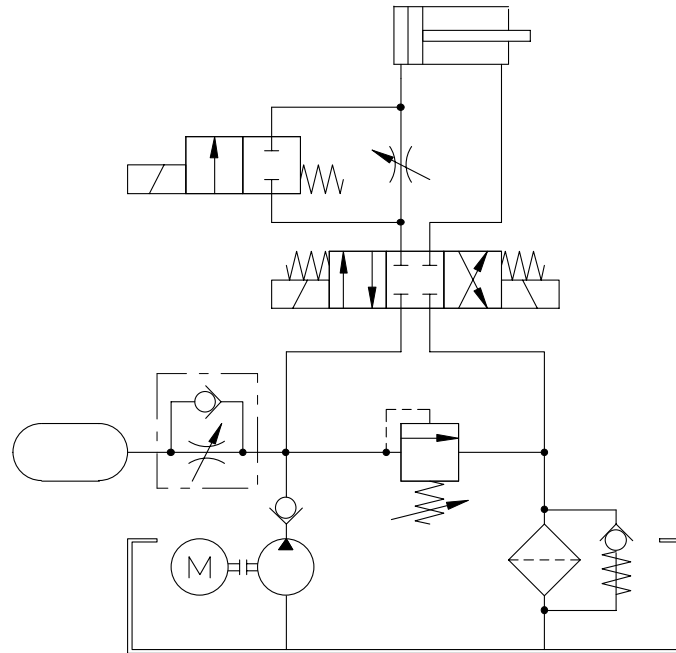
Levelling of shock-like loadings of actuators

compression - storage (1 minute) - expansion

p [Pa]

t [s]

Equation of state for ideal gas can be used for simple calculations
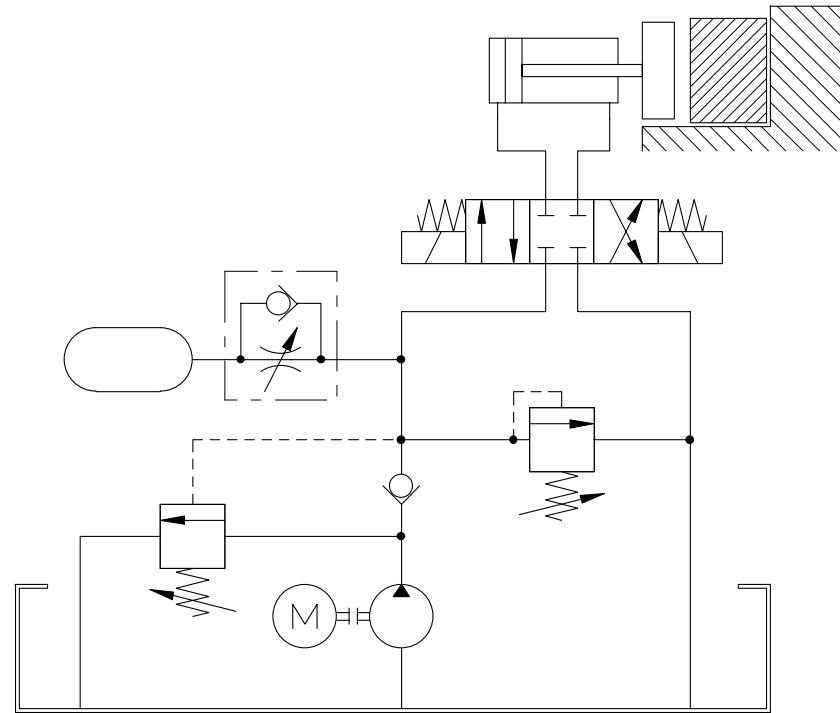Real gas equations for accurate results (especially at high pressures)
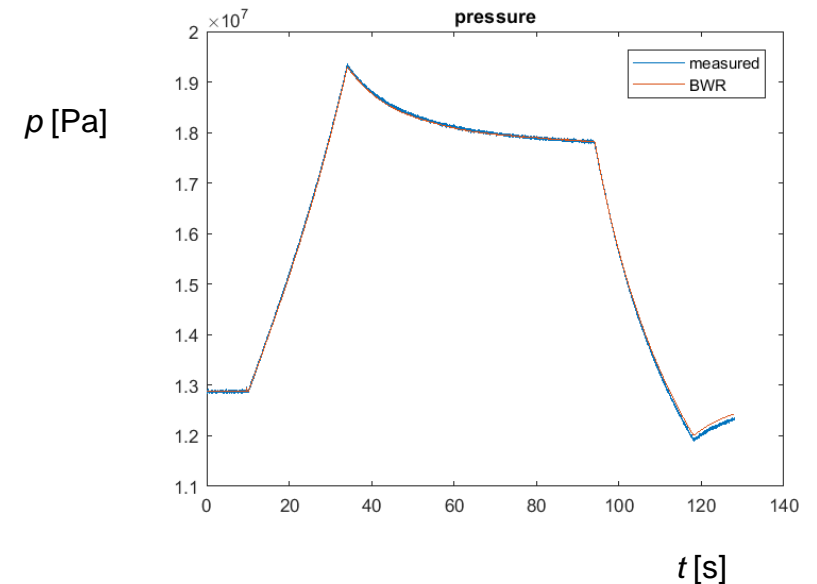
Measured and simulated pressure response indicates the effects of thermal losses especially during storage phase

Flow source

Upkeep of pressure
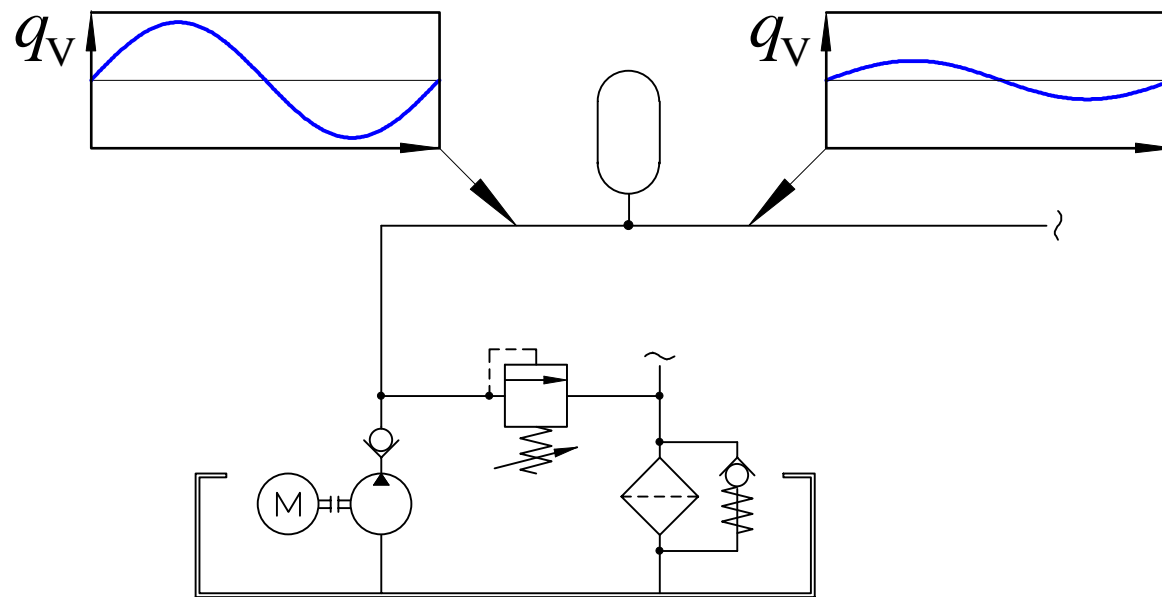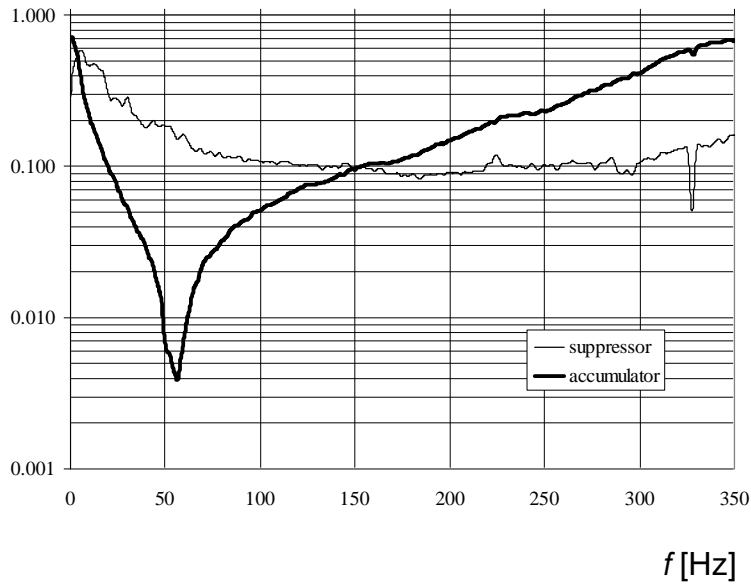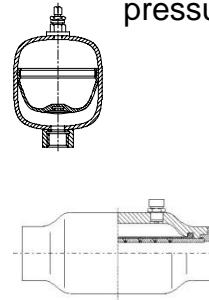
Levelling of flow fluctuation
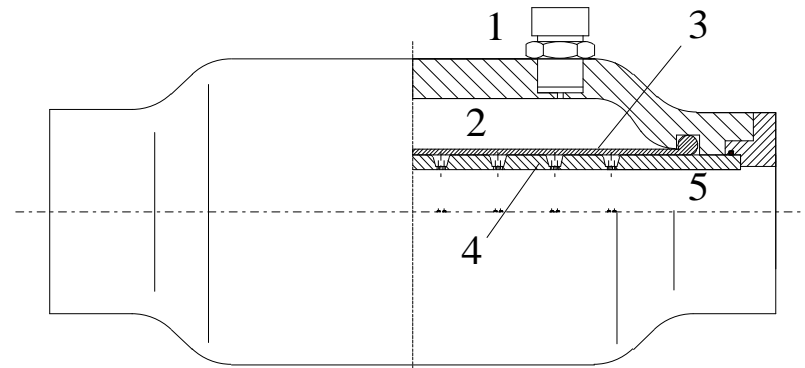
Membrane accumulator's
a) elastic gas volume (hydraulic capacitance) and
b) inlet throat
form together a hydraulic resonator capable of damping
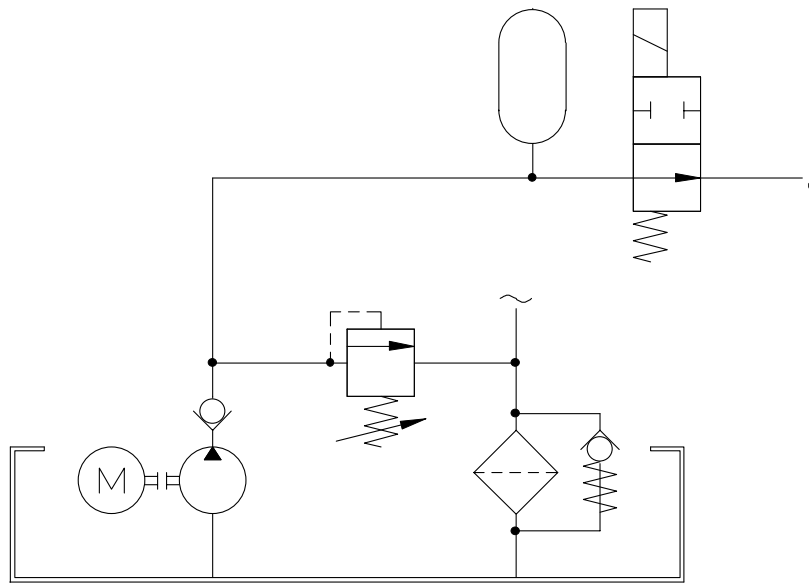pressure oscillations at frequencies 0 - 100 - 300 Hz

Pressure pulsation
amplitude as function
of frequency

Example:

Levelling of
flow fluctuation

Special accumulators "pulsation and noise dampers" or "inline
suppressors" have wider frequency range

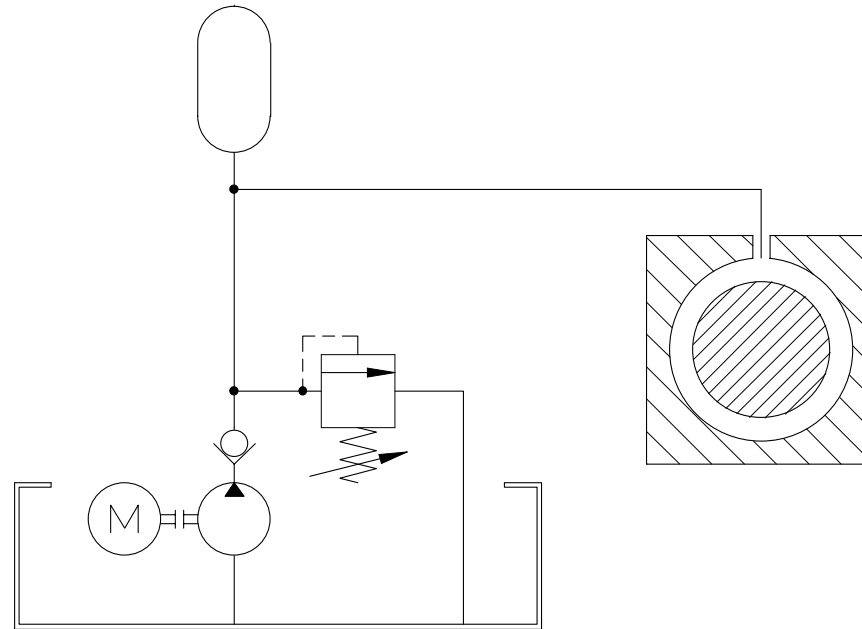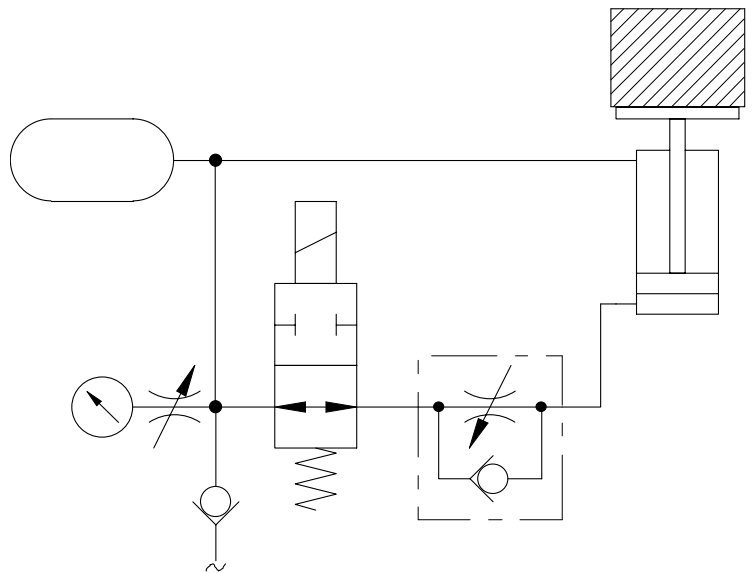Inline suppressor   ®

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Suppression of pressure shocks

Enery storage for exceptional situations

Storing of external energy

Lecture themes - Recap

Pump's task in hydraulic system?

Converting hydraulic power into
- rotational movement?
- linear movement?

Operation principle of hydrostatic power converters?

Utilization possibilities of pressure accumulators?