Aalto University
School of Engineering

MEC-E5003

FLUID POWER BASICS

Study Year 2020

Pumps
Actuators
Accumulators

Aalto University
School of Engineering
Mechanical Engineering / Engineering Design / Mechatronics / Fluid Power

Lecture themes

Flow to the system – How?

Making use of the hydraulic power – How?

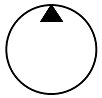
Storing energy in hydraulic system – Why and is that even possible?

Hydrostatic pumps

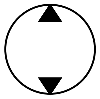
Convert mechanical power into hydraulic power

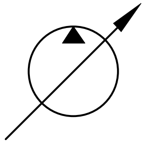
Hydrostatic pumps produce <u>flow</u>, not pressure

Unidirectional | Bidirectional

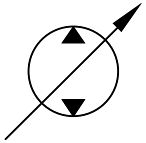


Constant displacement





Variable displacement



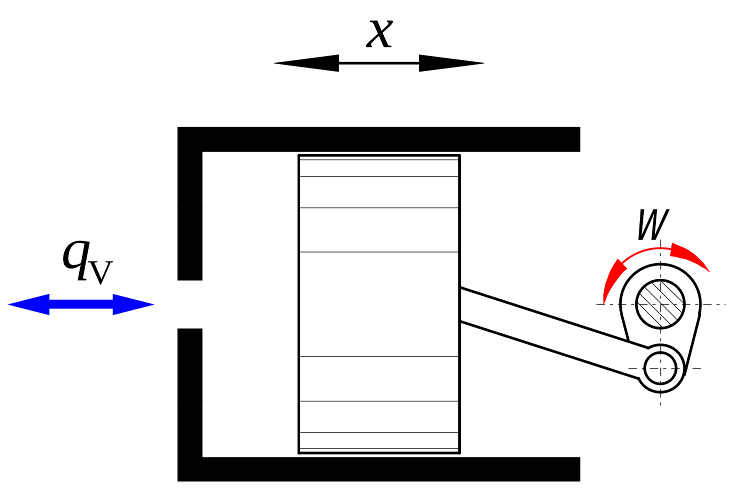
Construction

Most common construction types:

- gear
- vane
- screw
- piston

All operate on positive displacement principle

Positive displacement principle



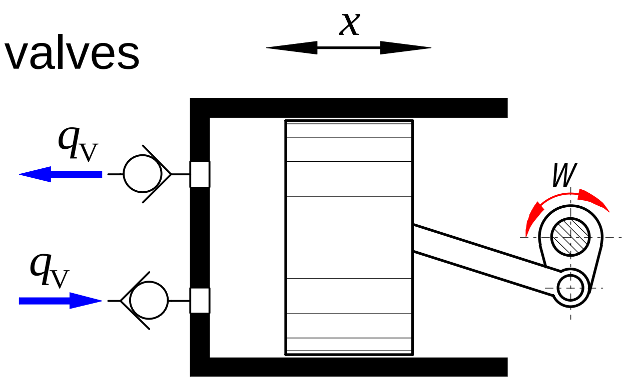
Operating phases:

Fluid flows into transfer volume – suction phase

Fluid flows out from volume – pressure phase

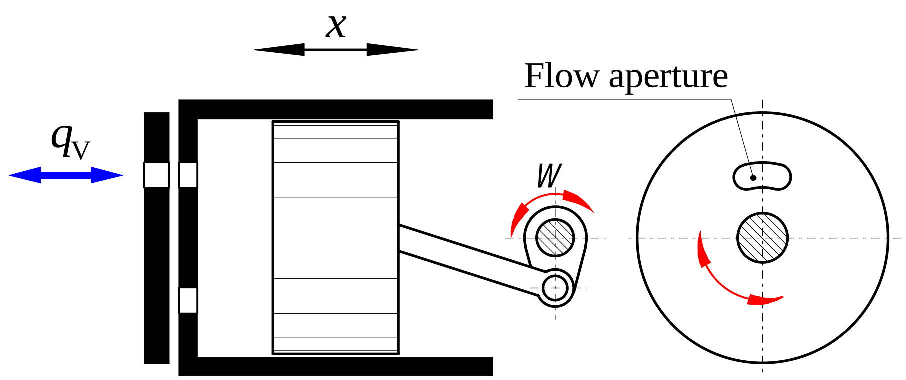
Control of flow direction

Check valves

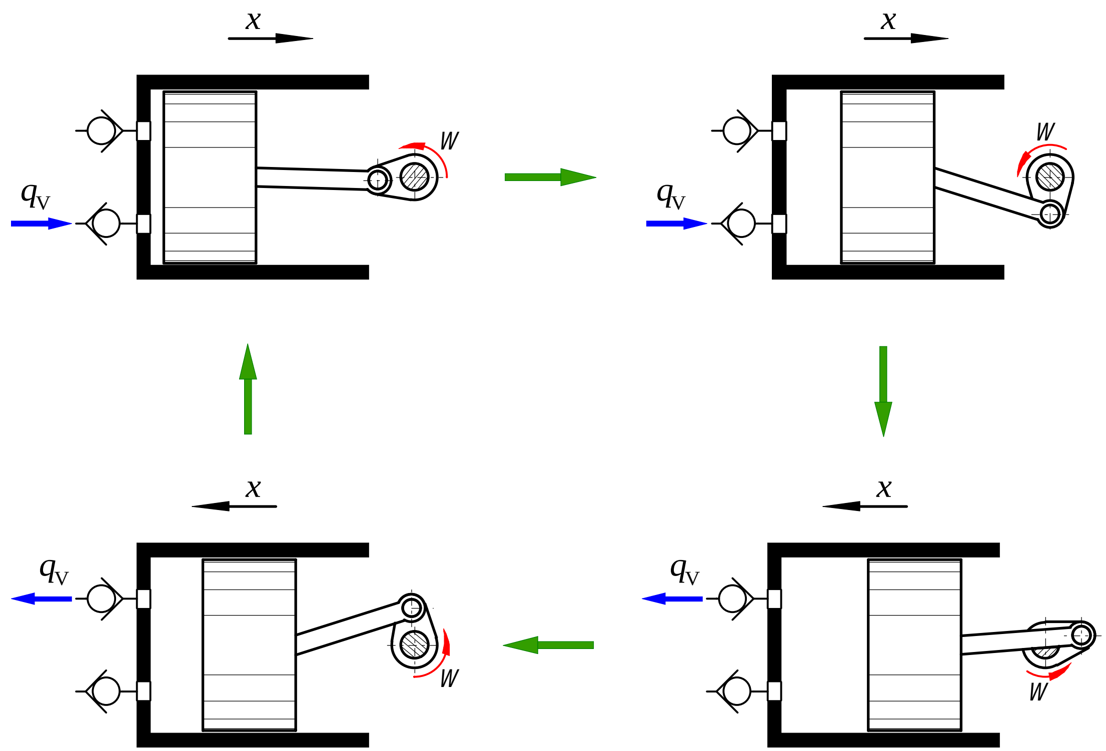


¬ Pressure control
aka valve control

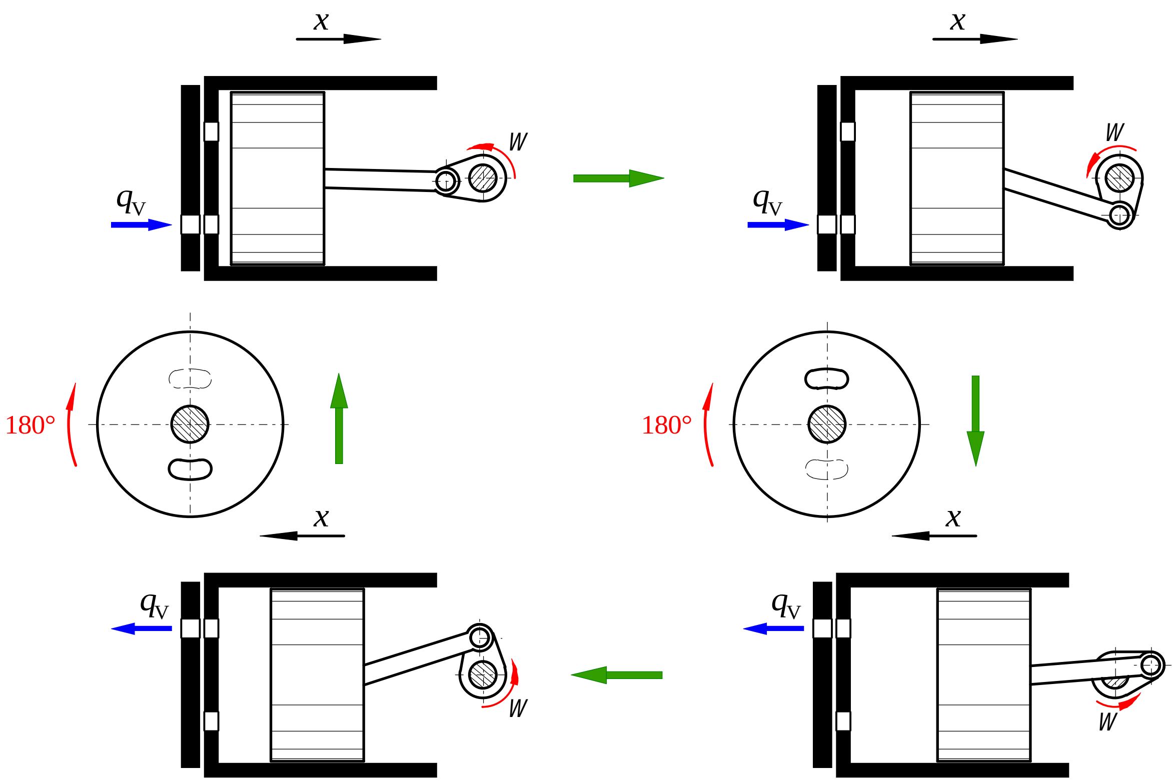
Forced control ®



Pressure control aka valve control

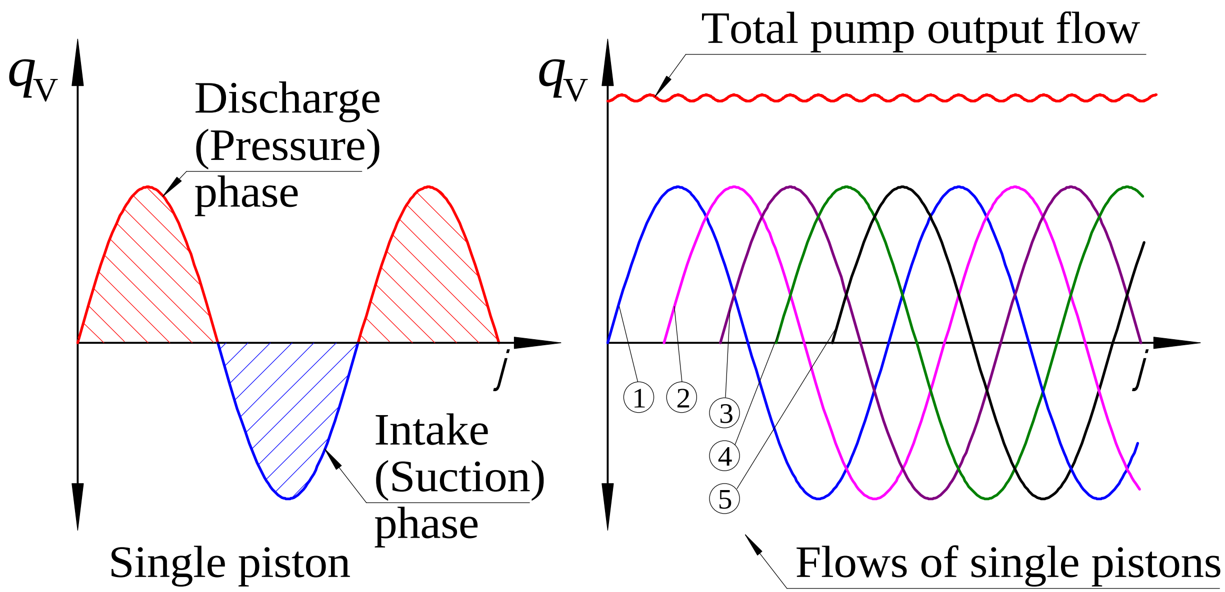


Forced control



Flow pulsation

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



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 $$q_{\mathrm{V,theor}} = n \cdot V_{\mathrm{g}}$$

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

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

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

$$\omega = 2\pi \cdot n$$

Angular velocity ω [rad/s]

$$V_{\mathrm{rad}} = \frac{V_{\mathrm{g}}}{2\pi}$$

Swept volume per radian V_{rad} [m^3/rad]

Effective output flow

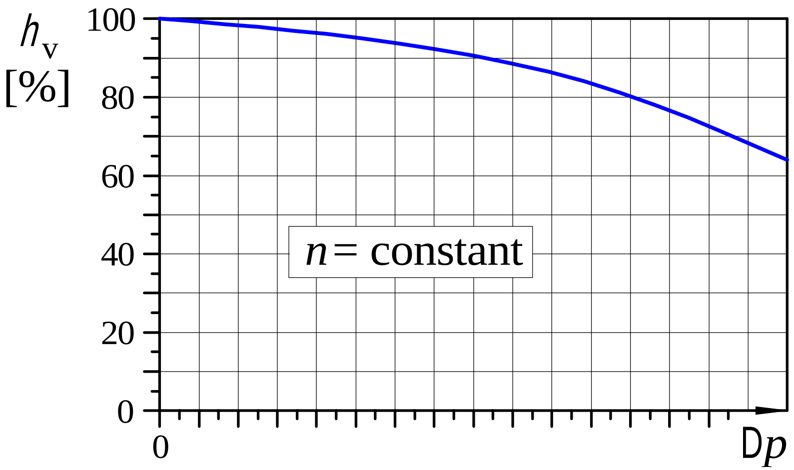
$$q_{V,\mathrm{real}} = n \cdot V_g \cdot \eta_v$$

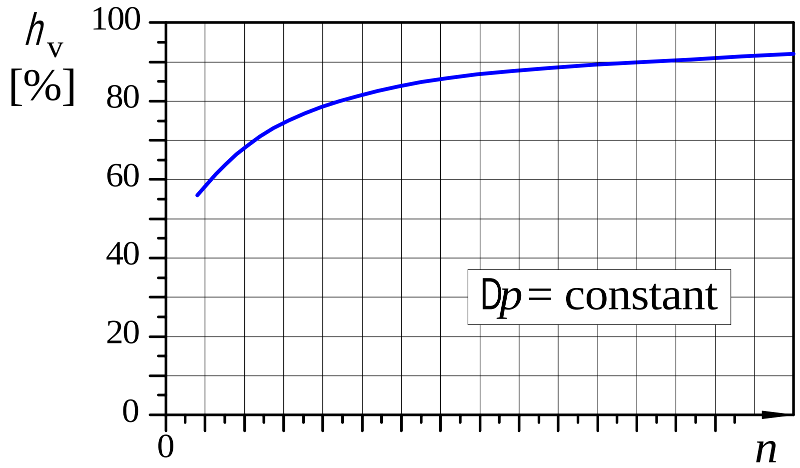
Leakage – volumetric efficiency η_v

Wilson's pump model

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

$\nu\rho$ dynamic viscosity

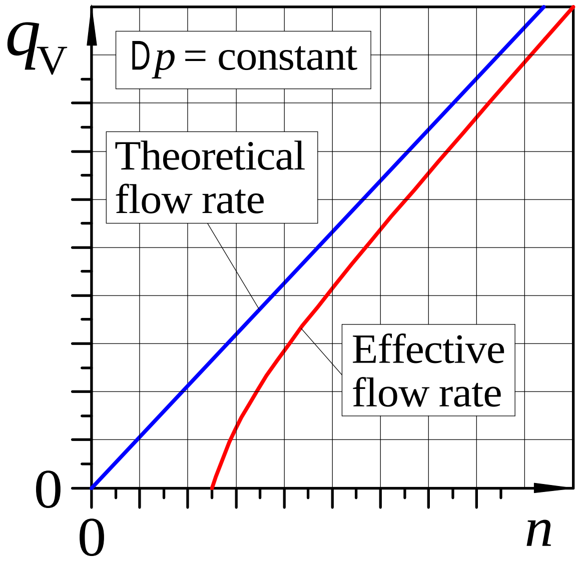


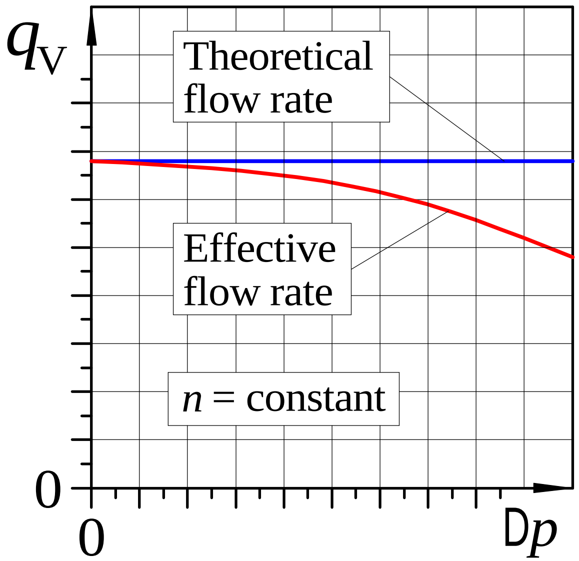


Wilson's pump model

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

Theoretical output flow – Effective output flow





ε pump angle set value (0 - 1)

V_i max. displacement (per revolution)

n rotational speed (1/s)

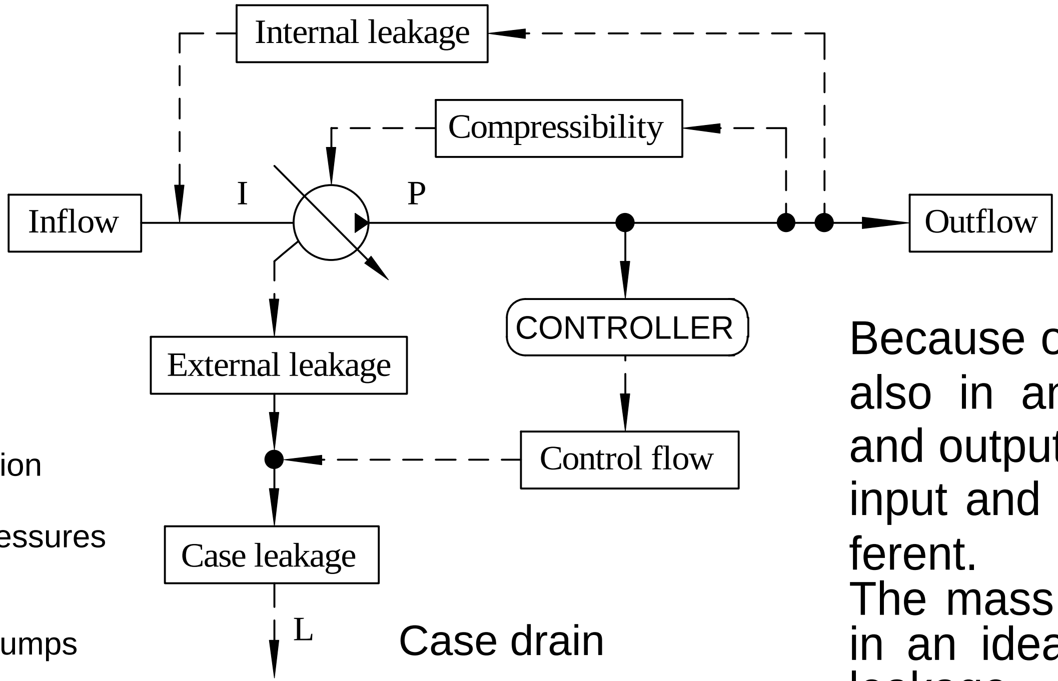
C_s laminar flow loss coefficient

Dp pressure difference over pump

ν fluid kinematic viscosity

ρ fluid density

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 $$T_{\text{theor}} = \frac{\Delta p \cdot V_{\text{g}}}{2 \cdot \pi}$$

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

Pressure difference Δp [N/m^2]

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)

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

Ideal pump and motor

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

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

Reference:

Ellmann, A., Kauranne, H. Kajaste, J. & Pietola, M.

EFFECT OF PARAMETER UNCERTAINTY ON RELIABILITY OF HYDRAULIC TRANSMISSION SYSTEM SIMULATION

Proceedings of IMECE2005 2005 ASME International Mechanical Engineering Congress and Exposition November 5-11, 2005, Orlando, Florida USA

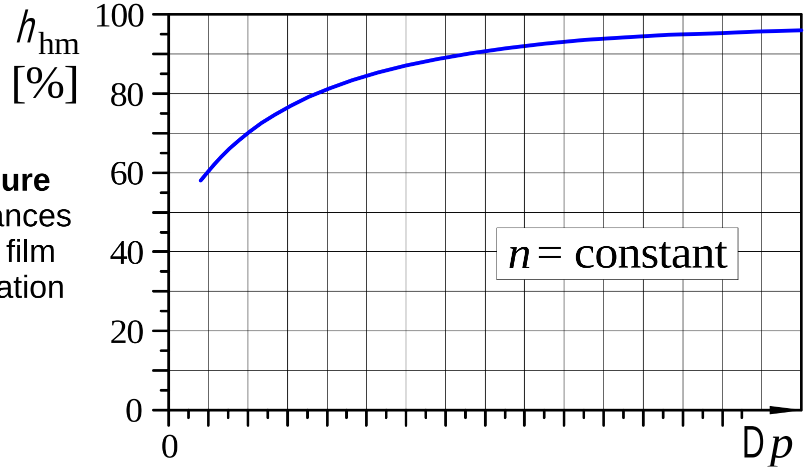
Effective drive torque $$T_{real} = \frac{\Delta p \cdot V_g}{2 \cdot \pi \cdot \eta_{hm}}$$

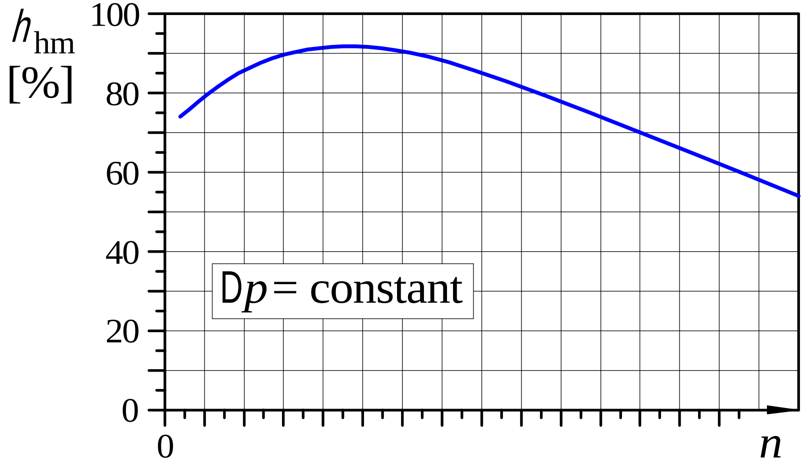
Wilson's pump model does not explain the phenomena seen in the figures below well

Friction – hydromechanical efficiency η_{hm}

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

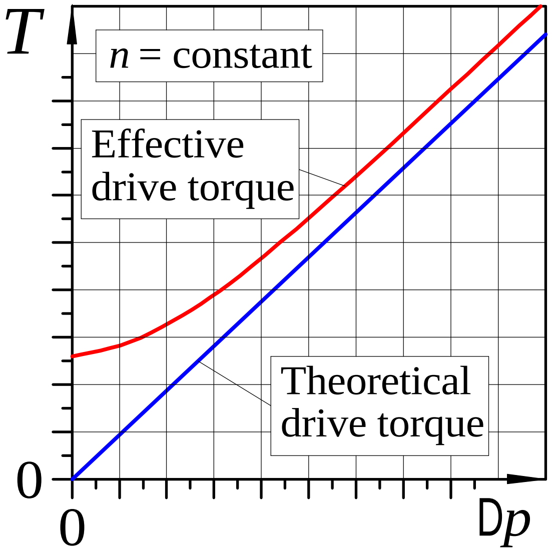
Higher pressure
- Wider clearances
- Thicker fluid film
- Better lubrication

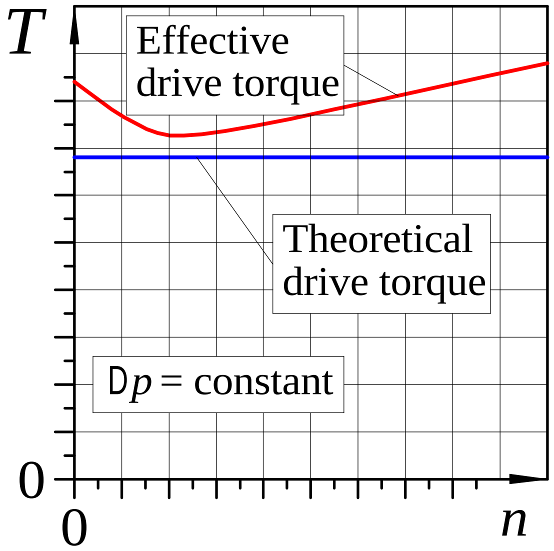




Effect of *n*
- At first Coulomb friction dominates
- Then better lubrication
- Finally hydraulic losses increase

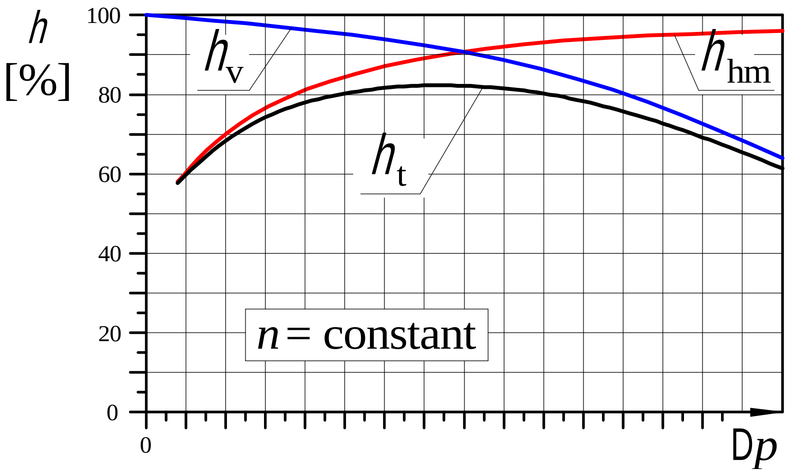
Theoretical drive torque – Effective drive torque

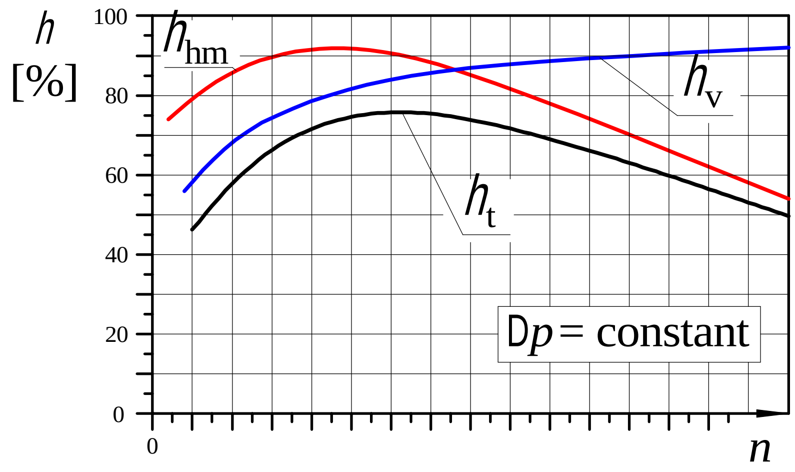




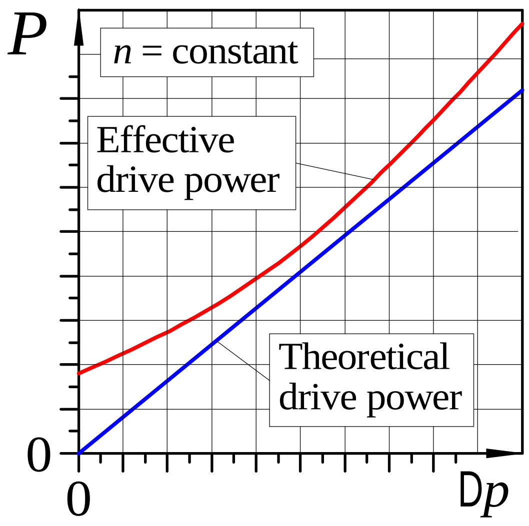
Theoretic drive power $$P_{\text{theor}} = q_V \cdot \Delta p$$

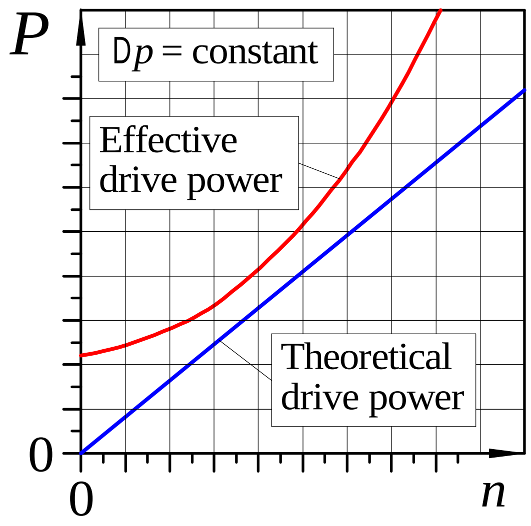
Effective drive power $$P_{\text{real}} = \frac{q_V \cdot \Delta p}{\eta_t}$$ $$\eta_t = \eta_v \cdot \eta_{hm}$$



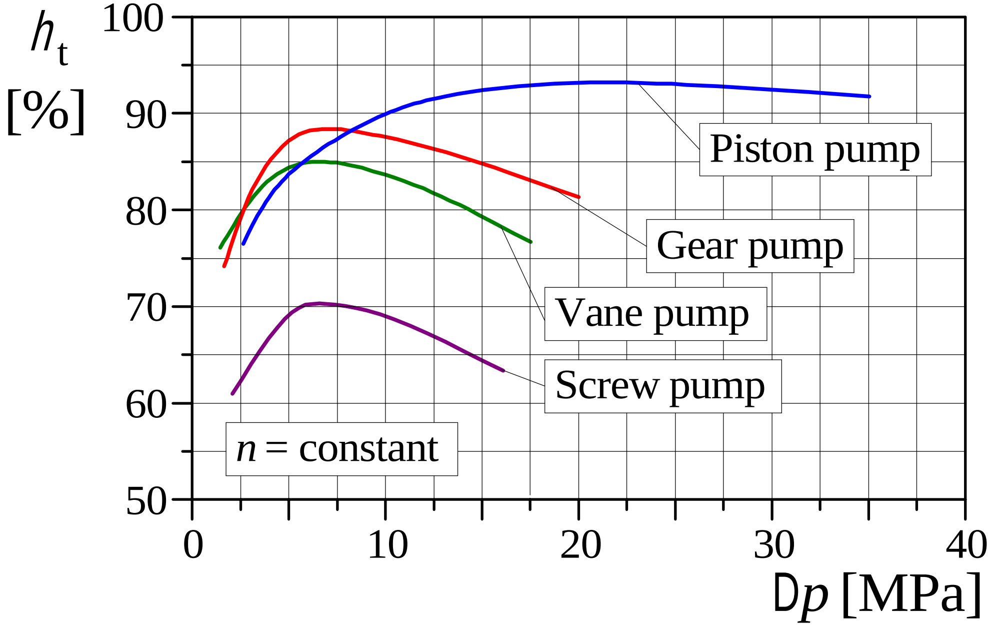


Theoretical drive power – Effective drive power

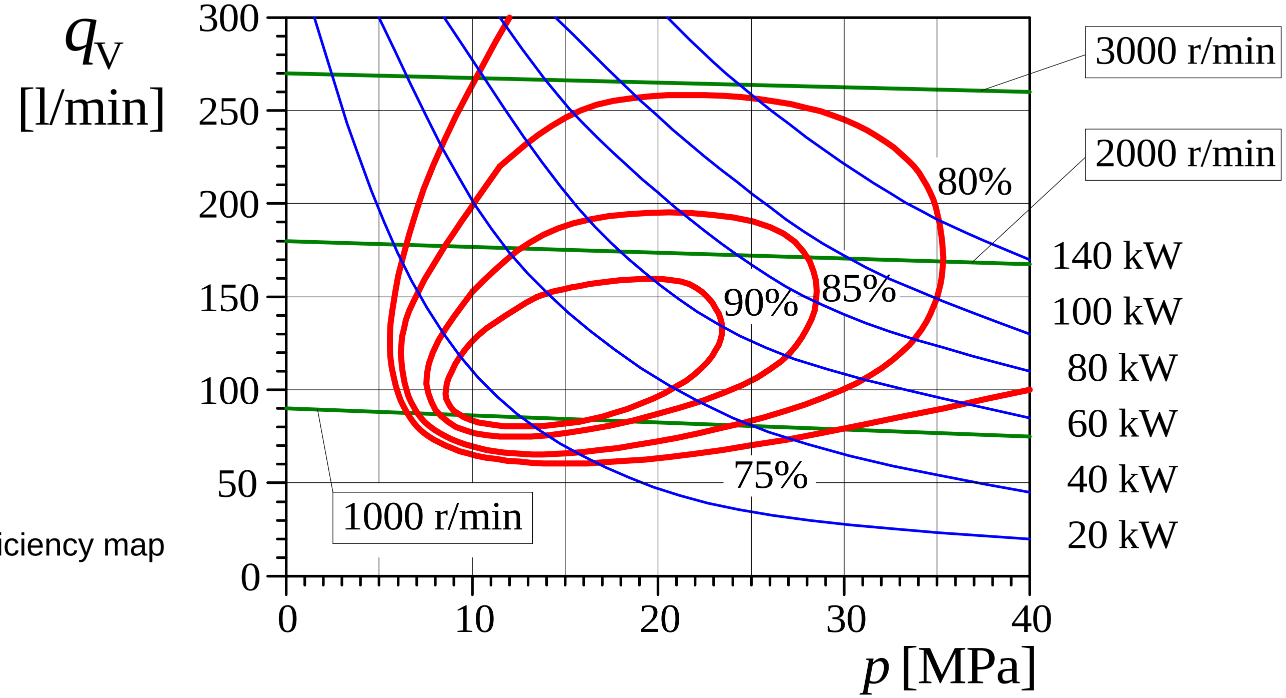




Comparison of structure types

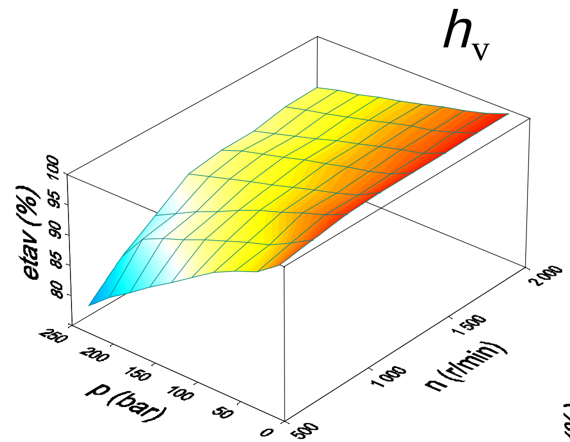


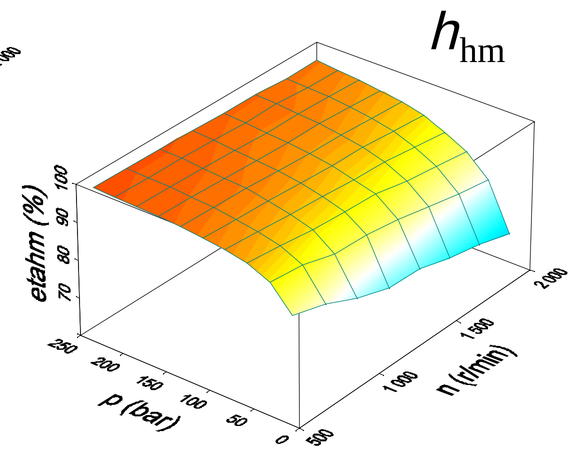
Characteristic curves of pump



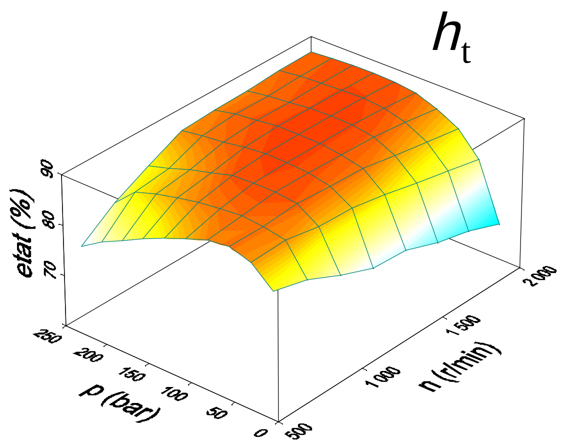
Input power

Efficiency map





Example:
Pressure-rotational speed-dependency of axial piston pump



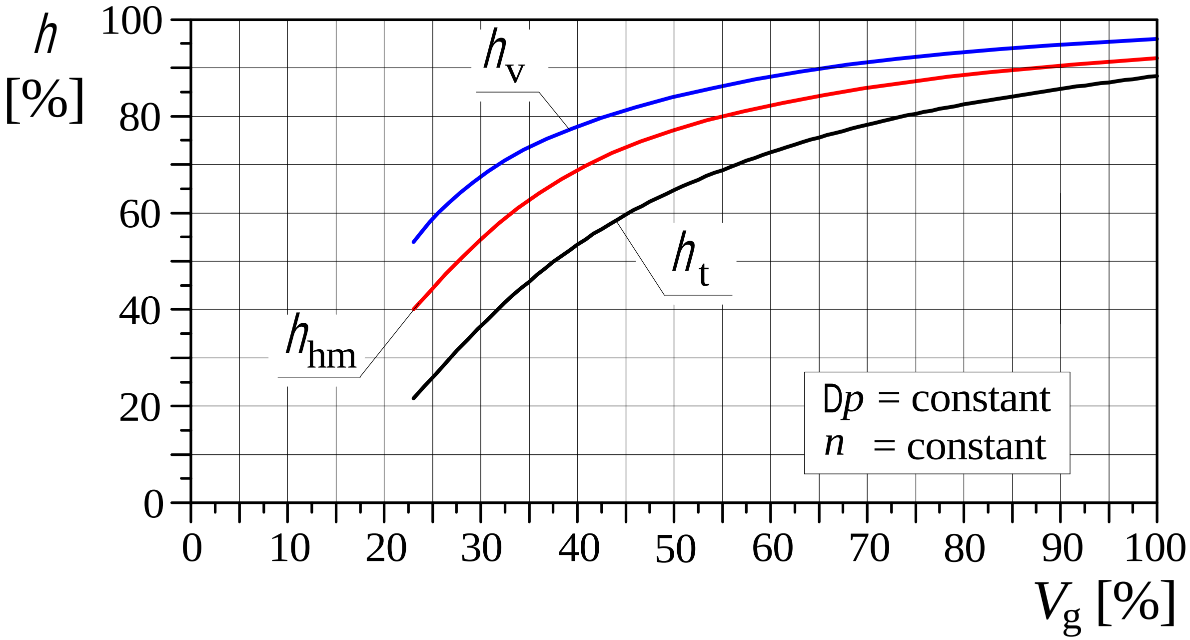
$$T = \varepsilon \frac{V_i \Delta p}{2\pi} + C_f \frac{V_i \Delta p}{2\pi} + C_v V_i n \nu \rho + T_c$$

⇐ Wilson's model (check the effect of decreasing ε)

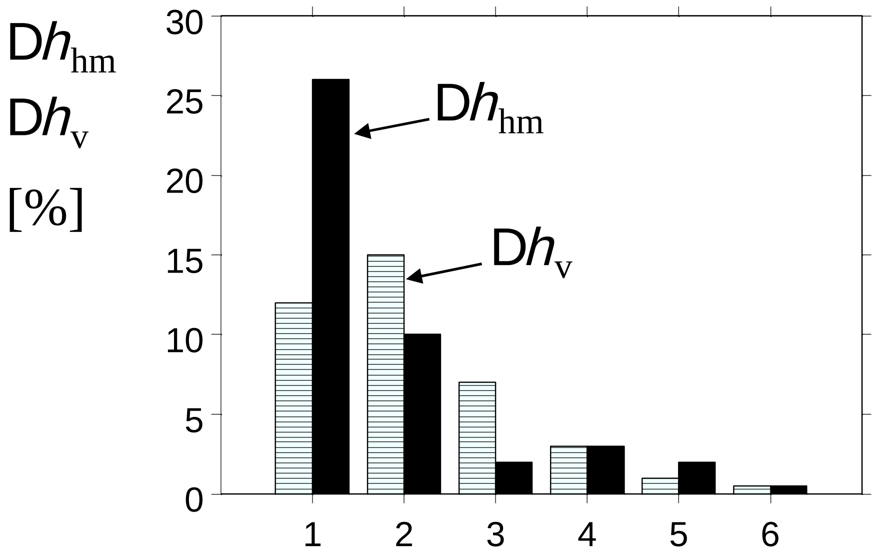
ε Pump angle set value (0 - 1) (axial piston pumps)

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

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



Factors affecting efficiency



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

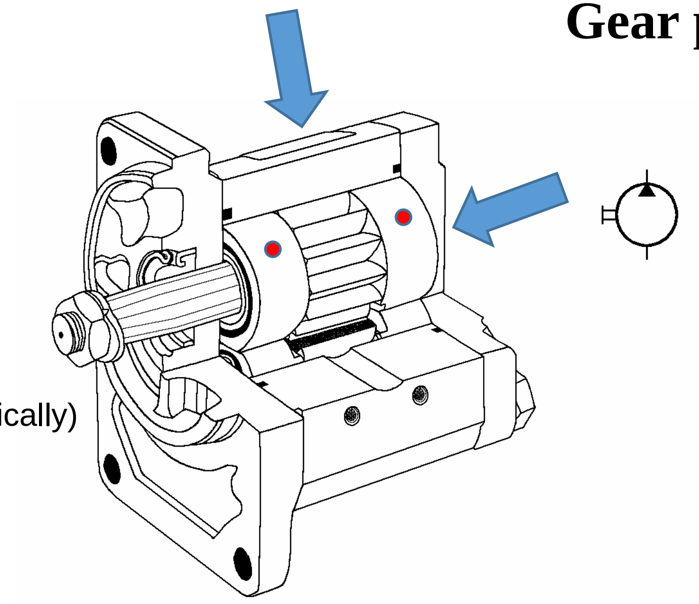
Vane pumps
- vanes in rotor
- vanes in stator

Piston pumps
- line piston pumps
- radial piston pumps
- axial piston pumps

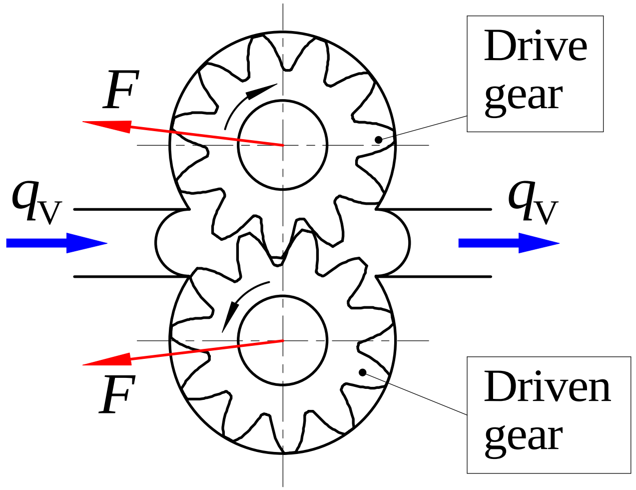
Gear pumps

Reduction of leakage by
- axial gap compensation
- radial gap compensation
in some models

- Hydrodynamic bearings (typically)
- Roller bearings



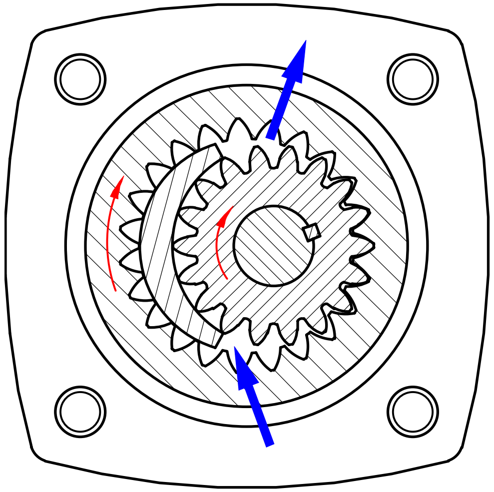
External gear
- two gear
- multi gear



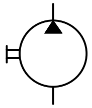
Internal rotor
External rotor

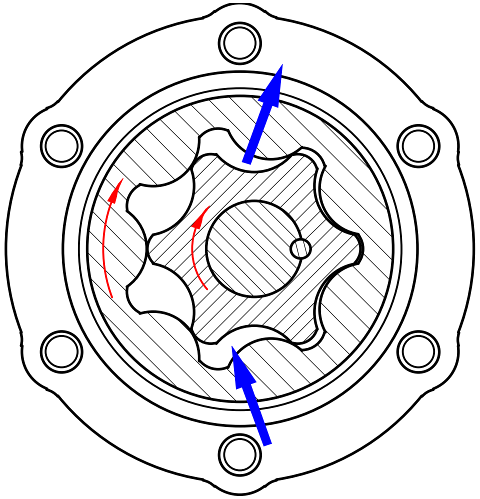
Axial compensation
Radial compensation
possible

- Low noise
- Even flow



q_V





q_V

Internal rotor
External rotor

Also "roller rotor" design

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

Performance characteristics of gear pumps

Total efficiency max. h_t » 0.8 – 0.93

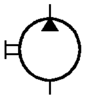
Rotational speed range n » 500 – 5000 r/min

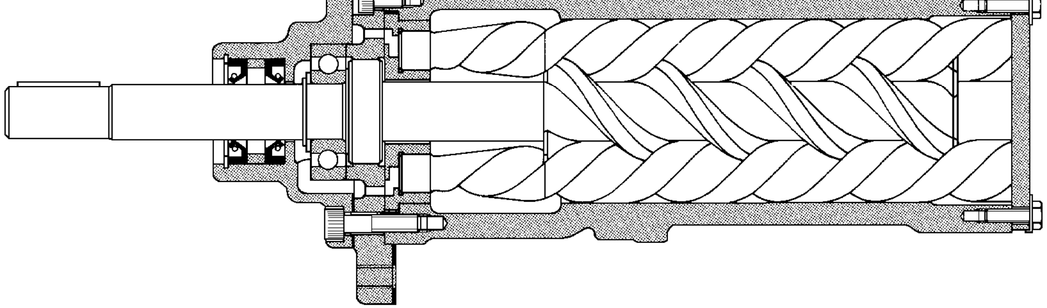
Operating pressure max. p » 14 – 21 (– 32) MPa

- depends on compensation of leakage
  and radial forces

Screw crests roll against screw roots and seal fluid chambers

Screw pumps





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

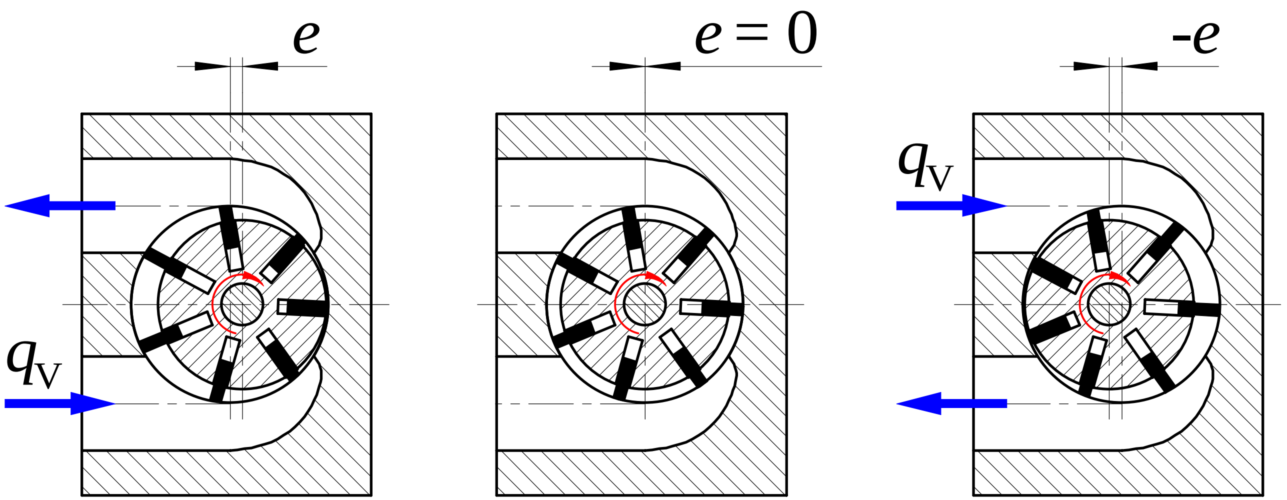
Rotational speed max. n » 30000 r/min

Operating pressure max. p » 14 – 20 MPa

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

Vane pumps

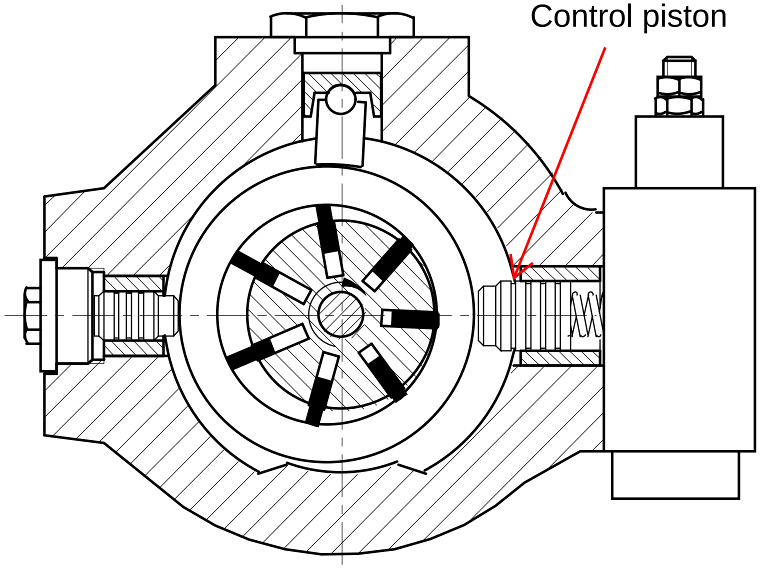
Eccentricity

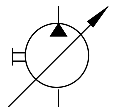


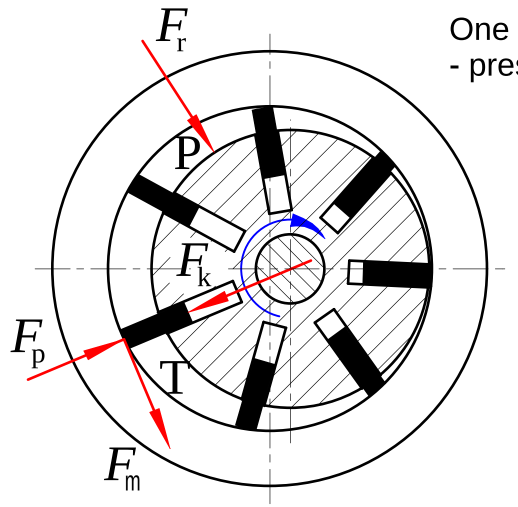
Radially moving (OUT - IN) vanes taking care of sealing

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





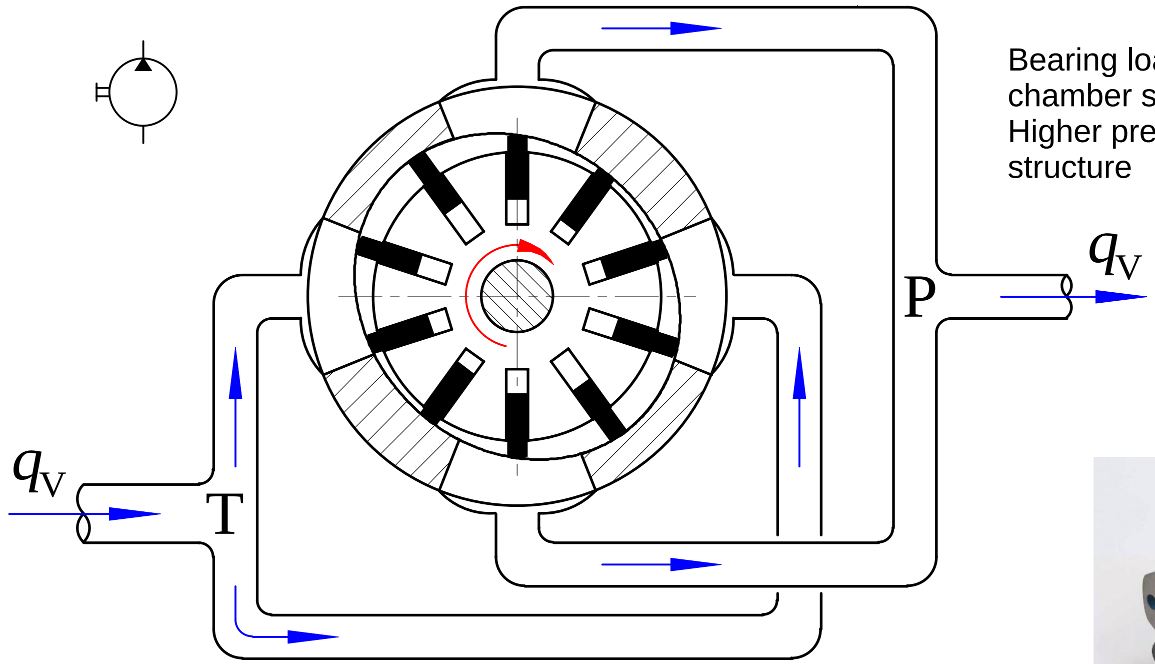


One chamber structure
- pressure forces acting on the shaft

Even flow
- low noise

Vanes in rotor
- one chamber

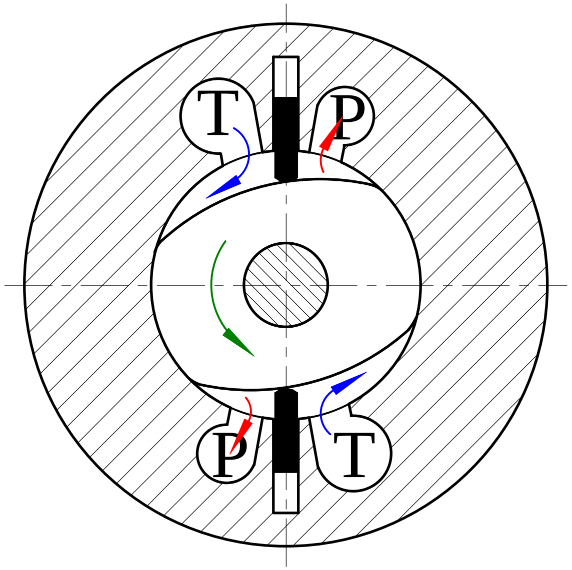
Two chamber structure doubles the flow rate
(compared with one chamber models)

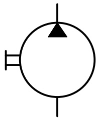


Bearing load forces are smaller than in one chamber structure
Higher pressures possible than in one chamber structure

Variable displacement operation possible with two pump rings in series

Vanes in rotor
- two chamber





Flow rate varies slightly more than in other structures

Vanes in stator

Performance characteristics of vane pumps

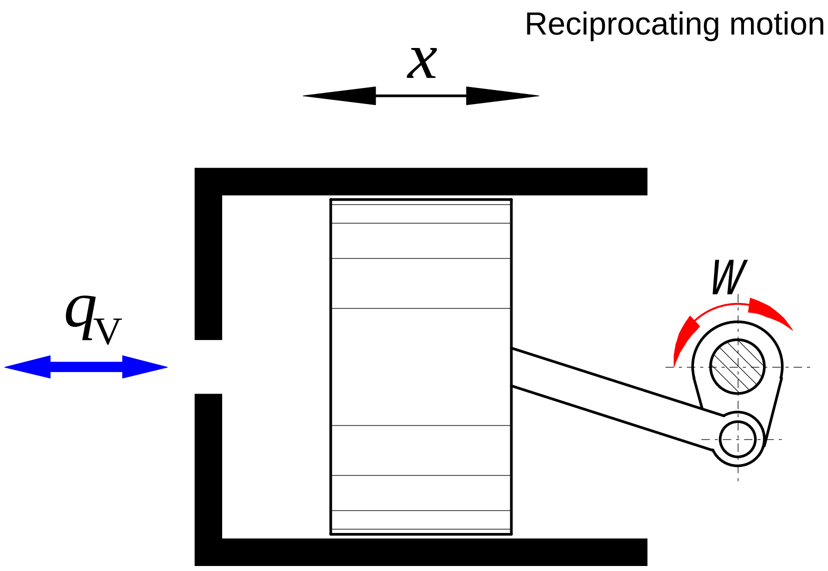
Total efficiency max. h_t » 0.8 – 0.92

Rotational speed range n » 600 – 2500 r/min

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

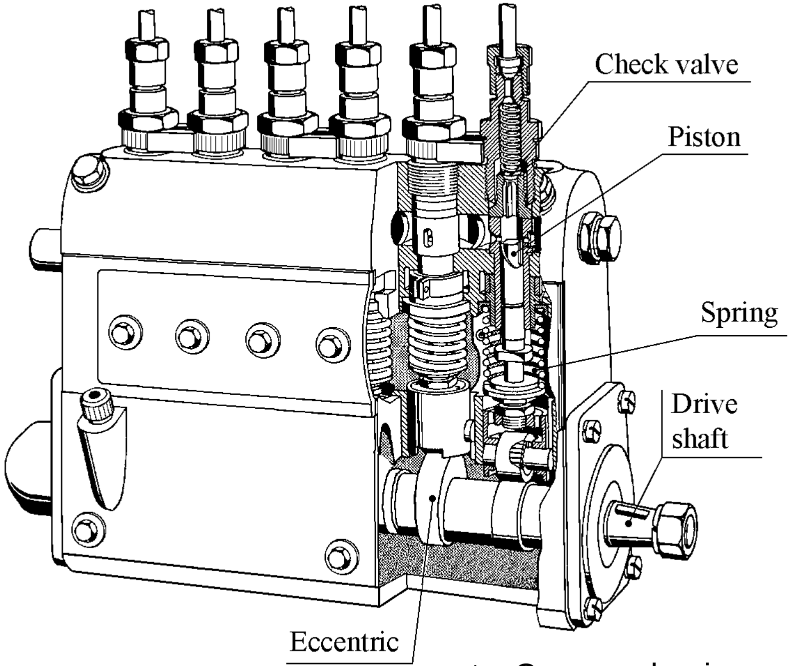
- depends on compensation of leakage, radial forces and number of chambers

Piston pumps

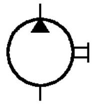


Small clearances ⇒ Small leakages ⇒ Good volumetric efficiency

Piston pumps
- line piston pumps
- radial piston pumps
- axial piston pumps

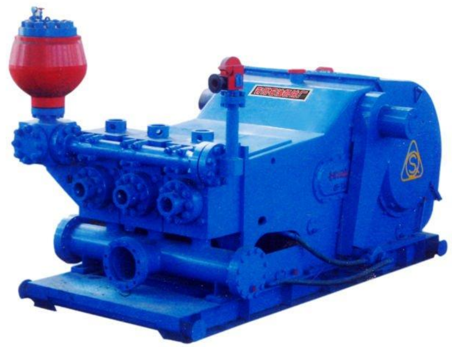


etc. Cam mechanism



Mainly for very high pressures

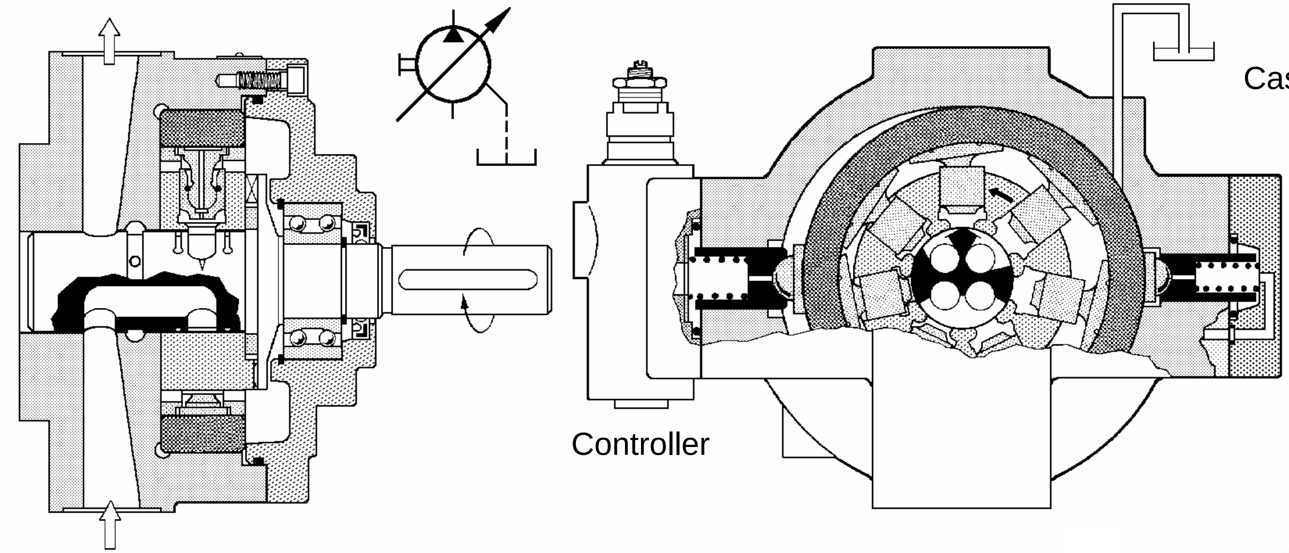
$\Rightarrow$ 1200 bar $\Rightarrow$2500 bar



Line piston pumps

Hydrostatic bearings between pistons and pump ring

Cylinder flows through hollow shaft



Case drain

Variable displacement de-sign
Movable pump ring

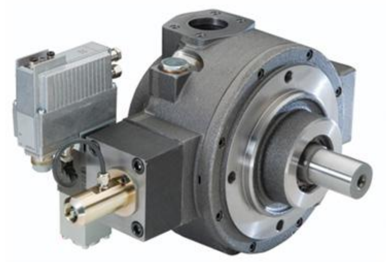
Eccentricity $\Rightarrow$ stroke

Controller

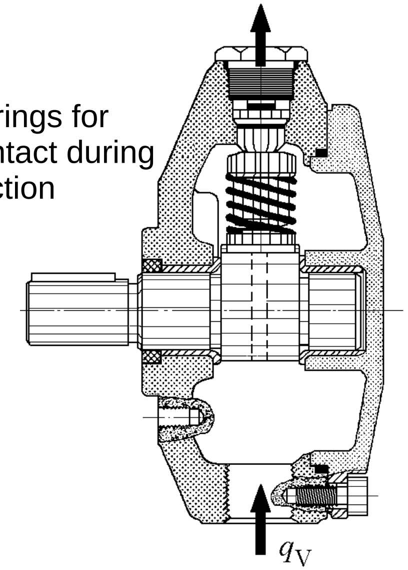
Radial piston pumps

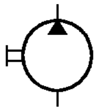
- internal flow channels
- external flow channels

High pressures possible up to 450 bar

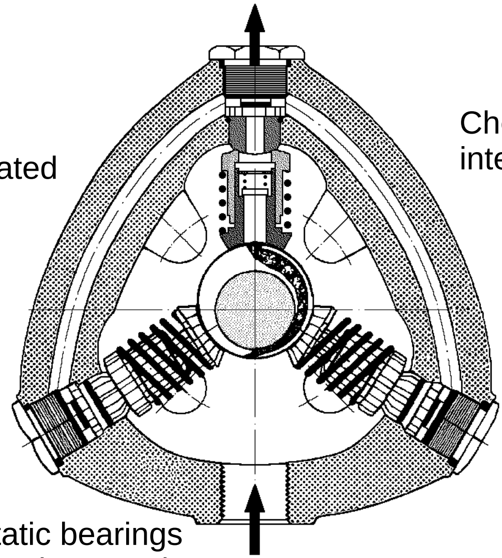


Springs for
contact during
suction





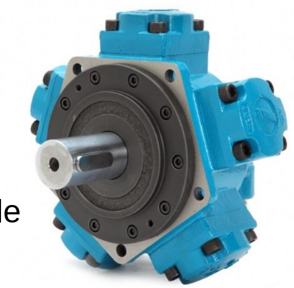
Cam operated



Check valves
integrated in pistons

Pistons do not rotate
in this model

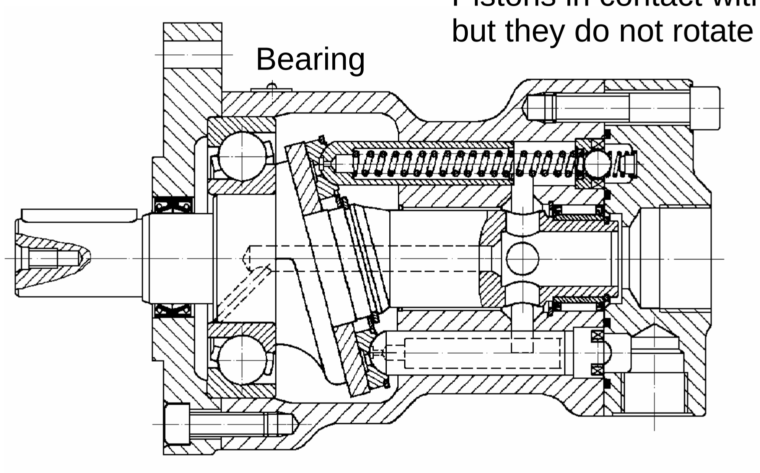
Hydrostatic bearings
between piston and
cam

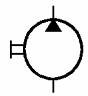


High pressures up to 600 - 700 bar possible

Radial piston pumps
- internal flow channels
- external flow channels

Pistons in contact with the plate, but they do not rotate





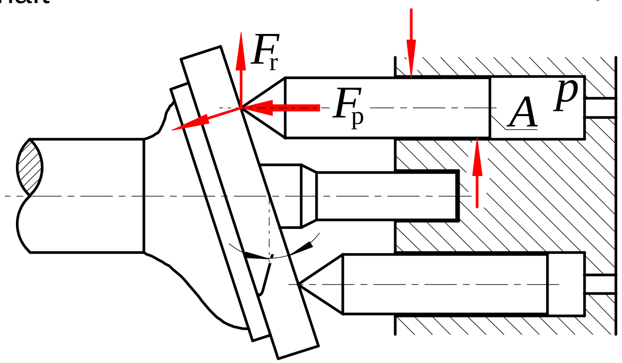
Possibility to transform this type to "digital" by disconnecting some of the pistons

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

Pistons in parallel with the shaft

Wobble plate rotates with shaft

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



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°

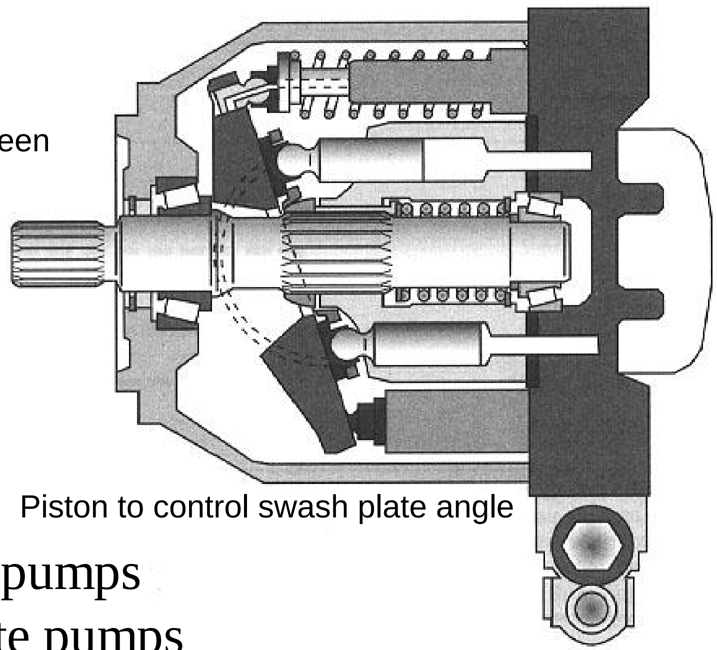
Contact forces between pistons and swash plate limit the control angle

Rotational speeds can be 1500 - 3000 rpm

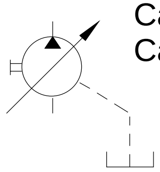
Cylinder block rotates with the shaft
Swash plate does not rotate, it can turn to control the piston stroke and pump displacement

Counter piston for swash plate control

Hydrostatic bearing between pistons and swash plate



Piston to control swash plate angle



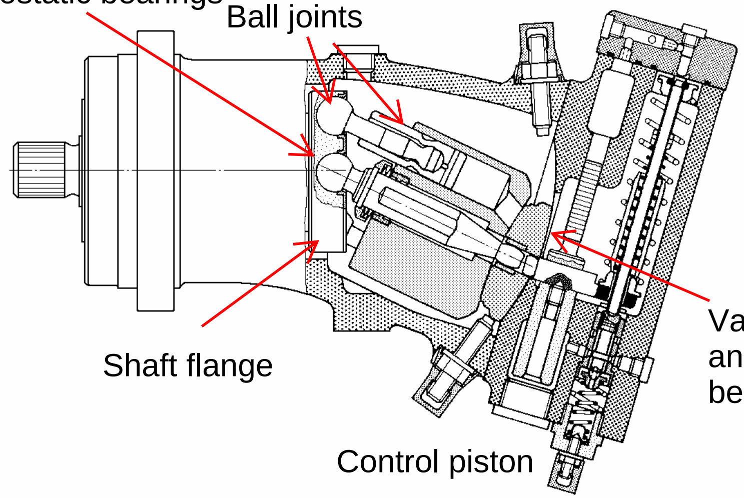
Case drain because of hydrostatic bearings
Case pressure must be kept small!

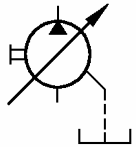
Axial piston pumps

- wobble plate pumps
- swash plate pumps
- bent axis pumps

swashplate pump - inline piston pump

Cylinder block rotates with the shaft

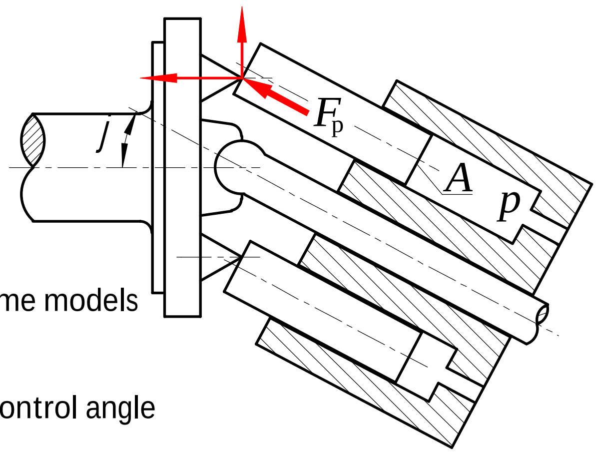




Size is typically large especially with controllers

Rotation transmitted with
- Cardan shaft
- Bevel gear
- Pistons

Radial forces on piston are small



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

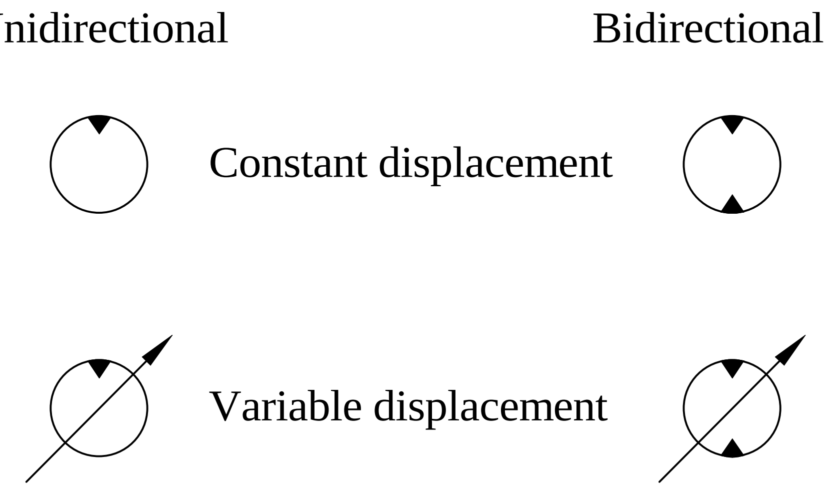
Rotational speed range n » 300 – 8000 r/min

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

- depends on structure type

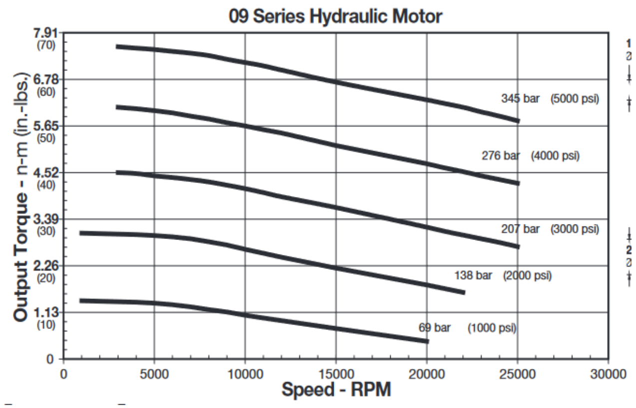
Hydraulic motors

Convert hydraulic power into mechanical power



Speed ranges and structures

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



Parker Oildyne 09 gear motor

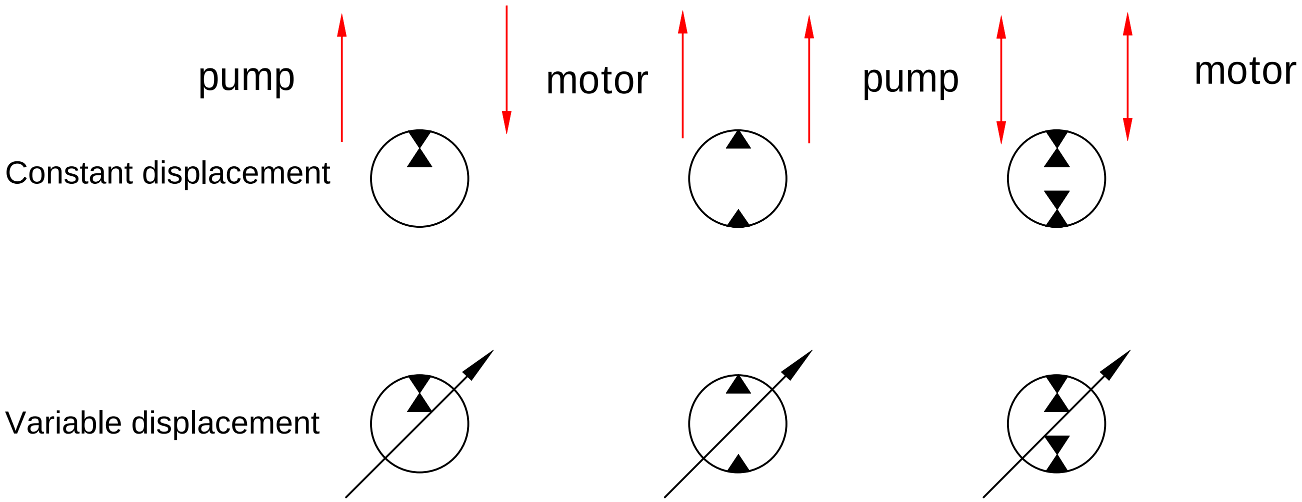
Most common construction types:

- gear
- vane
- piston

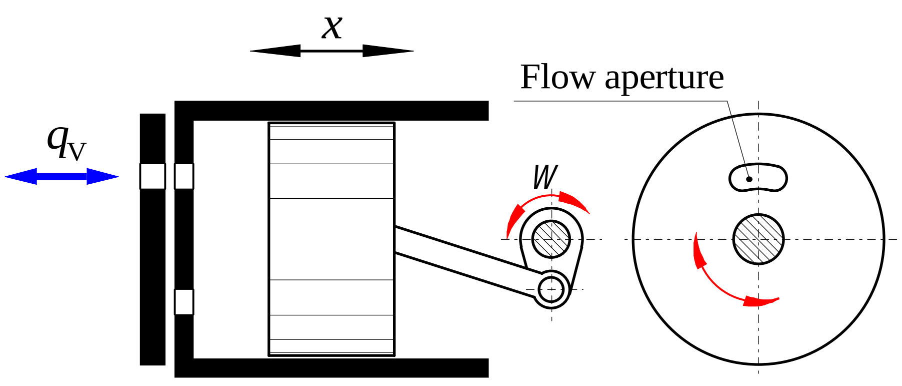
All operate on positive displacement principle

http://www.parker.com/literature/Oildyne/Oildyne%20-%20PDF%20Files/07%20-%2009%20Series%20hydraulic%20gear%20motors.pdf

Pump-motors


pump
motor
pump
motor
Constant displacement
Variable displacement

Control of flow direction



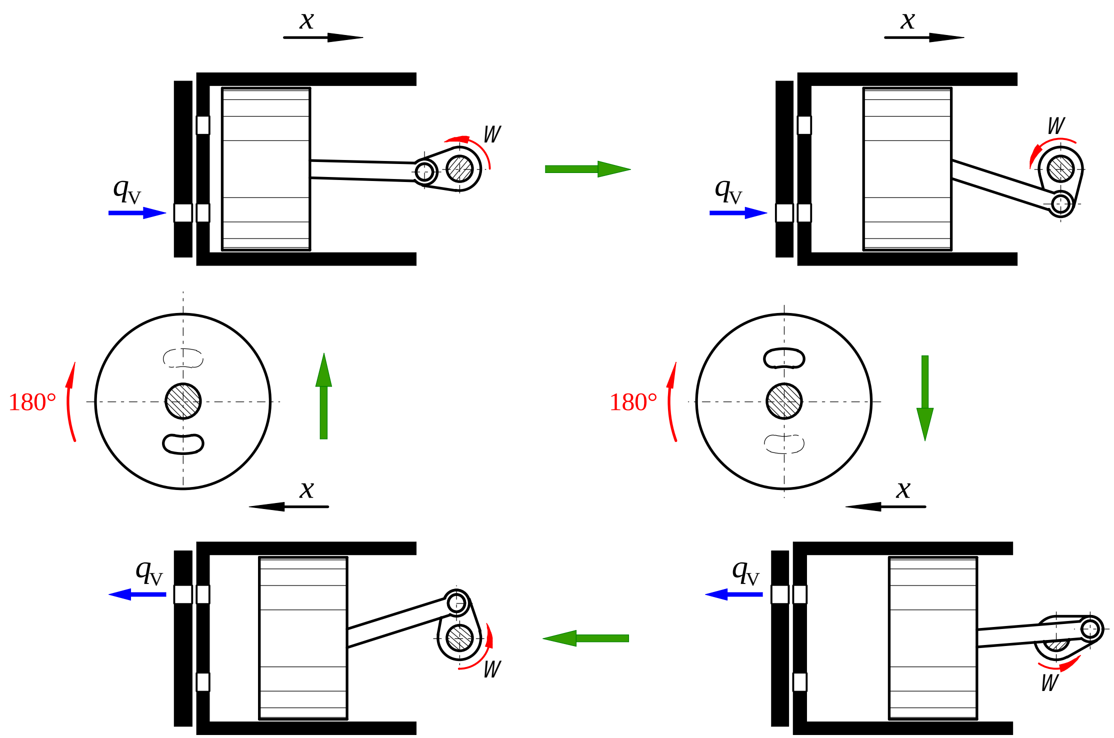
Only forced control is applicable

Operating phases:

Fluid flows into transfer volume – work phase

Fluid flows out from volume – free phase

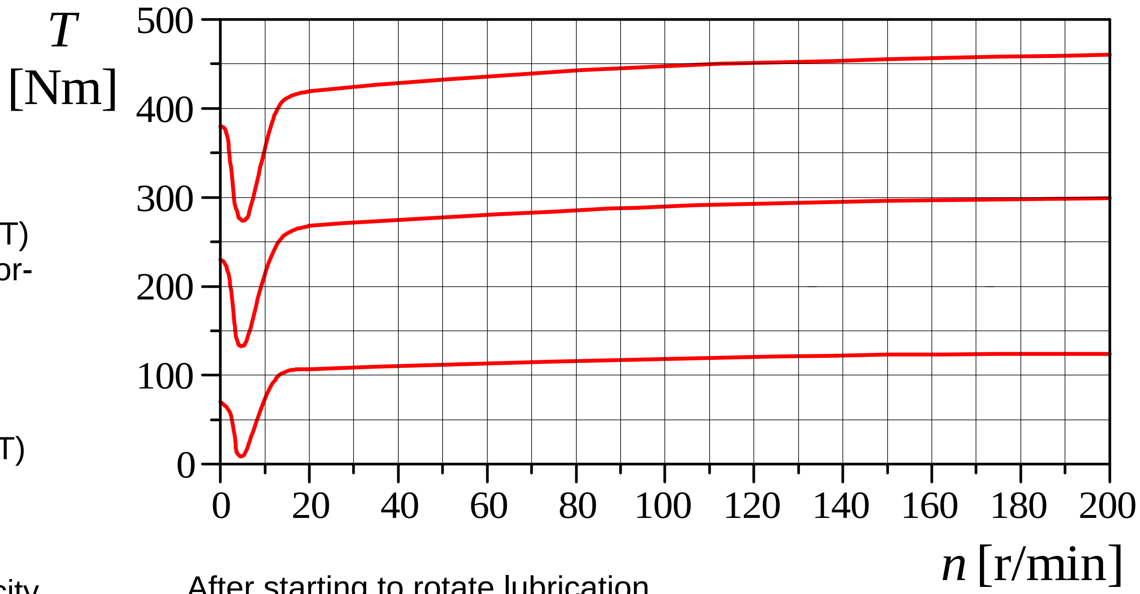
Forced control



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

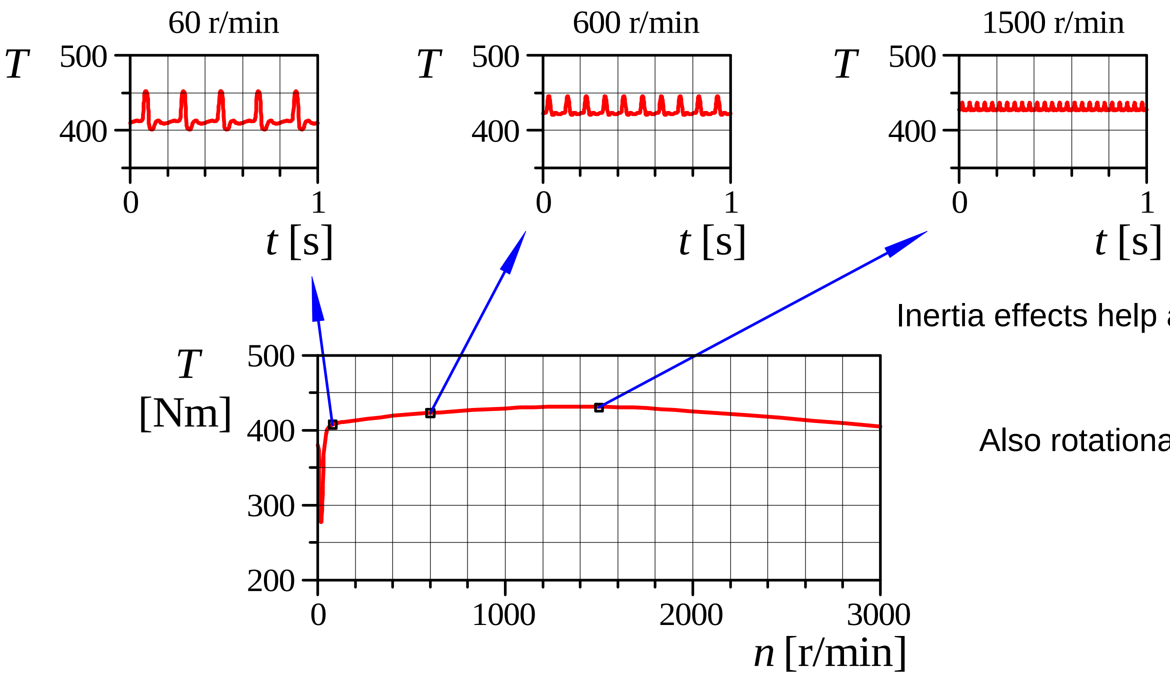


Low Speed High Torque (LSHT)
motors have good quality perfor-
mance at low speeds!
Bearings
hydrostatic - hydrodynamic
Loading of motor?
Full load at slow speeds (LSHT)

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

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

Running characteristics



Inertia effects help at high rotational speeds

Also rotational speed can be uneven

Theoretical flow demand $q_{\mathrm{V,theor}} = n \cdot V_{\mathrm{g}}$

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

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

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

$$\omega = 2\pi \cdot n$$

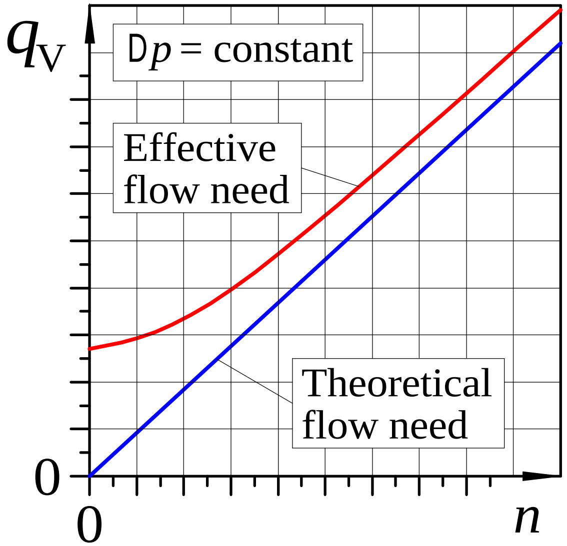
Angular velocity ω [rad/s]

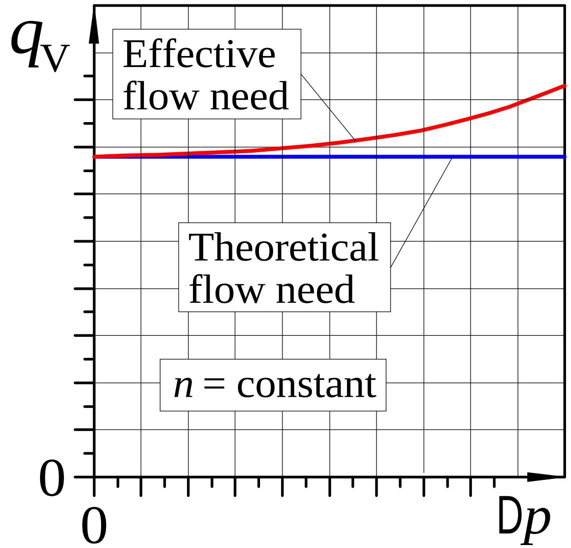
$$V_{\mathrm{rad}} = \frac{V_{\mathrm{g}}}{2\pi}$$

Swept volume per radian V_{rad} [m^3/rad]

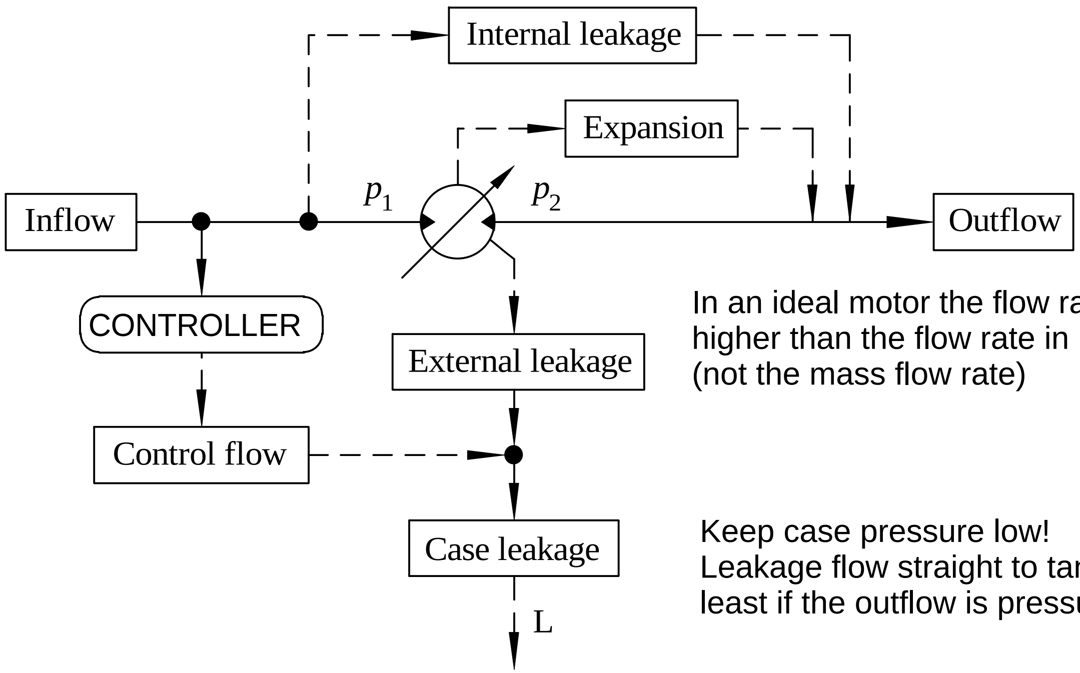
Effective flow demand $$q_{V,real} = \frac{n \cdot V_g}{\eta_v}$$

Leakage – volumetric efficiency η_v





Leakage flows in motors



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

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

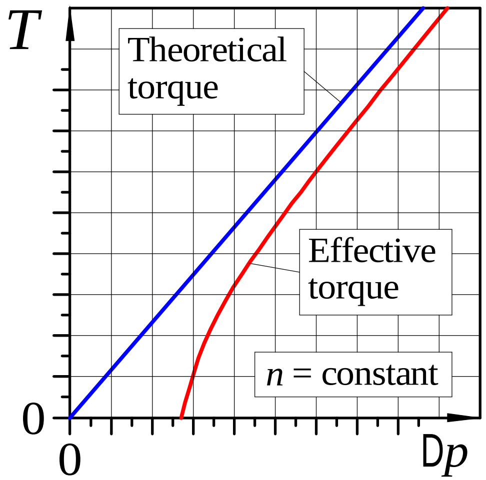
Theoretic pressure demand

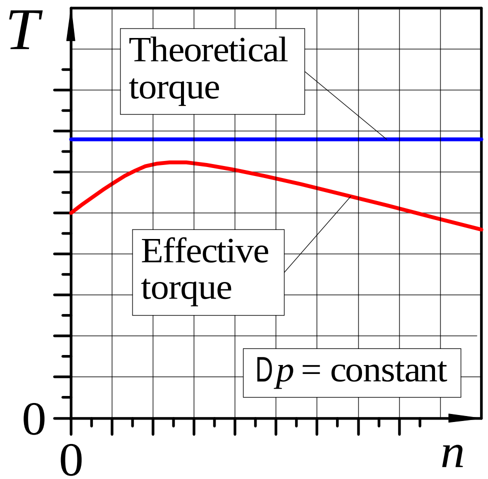
$$\Delta p_{\mathrm{theor}} = \frac{2 \cdot \pi \cdot T}{V_{\mathrm{g}}}$$

Effective pressure demand

$$\Delta p_{\mathrm{real}} = \frac{2 \cdot \pi \cdot T}{V_{\mathrm{g}} \cdot \eta_{\mathrm{hm}}}$$

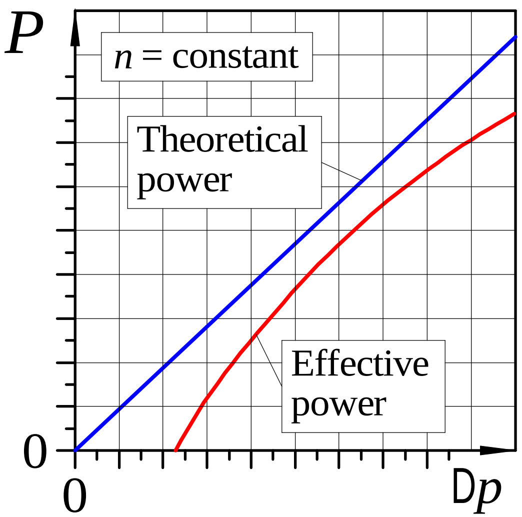
Friction – hydro-mechanical efficiency η_{hm}

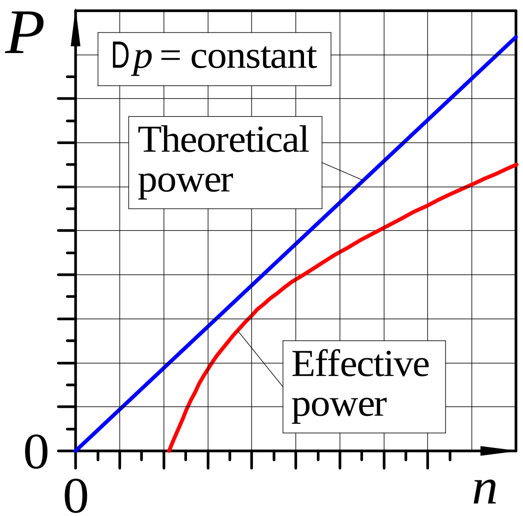




Theoretic power demand $P_{\mathrm{theor}} = q_{\mathrm{V}} \cdot \Delta p = T \cdot \omega$

Effective power demand $P_{\mathrm{real}} = q_{\mathrm{V}} \cdot \Delta p = \dfrac{T \cdot \omega}{\eta_{\mathrm{t}}}$ $\eta_{\mathrm{t}} = \eta_{\mathrm{v}} \cdot \eta_{\mathrm{hm}}$

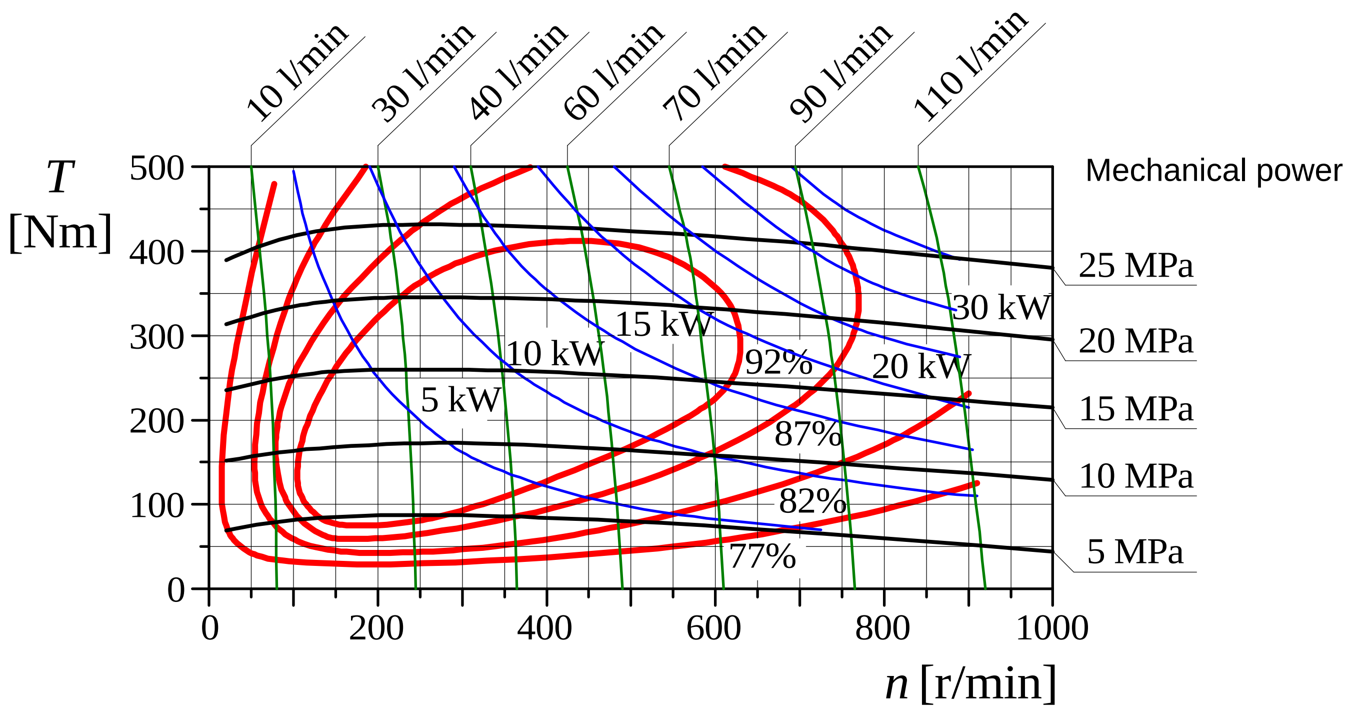




Power demand of load

$$P_{\mathrm{mech}} = T \cdot \omega = 2 \cdot \pi \cdot n \cdot T$$

Characteristic curves of motor



Low speed high torque motors (LSHT)

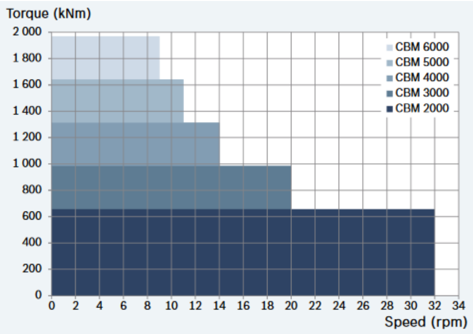
Large swept volume
- large displacement area of working elements
- several work stages per one rotation of the axle

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

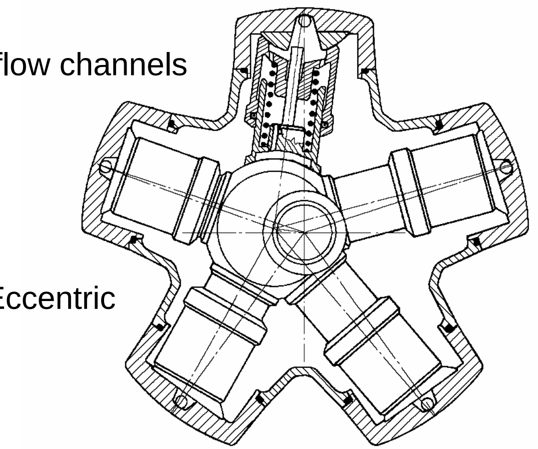
https://www.parker.com/literature/Vane_Pump/PDF%20Literature/MRD-MRDE-MRV-MRVE.pdf
https://www.parker.com/literature/Vane_Pump/PDF%20Literature/MR-MRE.pdf

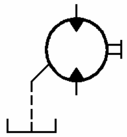
Radial piston motors
- external flow channels
- internal flow channels

Hydrostatic bearings

External flow channels

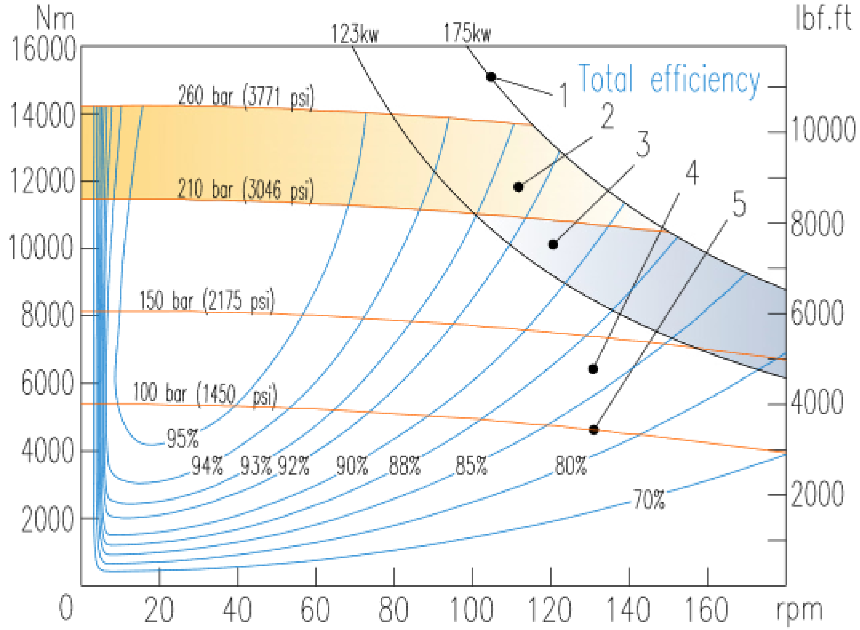
Eccentric



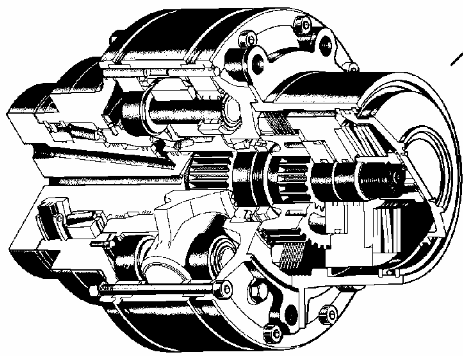


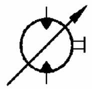
Case drain

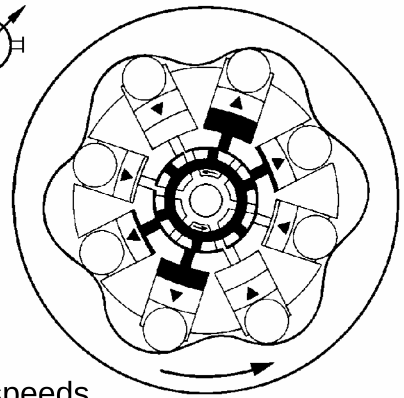
Up to 420 bar



Internal flow channels







Cam mechanism, several cams for one revolution

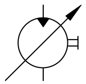
Rotating housing

Opposite cylinder forces cancel internal bearing loadings

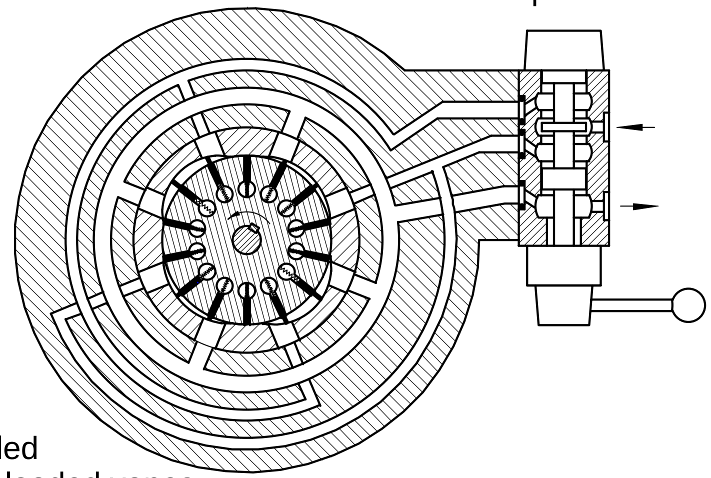
Can be freewheeled (springs) during on-road operation (trailer), 2 speeds

https://www.blackbruin.com/

Stepwise change of displacement



Three displacements: 1, 2 and 1+2



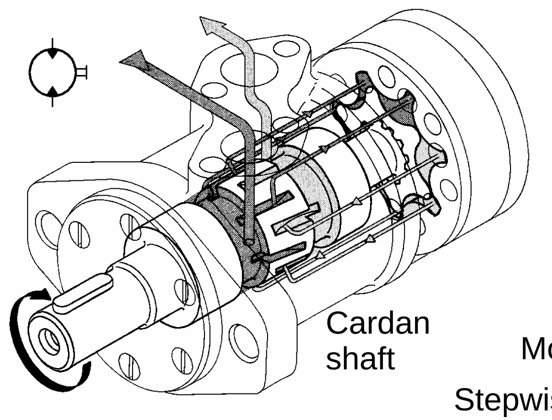
¬ Multi chamber vane motor

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

4 chamber model
Radial forces cancelled
Spring and pressure loaded vanes

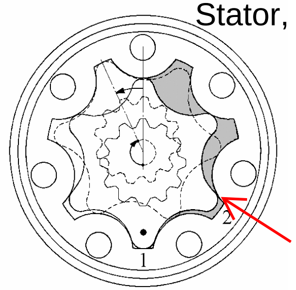
Drum distributor valve (or disc valve)

Orbital motor ®



Cardan shaft

External gear
Stator, 7 cams



For one rotation 42 tooth volumes have to be filled

Rotor, 6 cams
Internal gear

Models with rollers also

Stepwise displacement changes possible

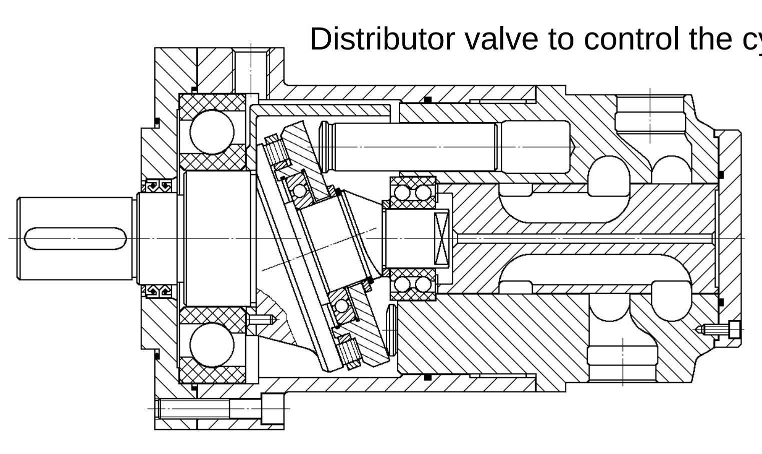
Middle speed range motors

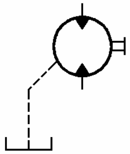
Gerotor motors (ring motors)
Wobble plate motors

Total efficiency max. h_t » 0.8 – 0.88

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

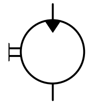
Torque max. T » 20 – 200 (– 1200) Nm

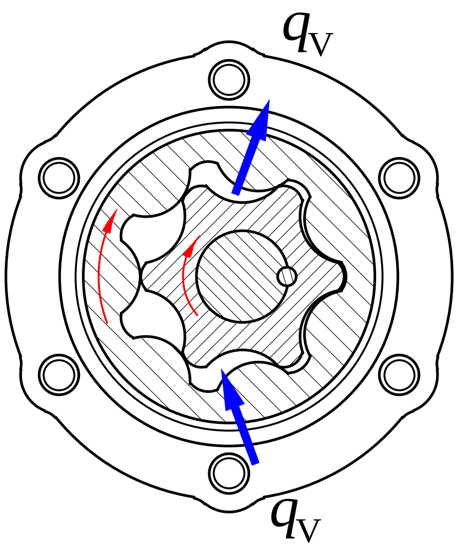




¬ Wobble plate motors

Constant displacement
Medium speed because of rotor's mass imbalance





Gerotor motor (ring motor) ®

High speed range motors

External gear motors
Vane motors
Axial piston motors

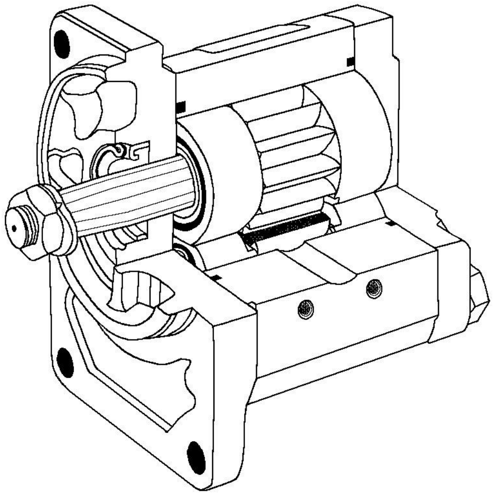
Total efficiency max. h_t » 0.82 – 0.9

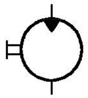
Rotational speed range n » 100 – 3000 (– 6000) r/min

Torque max. T » 10 – 700 (– 3000) Nm

Rotational speed may change frequently -> accelerations

Medium pressure (180 bar) unless radial compensation (250 bar)





External leakage connection needed if outlet flow is pressurized

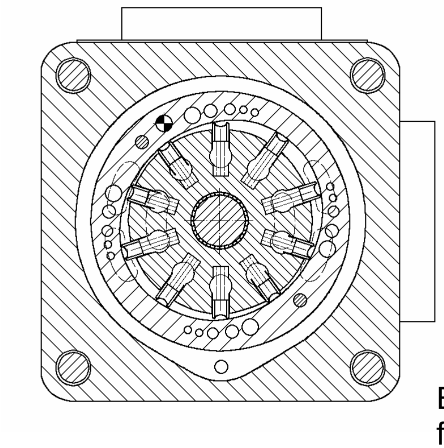
¬ External gear motor

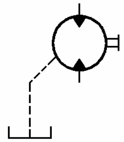
Some models with needle bearings for better performance at low speeds

Medium pressure (up to 210 bar)

Constant or variable displacement

Vane motor ®

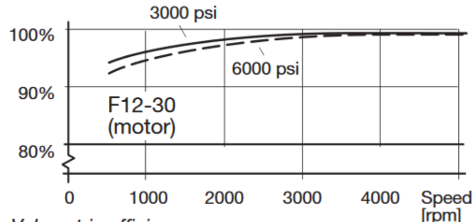




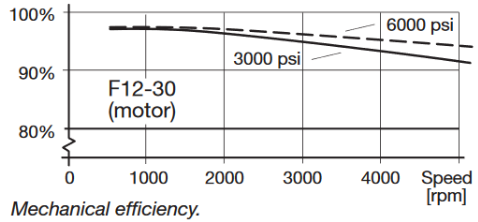
Hydraulic forces in balance

External leakage connection for pressurizing of both flow channels

Parker F12 bent axis motor



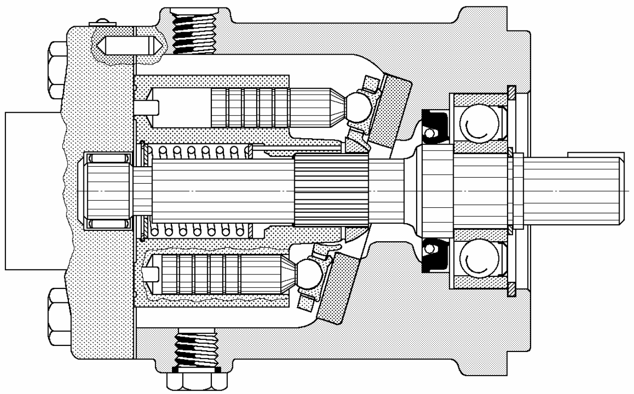
Volumetric efficiency.



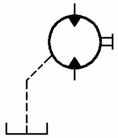
Mechanical efficiency.

Distributor valve

Static swash plate



Up to 350 bar
High efficiency



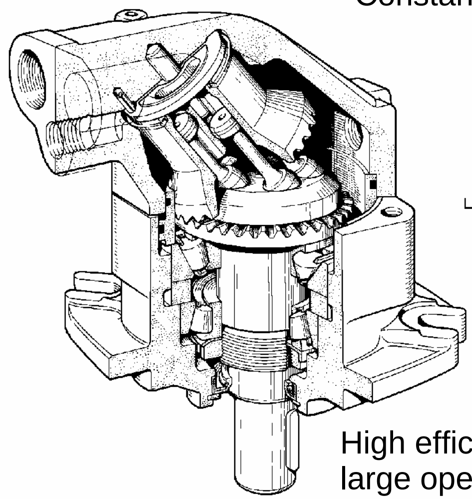
¬ Swash plate piston motor

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

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

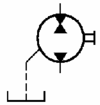
Bent axis ® piston motor

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

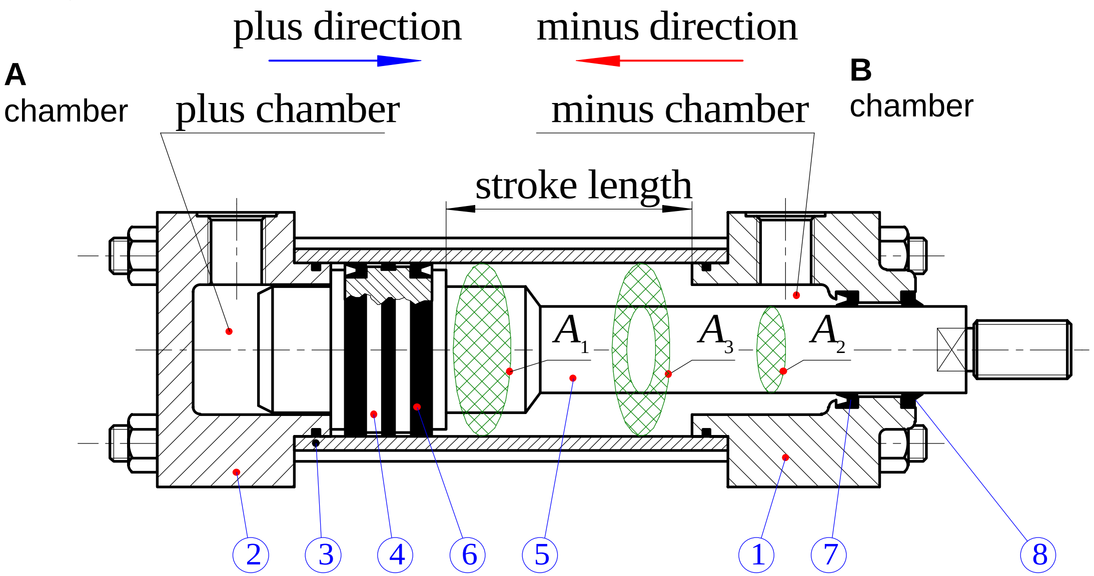
Convert hydraulic power into mechanical power

Construction types:

- single acting
- double acting

Both operate on positive displacement principle

Terminology



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

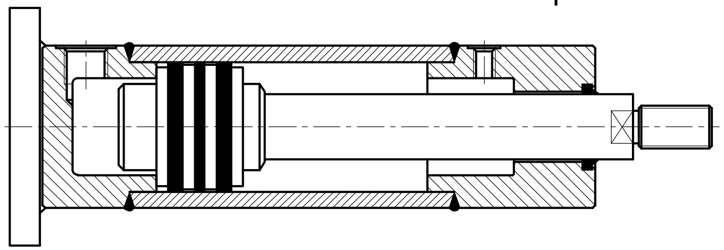
Plunger cylinder integrated in
reach truck's mast

http://www.rocla.com/en/products/roclas-humanic2-reach-trucks

Single acting cylinders

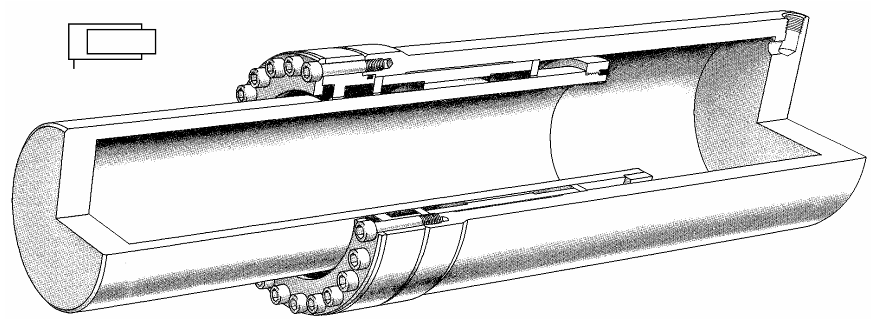
Operate hydraulically to only one direction

Vent port - atmospheric pressure



¬ Piston type cylinder

Plunger cylinder ®

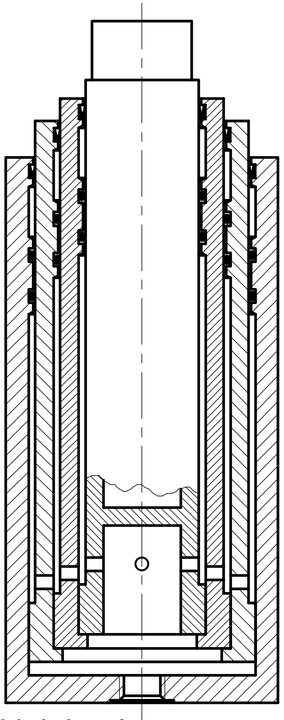


These models are single acting

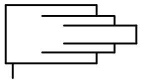
Telescopic cylinder

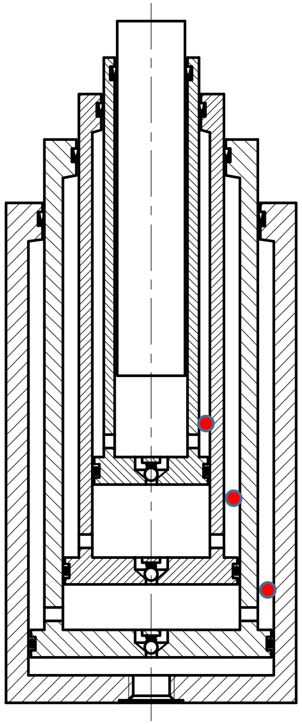
Variable speed
- the largest piston starts
(the least pressure need)
- the smallest piston is the last
(the highest pressure need)

Used in dump trailers etc.



a



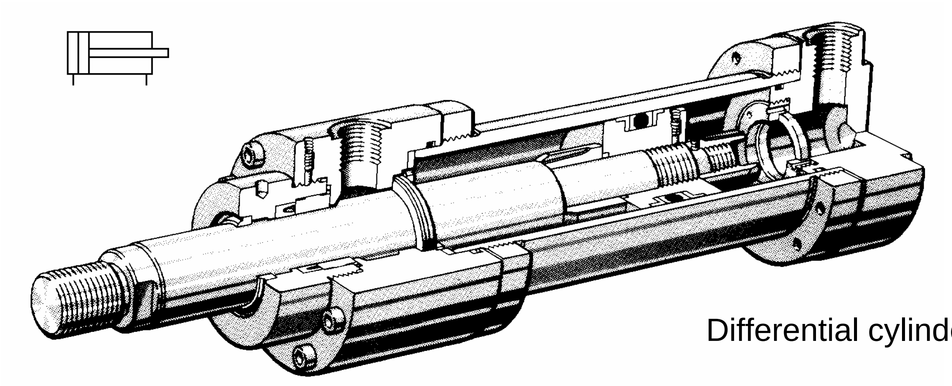


b

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

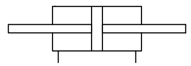
http://www.nurmihydro.fi/products_akipit.html

Double acting cylinders



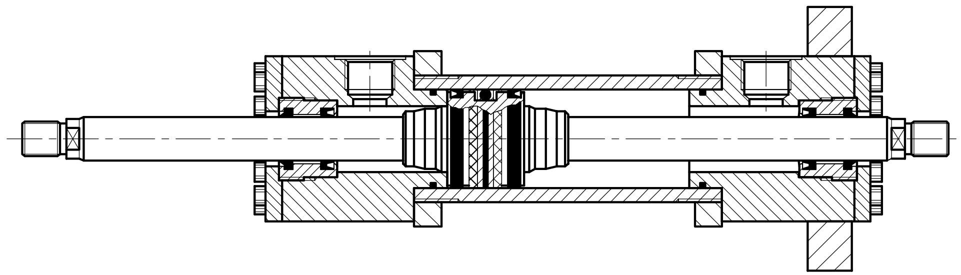
¬ Single piston rod

Differential cylinder



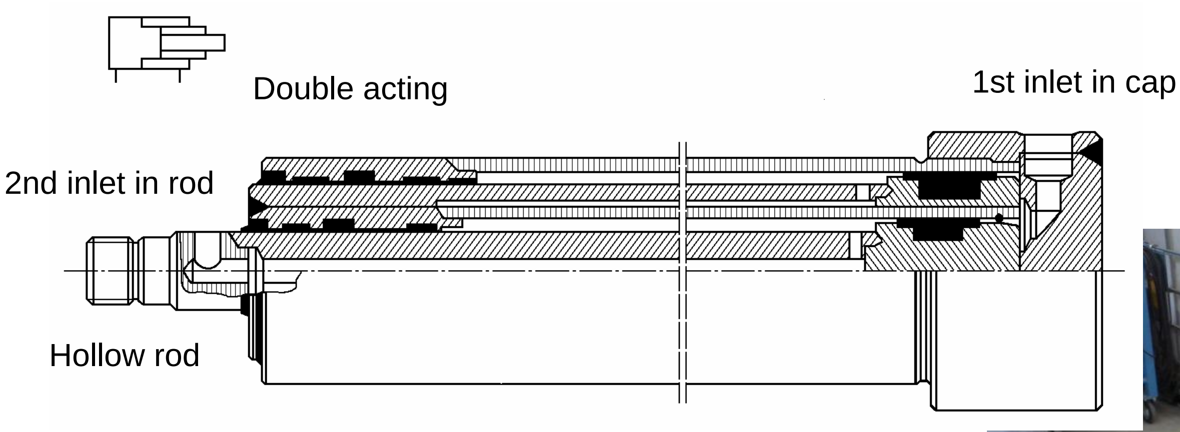
Double piston rod

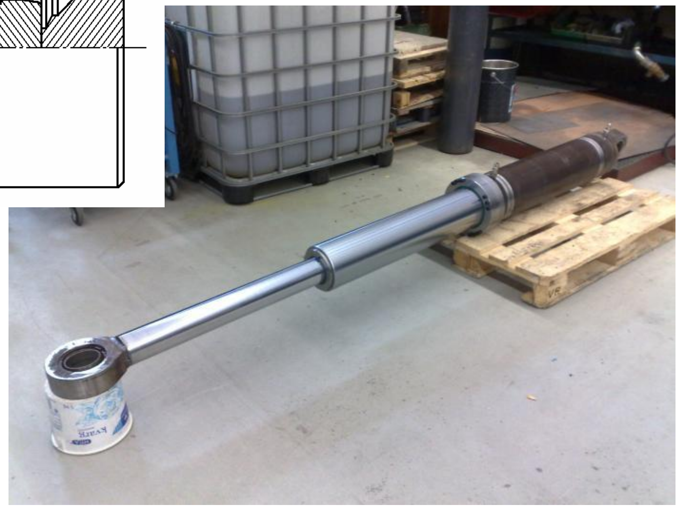
® Symmetric cylinder



Aalto University
School of Engineering
Mechanical Engineering / Engineering Design / Mechatronics / Fluid Power

Telescopic cylinder





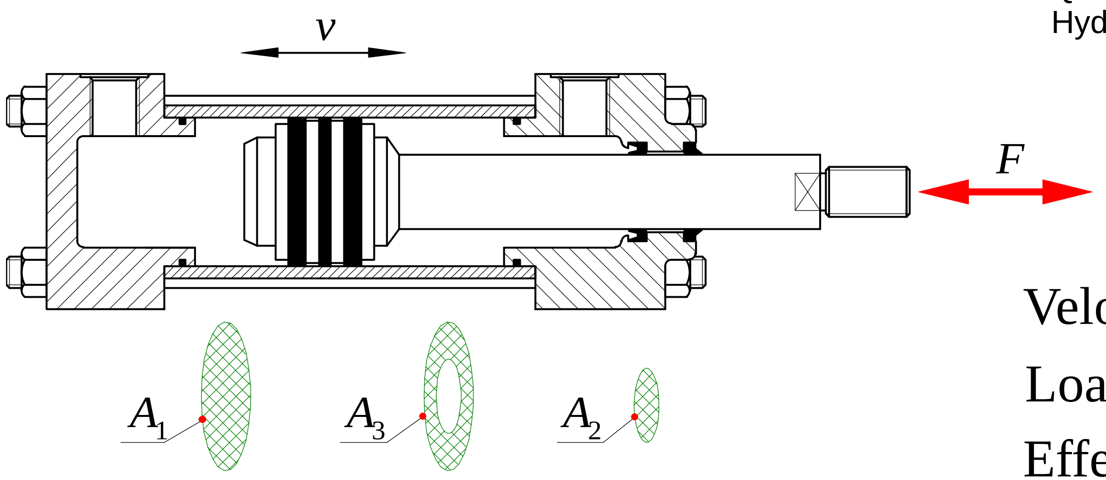
hydraulicpower.fi

Characteristics of cylinders

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

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

Quasi-static process
Hydraulic variables do not change



Velocity v [m/s]

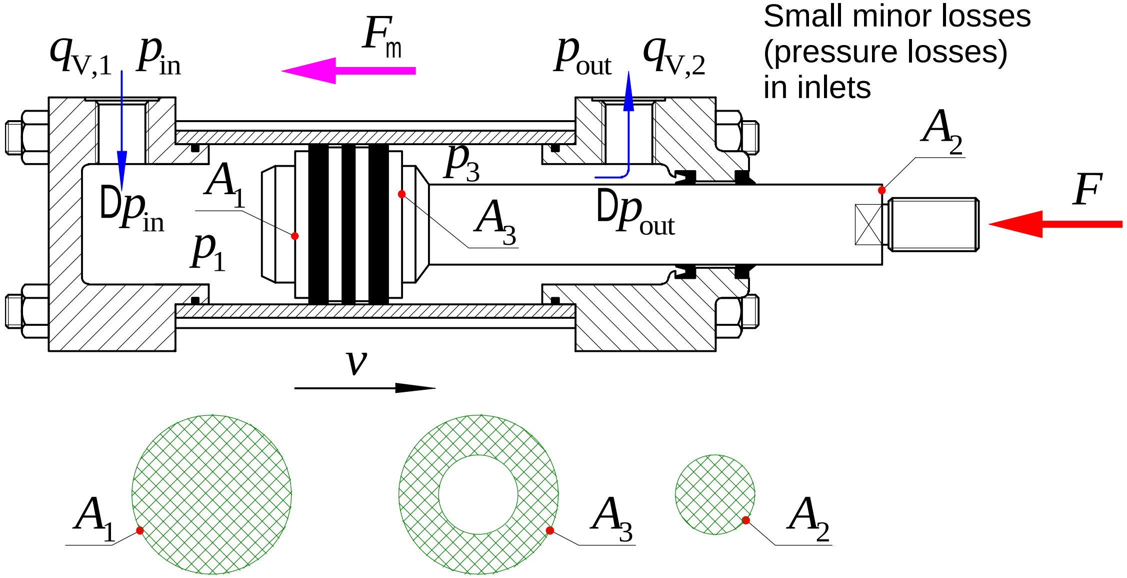
Load force F [N]

Effective piston area A [m^2]

Reality

Small minor losses (pressure losses) in inlets

Small minor losses (pressure losses) in inlets



Example case:
plus-direction movement,
opposing force

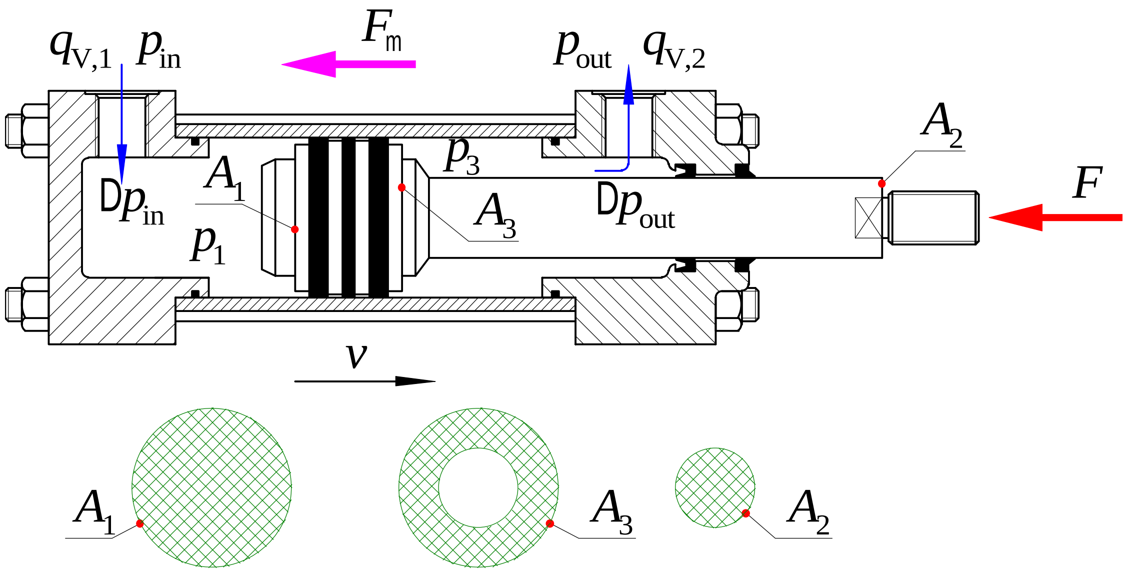
Friction – hydromechanical efficiency η_{hm}

Leakage – volumetric efficiency η_{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

$$p_{in} \cdot A_1 - \Delta p_{in} \cdot A_1 = p_{out} \cdot A_3 + \Delta p_{out} \cdot A_3 + F + F_{\mu}$$

Load force

Friction force (opposing movement)

Force equation in general form

Flow OUT increases the actual chamber pressure because of minor loss

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

Flow IN decreases the actual chamber pressure because of minor loss

Actual pressure demand

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

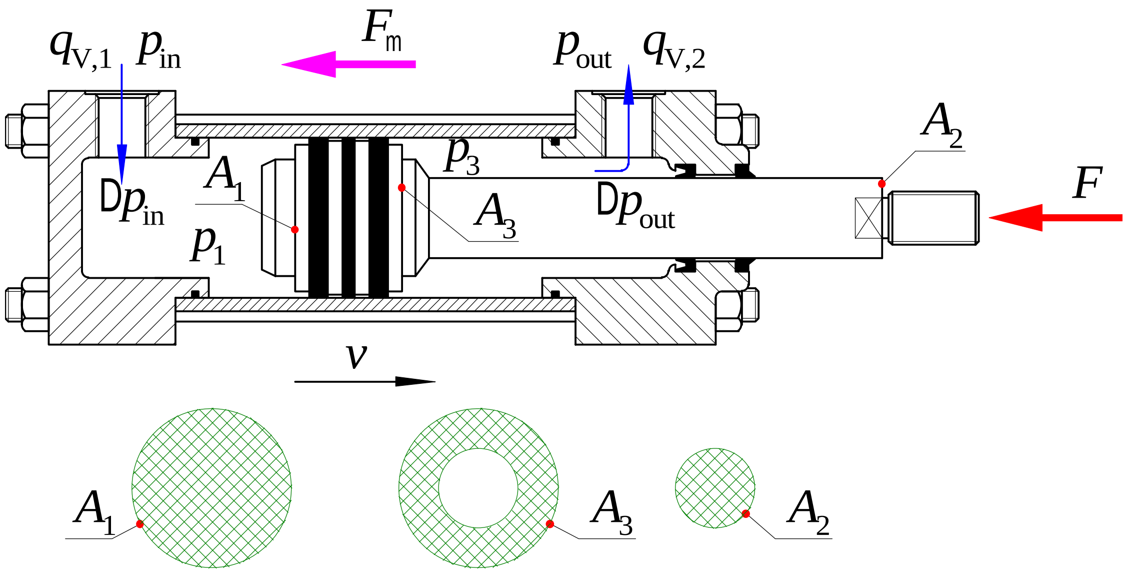
In efficiency form

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

Correct or not?
Hydromechanical efficiency depends on what?

Flow

Example case:
plus-direction movement,
opposing force

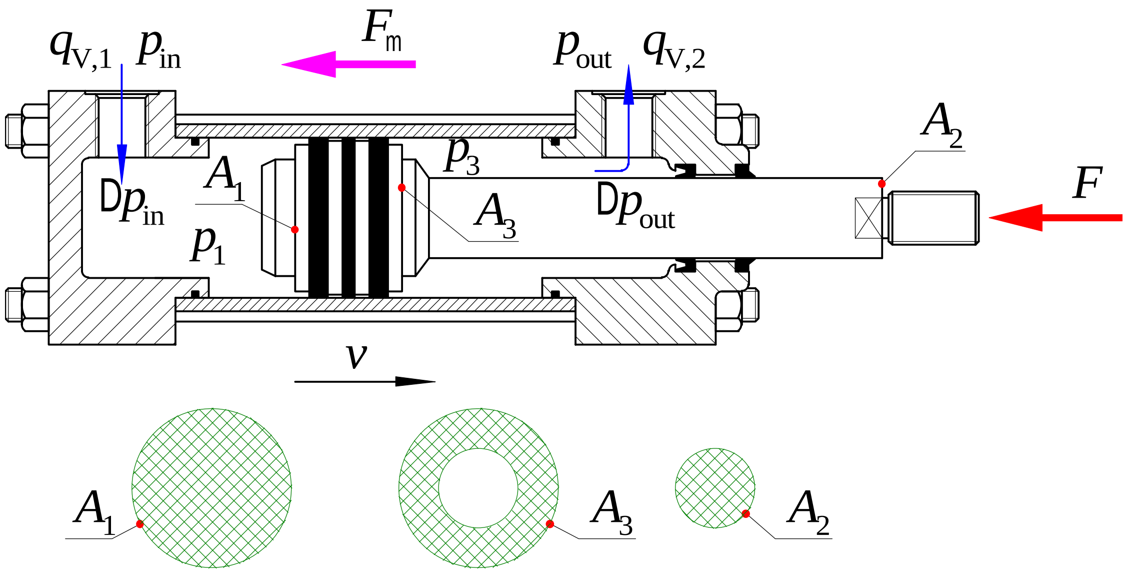


Actual flow demand $$q_{V,in,real} = \frac{A_{in} \cdot v}{\eta_v}$$

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 $$P_{mech} = F \cdot v$$

Theoretic power demand

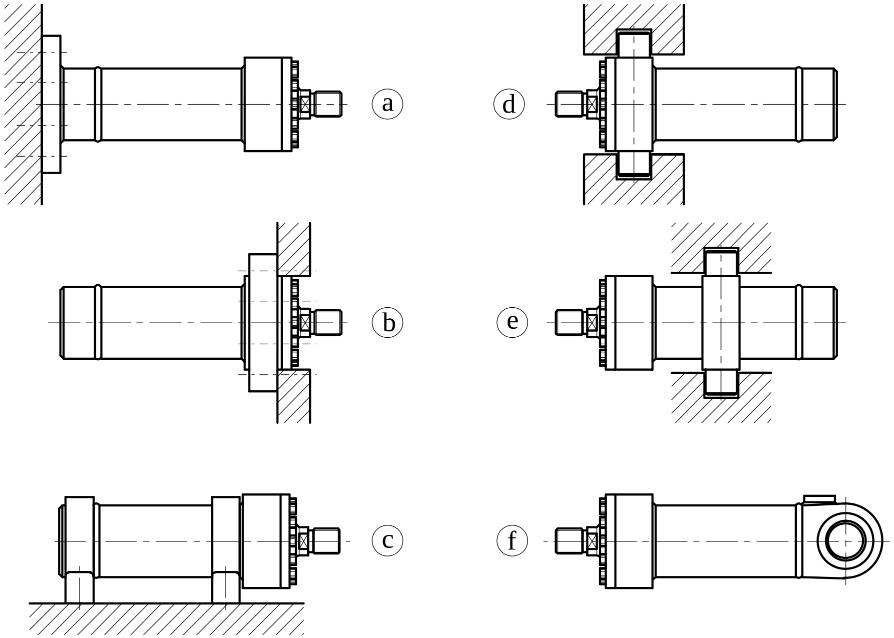
$$P_{\text{theor}} = q_{\text{V,in}} \cdot \left(p_{\text{in}} - \frac{A_{\text{out}}}{A_{\text{in}}} \cdot p_{\text{out}} \right) = F \cdot v$$

Actual power demand

$$P_{\text{real}} = q_{\text{V,in}} \cdot \left(p_{\text{in}} - \frac{A_{\text{out}}}{A_{\text{in}}} \cdot p_{\text{out}} \right) = \frac{F \cdot v}{\eta_{\text{t}}}$$

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.

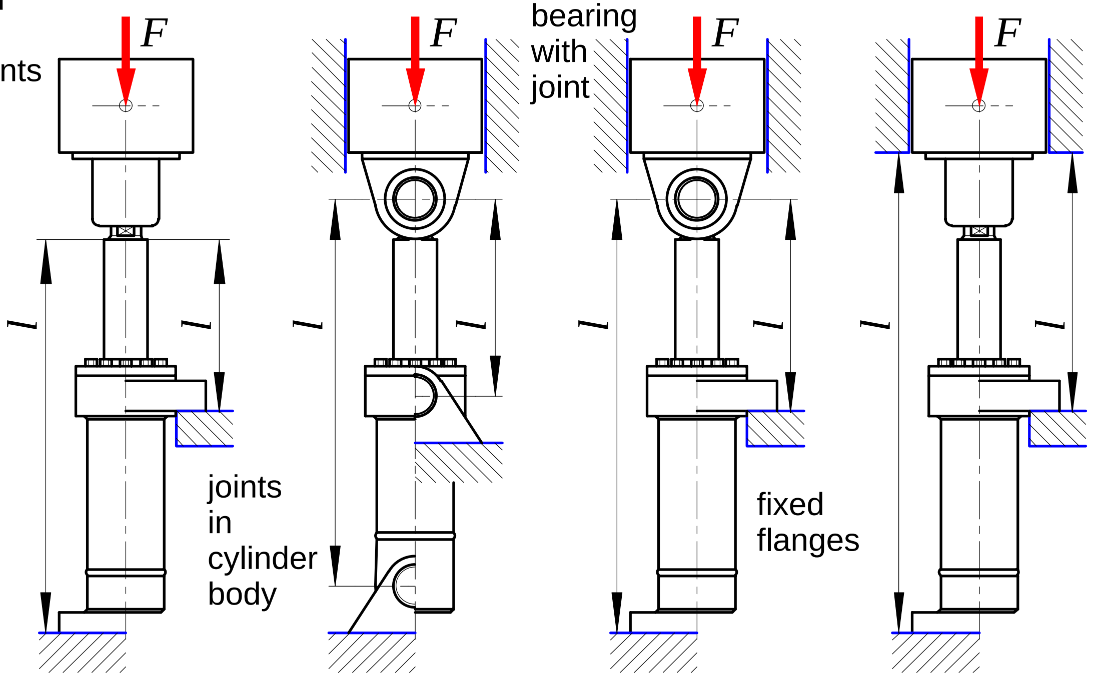
Loading and buckling of cylinders



Loading only in parallel to the piston rod!

Mounting and buckling length

Reduced length l_R of cylinder depending on the
a) point and
b) way of attachment



$l_R = 2l$ $\quad$ $l_R = l$ $\quad$ $l_R = \sqrt{\frac{1}{2}}\, l$ $\quad$ $l_R = \frac{1}{2}\, l$

Large reduced length l_R indicates high buckling risk

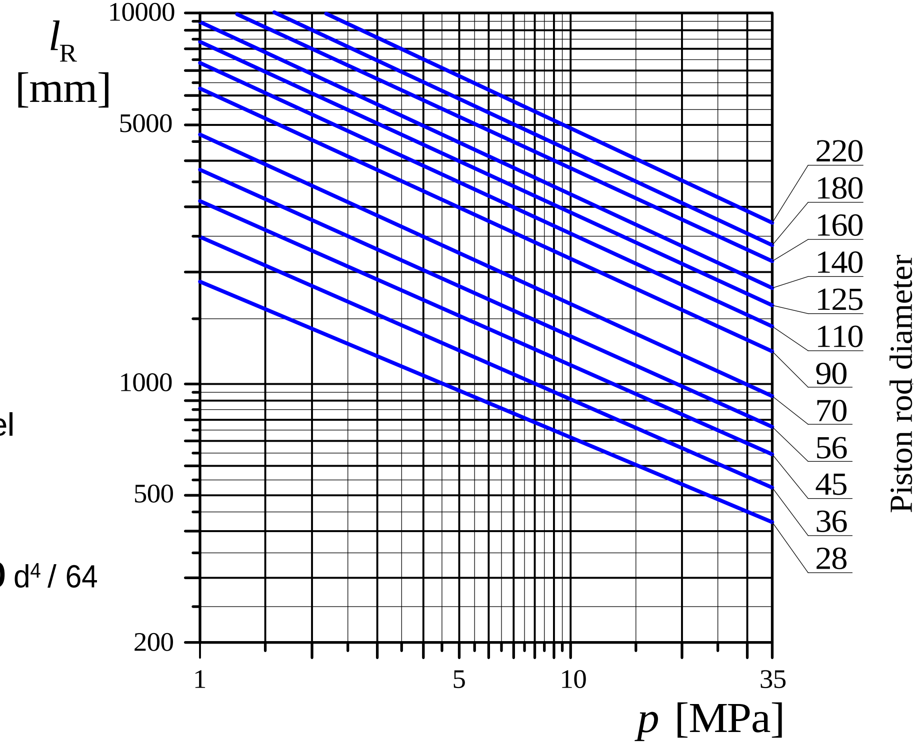
Buckling diagram

Safety factor typically 4

$$F = \frac{\pi^2 \times E_m \times I}{C_n \times l_R^2}$$

Young's modulus ~ 210 GPa for steel

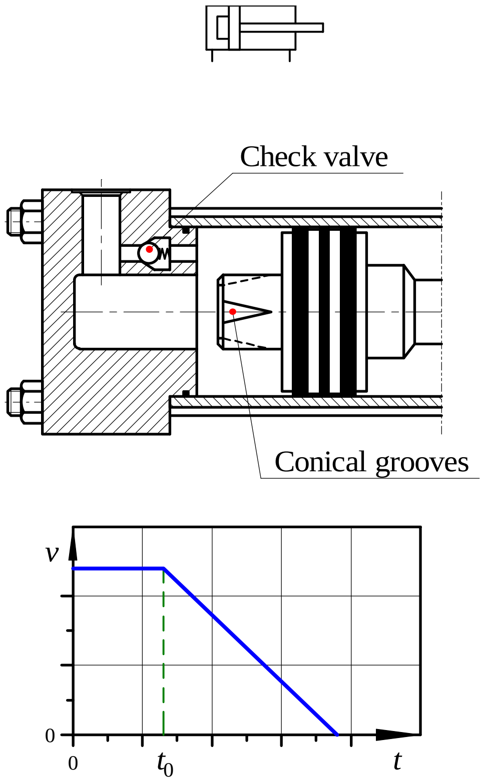
E_m	modulus of elasticity	
I	area moment of inertia	$\pi d^4 / 64$
C_n	safety factor	
l_R	effective length	

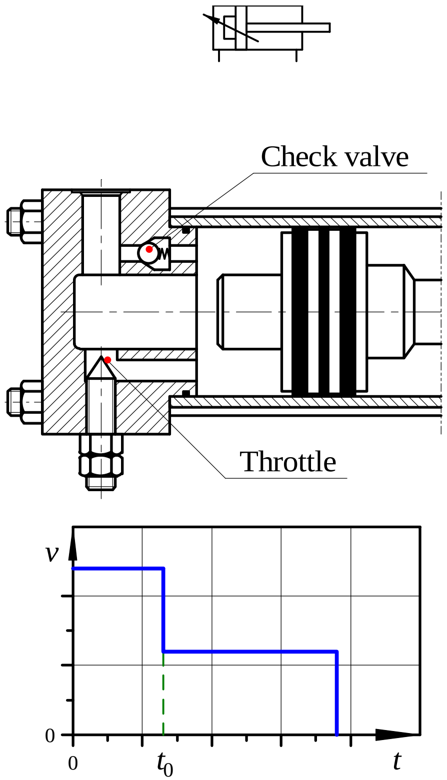


End cushioning of cylinders

At velocities
> 0.1 m/s

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





Flow area and flow resistance remain 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°

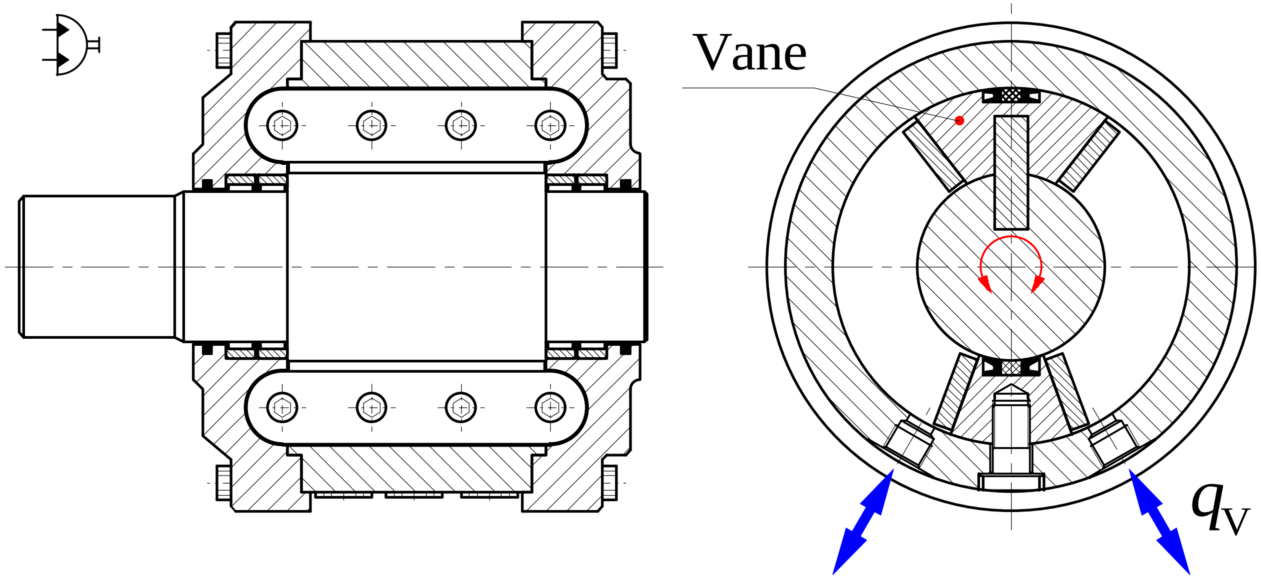
Construction types:
- vane
- piston

Both operate on positive displacement principle

Total efficiency max. h_t » 0.6 – 0.86

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

Vane type



Maximum rotation angle ~320°

With end cushioning ~240°

Uneven pressure distribution causes high bearing loads

With 2 vane structure higher torques and pressure forces can be compensated ⇒

smaller turning angle ~170°

Constant torque
Leakage reduces total efficiency

Maximum ~72%

Piston type

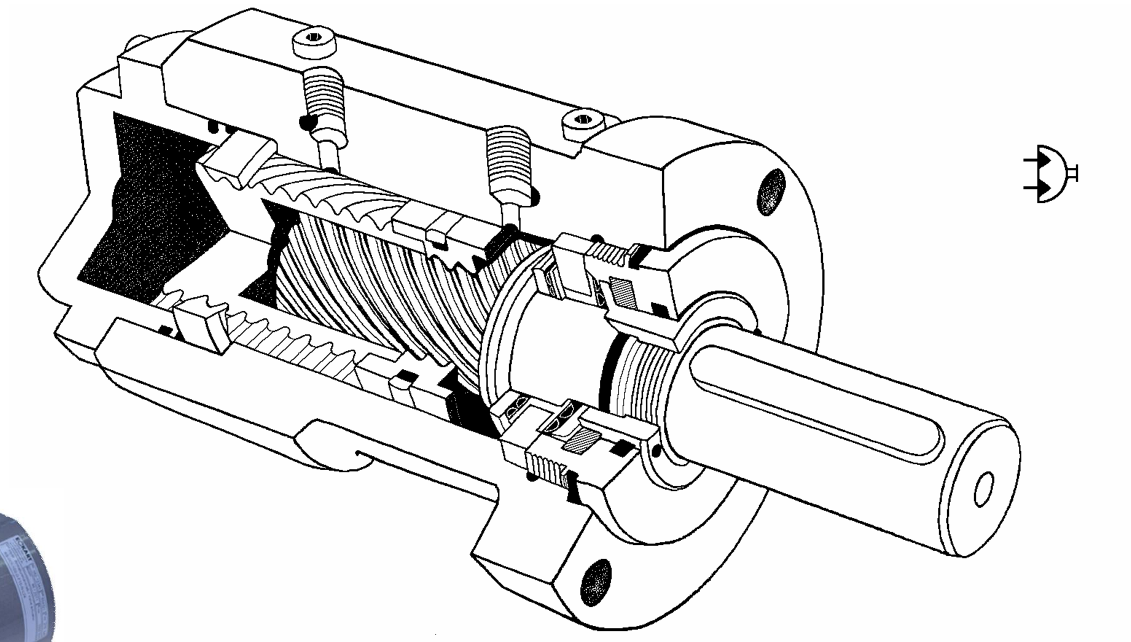
Gear type:

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%

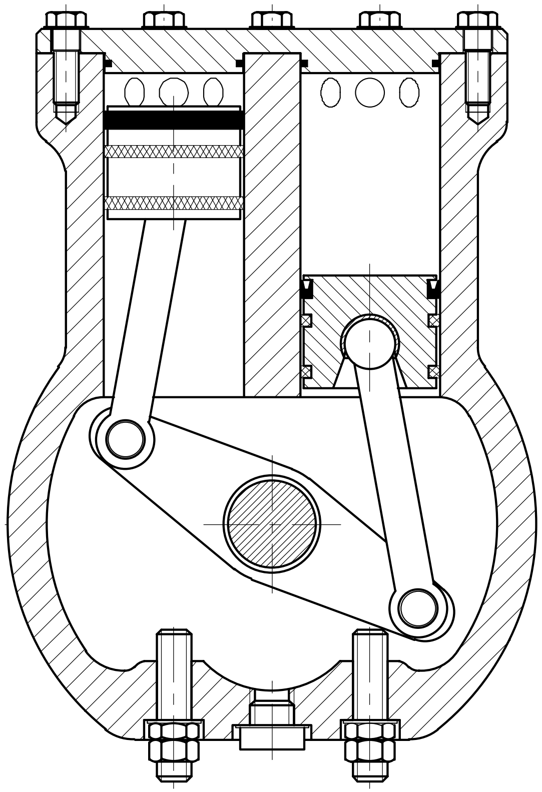


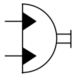
Screw gear
Crank mechanism
Cogwheel gear

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

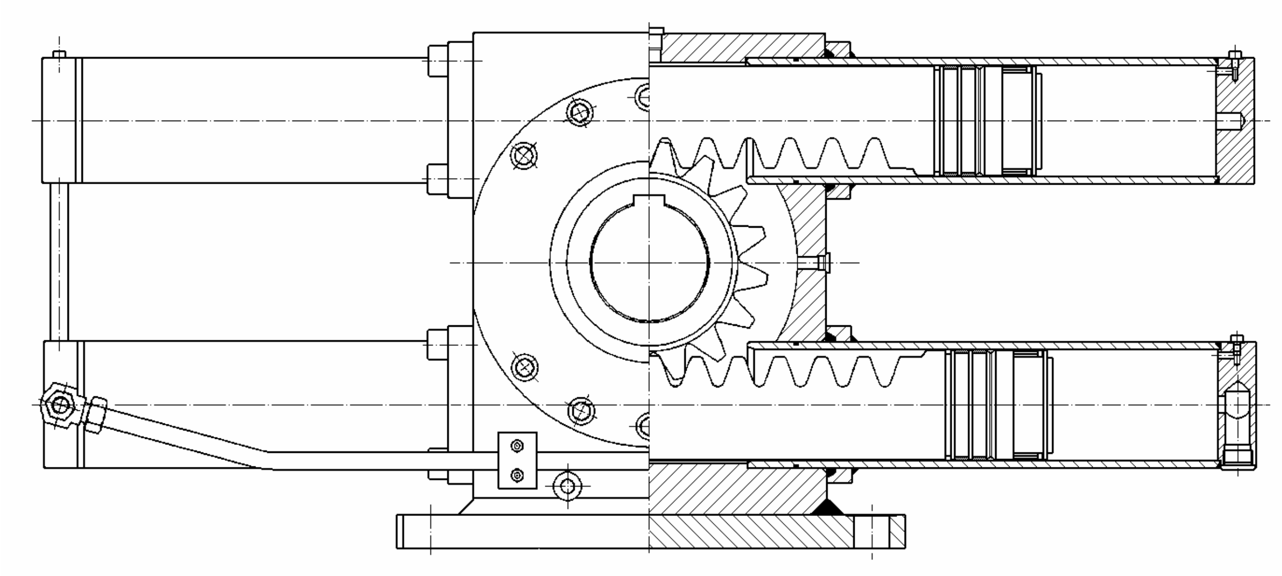




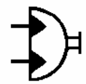
Torque motors

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

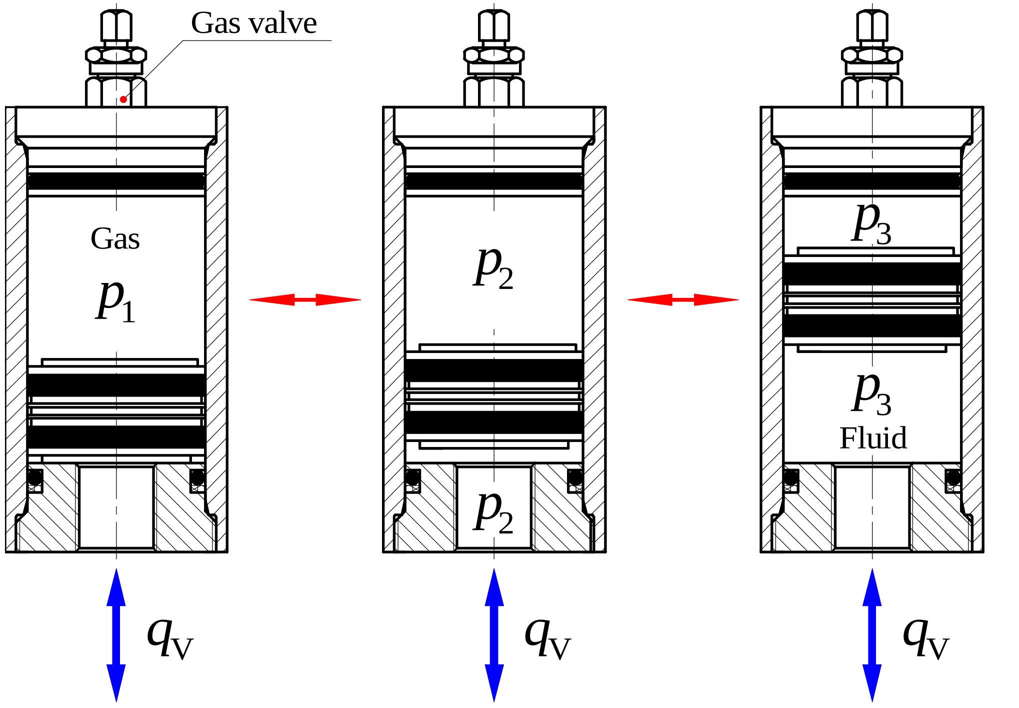
Accumulators are superior **power sources**

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))

Construction and characteristics

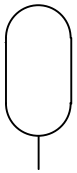
Construction types:

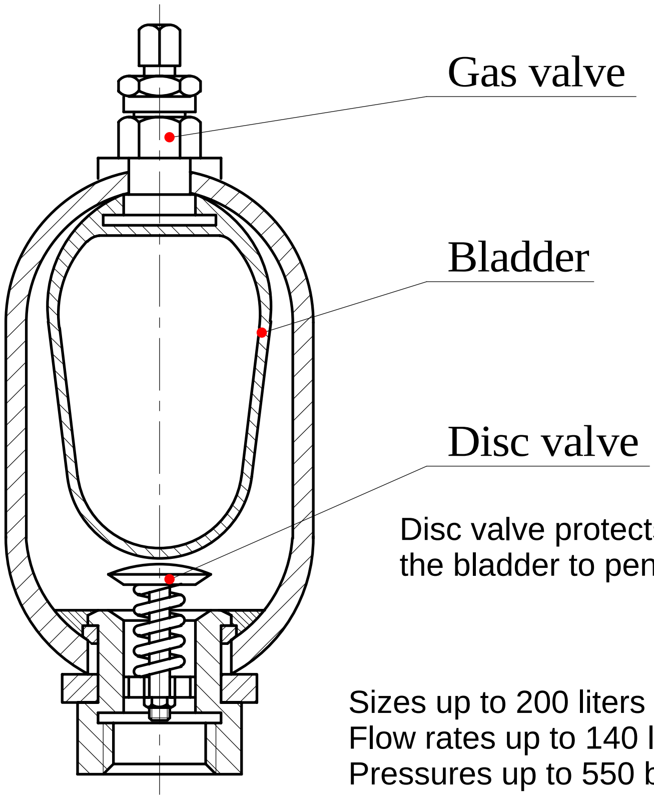
- bladder
- diagphram
- piston

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

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

Gas inside the bladder
Fluid outside



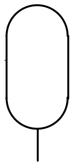


Disc valve protects the bladder and prevents the bladder to penetrate out

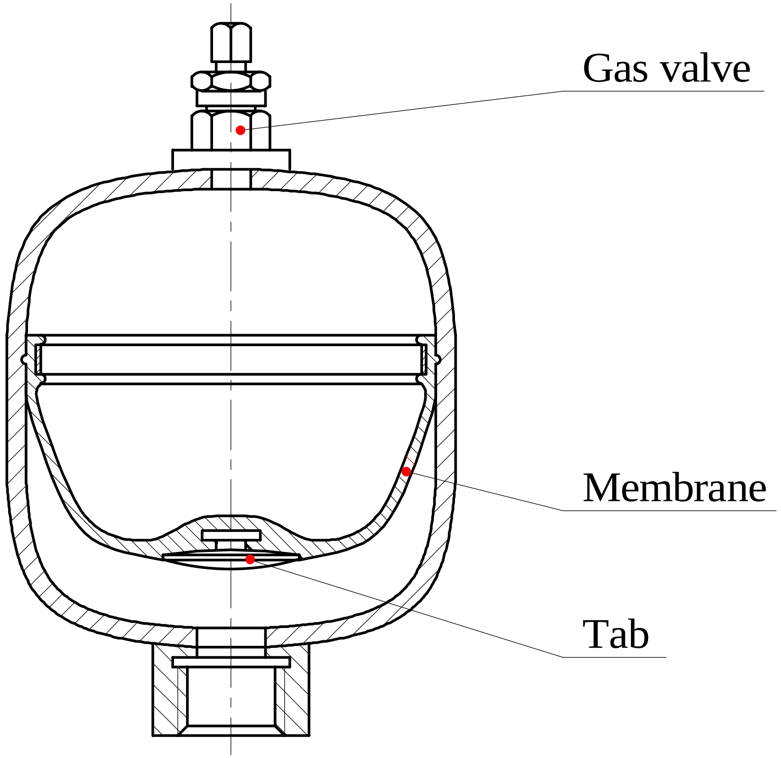
Sizes up to 200 liters
Flow rates up to 140 l/s (!)
Pressures up to 550 bar

Bladder
Diagphram
Piston

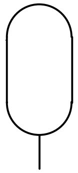
Membrane accumulators are often used for damping of hydraulic oscillations
Sizes up to 5 liters
Flow rates up to 140 l/s (!)
Pressures up to 500 bar



Bladder
Diagphram (aka Membrane)
Piston



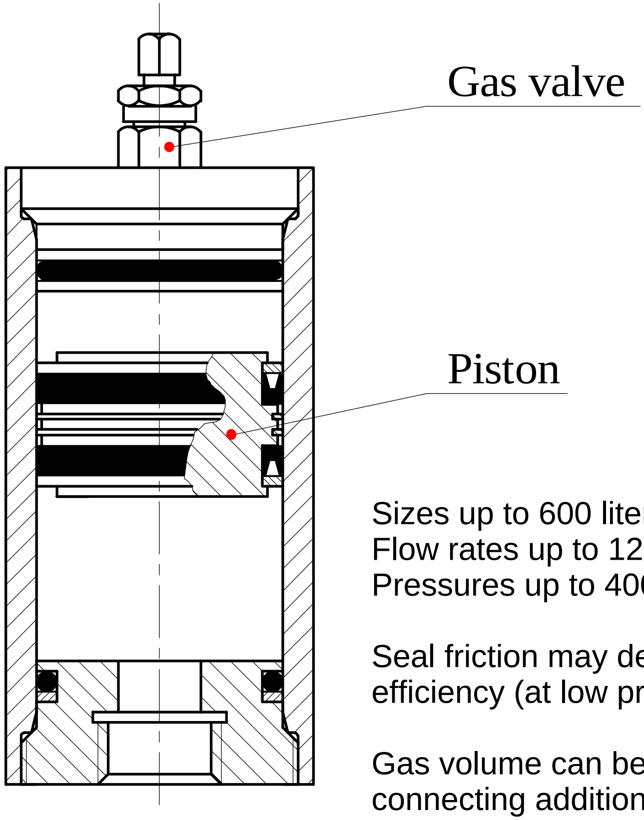
https://www.hydroll.com/fi/



Bladder

Diagphram

Piston



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

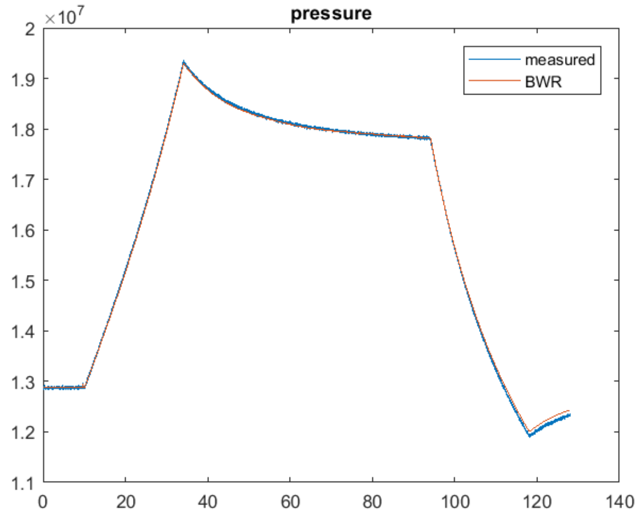
Seal friction may decrease the efficiency (at low pressures)

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

Application examples

compression - storage (1 minute) - expansion

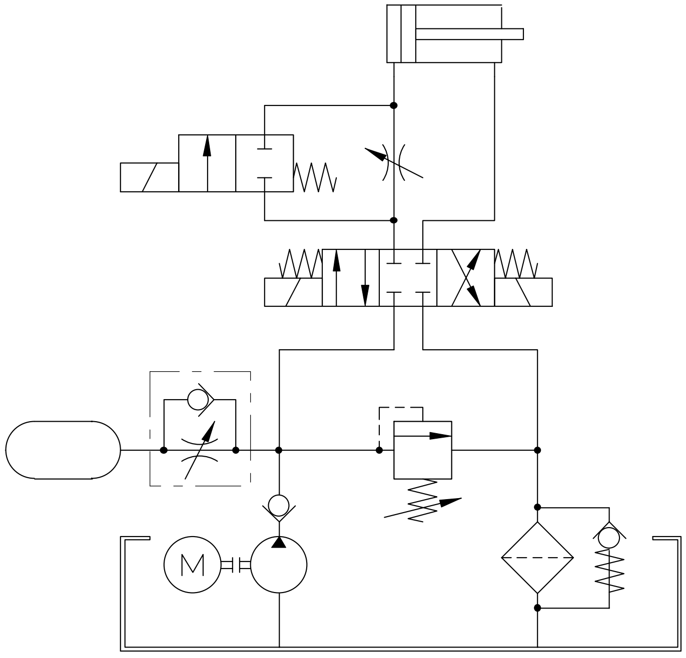
Flow source
Upkeep of pressure
Levelling of flow fluctuation
Suppression of pressure shocks
Energy storage for exceptional situations
Storing of external energy
Levelling of volume changes
Levelling of shock-like loadings of actuators



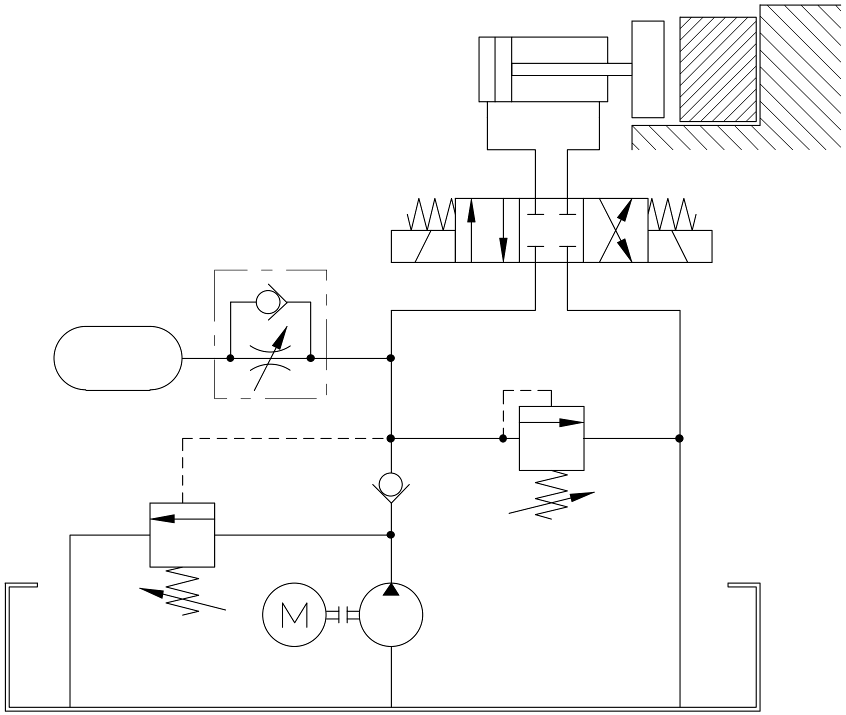
Equation of state for ideal gas can be used for simple calculations
Real gas equations for accurate results (especially at high pressures)

Measured and simulated pressure response indicates the effects of thermal losses especially during storage phase

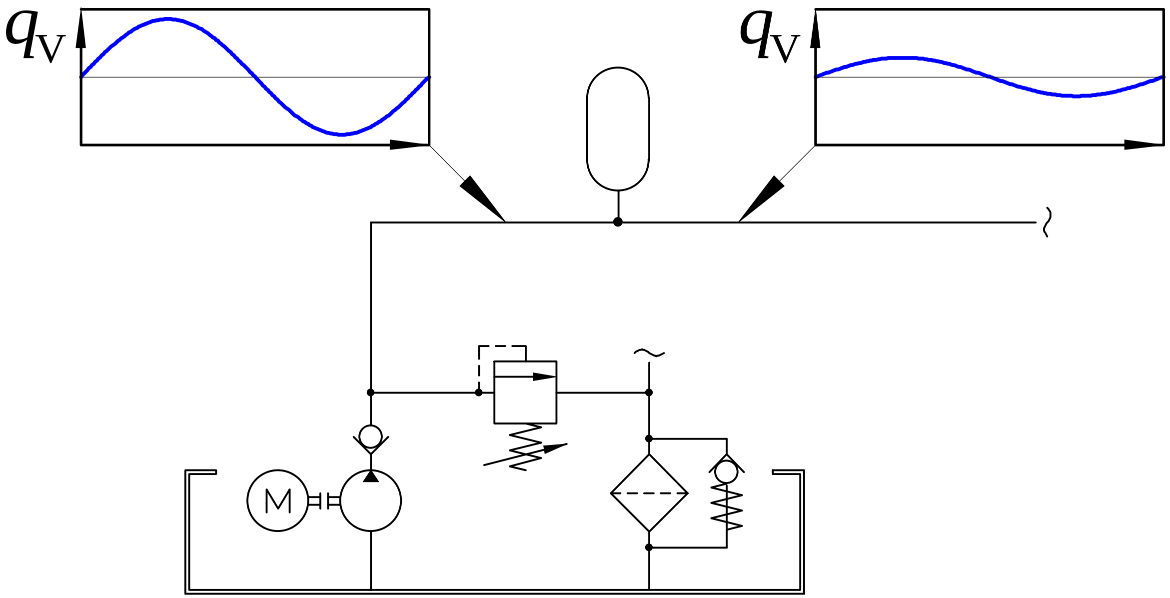
Flow source



Upkeep of pressure

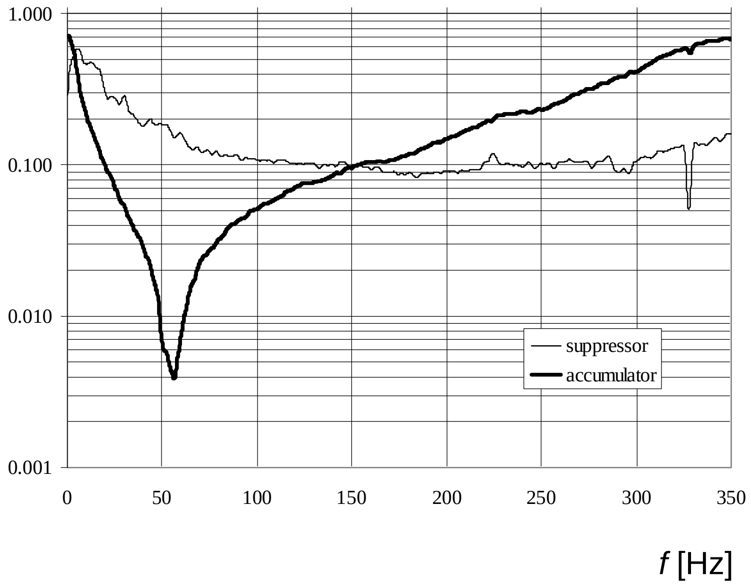


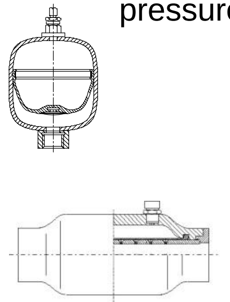
Levelling of flow fluctuation



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



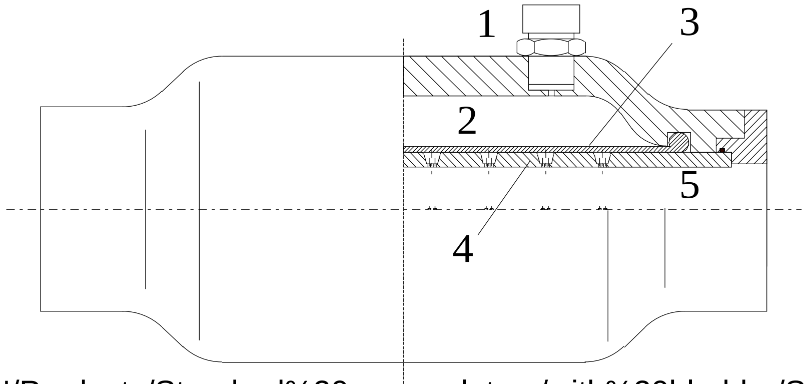


Example:

Levelling of flow fluctuation

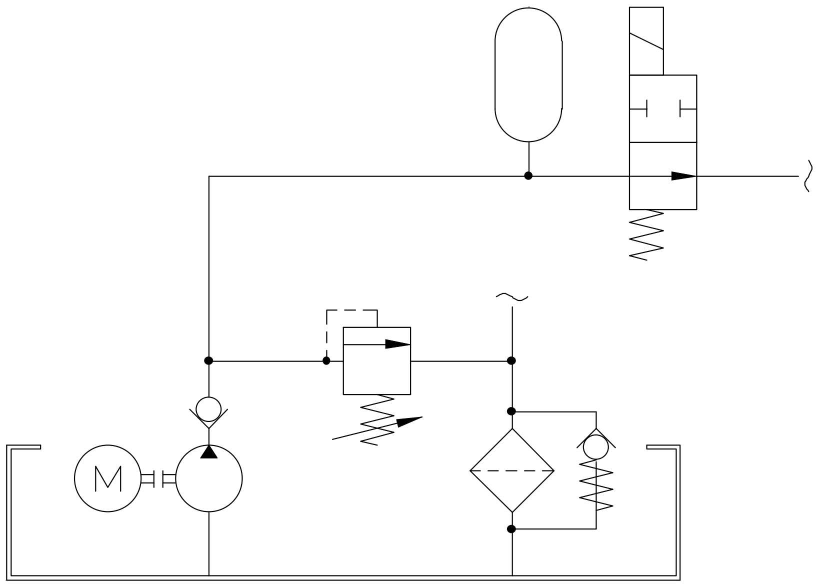
Special accumulators "pulsation and noise dampers" or "inline suppressors" have wider frequency range

Inline suppressor ®

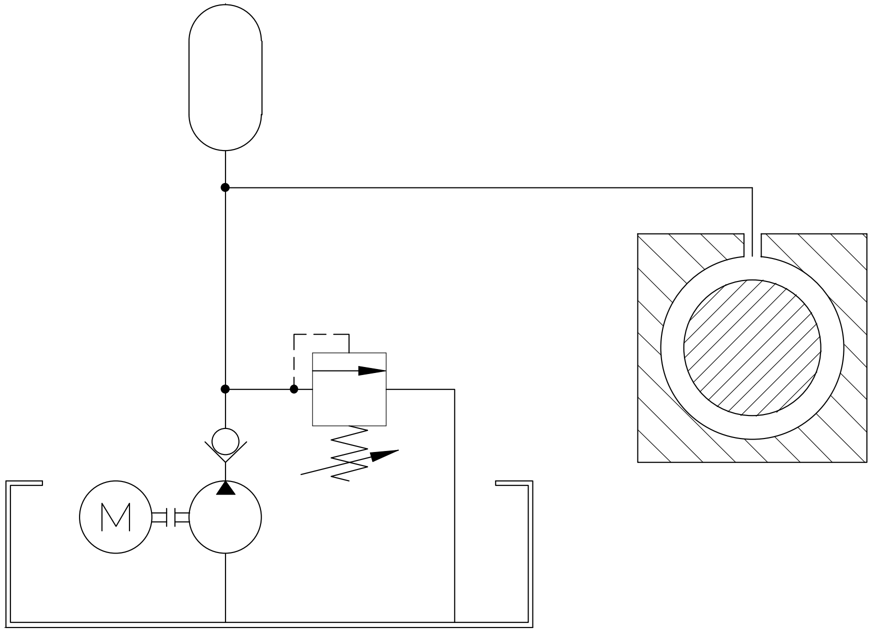


http://www.saip.it/EN/Products/Standard%20accumulators/with%20bladder/SPM%20type

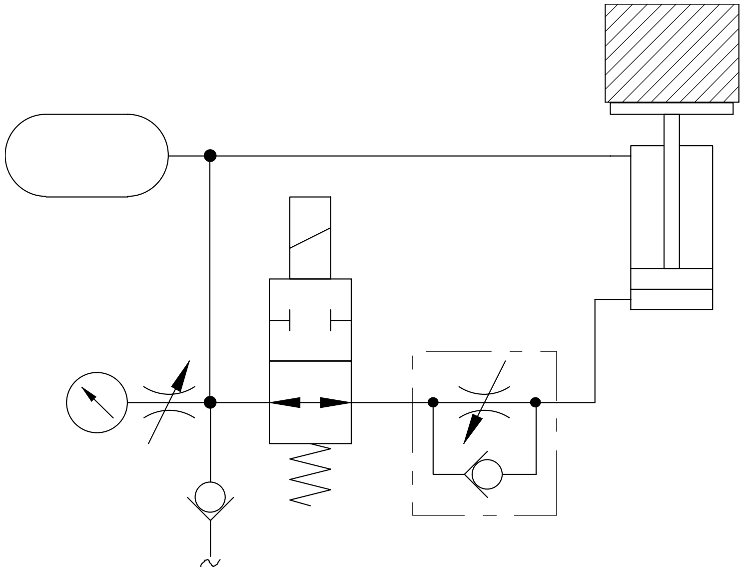
Suppression of pressure shocks



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?