Speed control

for reciprocating compressors

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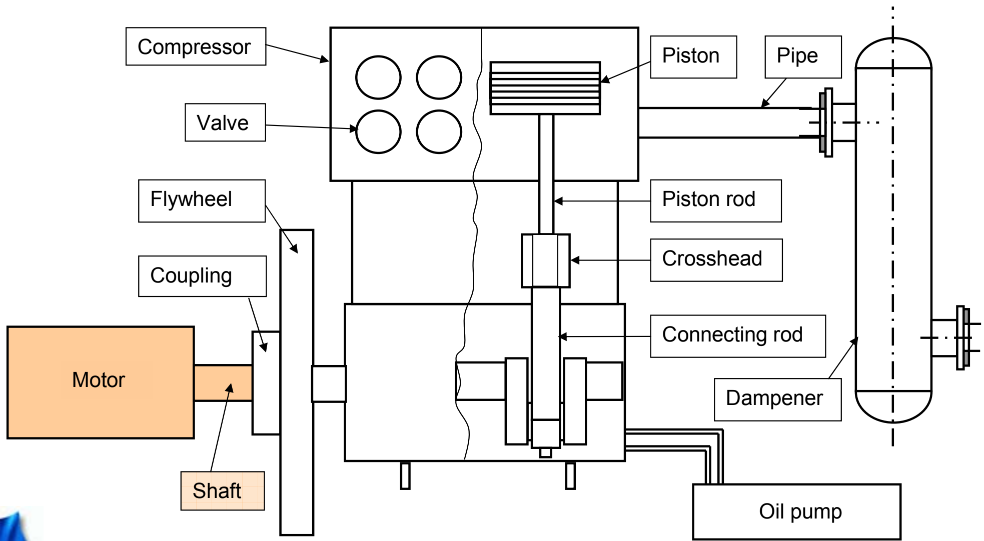
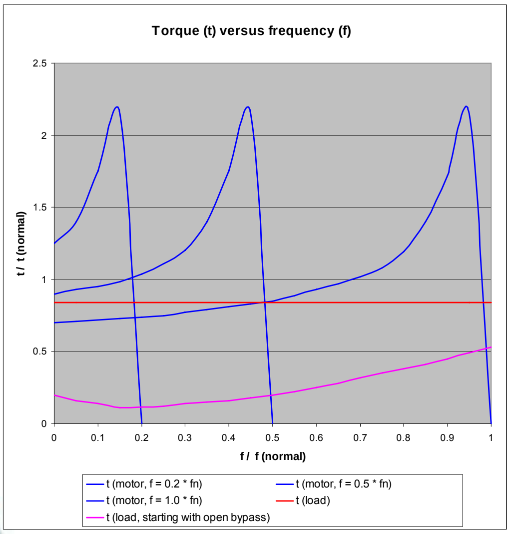
Most common VSDs (variable speed drives)



Most common VSDs (variable speed drives)

Frequency Converter and Alternatives



Alternatives:

- Altering the slip (for low power demand, poor efficiency)
- Pole changing (speed-changing stepwise)
- Mechanical variable speed control (belt drives, moving conical pulleys)
- Non-electrical drive (speed-variable combustion engine, turbine)

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•Frequency converter:

•Dimensionless notation (t / t_{normal}, f / f_{normal})

•Most usual way of changing the speed (step-less) in the reciprocating compressor sector

•A change in the frequency causes a proportional change in the speed

•Alternatives:

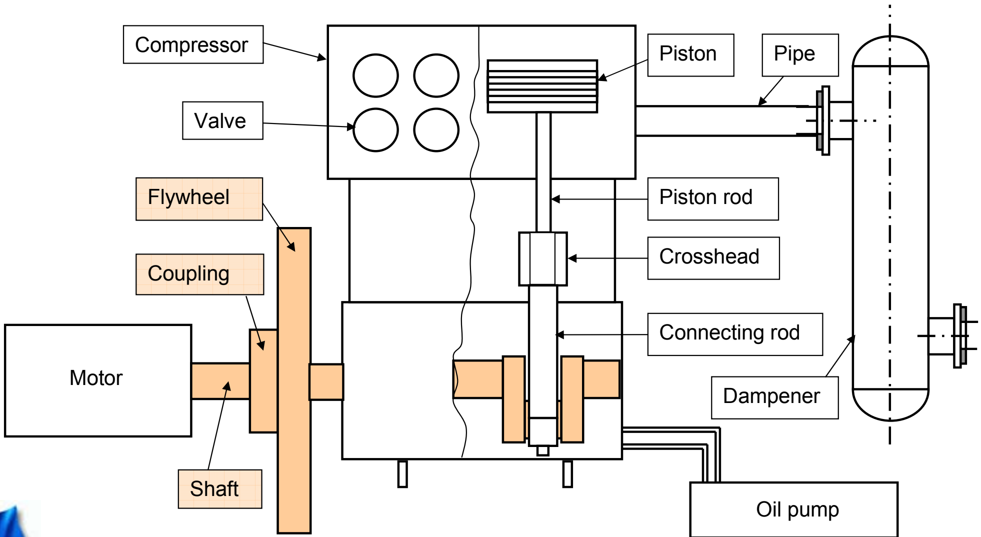
•Altering the slip
by additional resistors, (for low power demand < 50kW) → heat losses, poor efficiency

-Pole changing
Different numbers of poles in the stator, changing of speed only in a few steps possible (usually 2…4 steps)

-Mechanical variable speed control
moving conical pulleys in belt drives (belt drives sometimes used at reciprocating compressors for power consumptions < 150 kW), (but conical pulleys not usual)

-Non-electrical drive (speed-variable combustion engine, like diesel engine; or turbine → run at higher speed regions)

Torsion analysis and Coupling

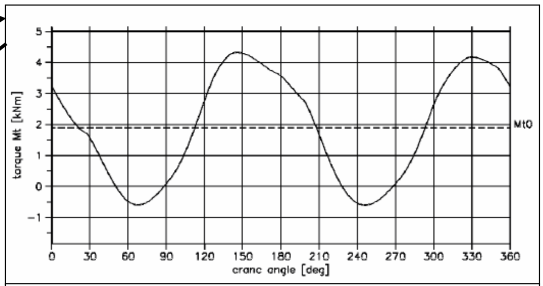


Torsion analysis and Coupling

Torque Diagram

Maximum torque

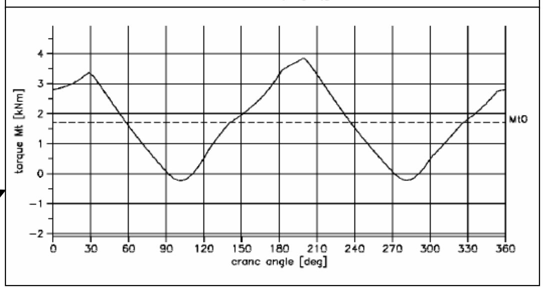
Operating condition: *ps-max, 600rpm*



Fourier-Analysis Mt=Mt0+SUM[Mt(j)*COS[j*ABS(w)*t-phi(j)]]		
ord. j	amp. Mt(j) [kNm]	phi(j) [deg]
1	0.123	165.3
2	2.404	315.3
3	0.046	205.2
4	0.226	151.0
5	0.105	258.8
6	0.117	119.9
7	0.068	275.3
8	0.060	286.2
9	0.018	308.3
10	0.010	258.7
11	0.014	8.5
12	0.021	332.8

Maximum irregularity

Operating condition: *design, 300rpm*



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•Typical torque diagrams which have to be taken in consideration when calculating a torsion analyses

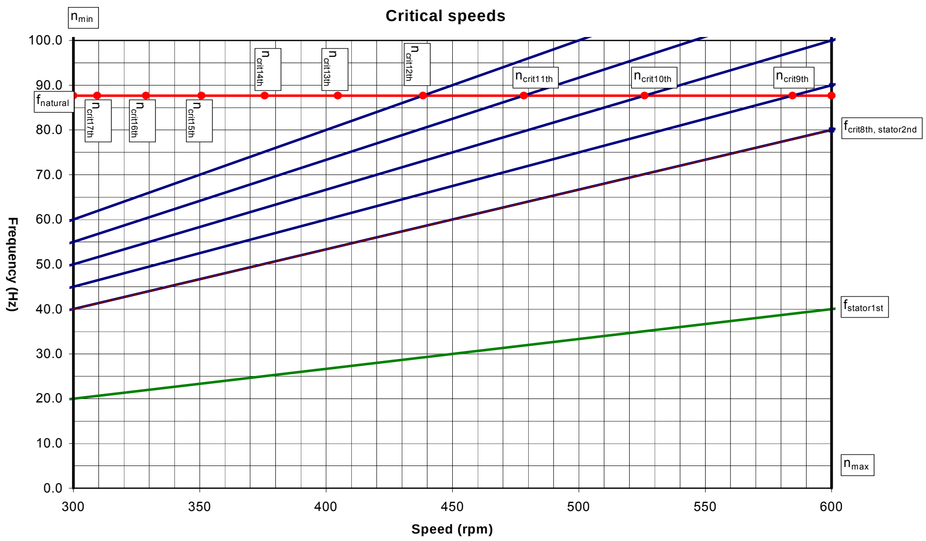
•Case with maximum torque at high speed → Evaluation in a Fourier-Progression

•case with maximum irregularity at low speed → Evaluation in a Fourier-Progression (not shown)

•Dominant order here: 2nd order; higher amplitudes at “maximum torque”

Torsion analysis and Coupling

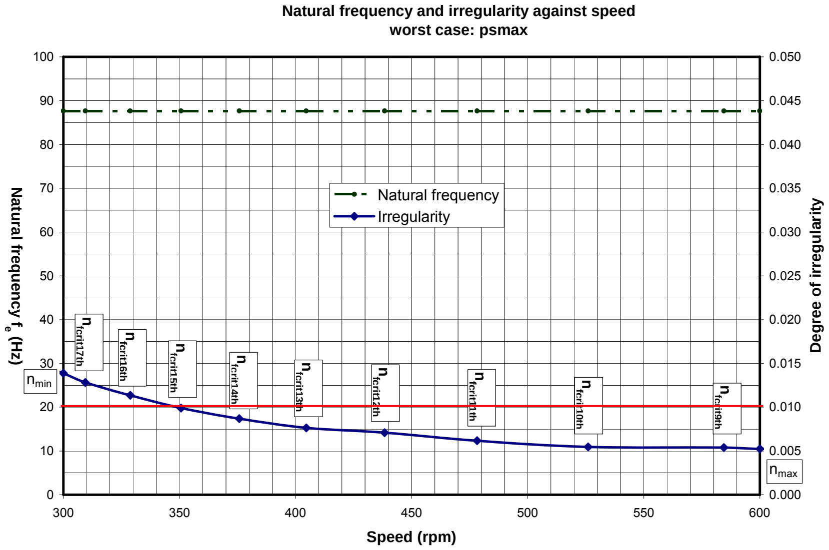
Critical speeds (example rigid coupling)



- Campbell diagram
- Rigid coupling → flange
- High natural frequency (in comparison to flexible couplings)
- Frequencies of higher orders at high speeds meet the natural frequency → critical speeds
- When critical speeds cannot be avoided, amplitudes must be low at these orders

Torsion analysis and Coupling

Degree of irregularity (example rigid coupling)

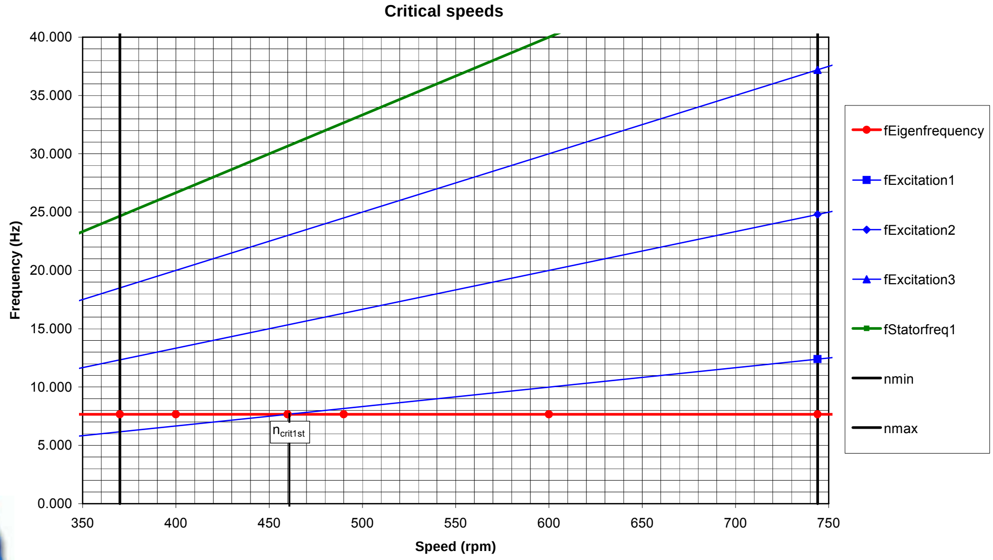


•Degree of irregularity
- •irregularity of angular speed

•Degree of irregularity should be smaller than 1/100 (often used limit coming from power supply)
- •At 350 rpm the curve crosses 1/100 irregularity
- •Frequencies higher than order 15 → relating critical speeds → high degree of irregularity

Torsion analysis and Coupling

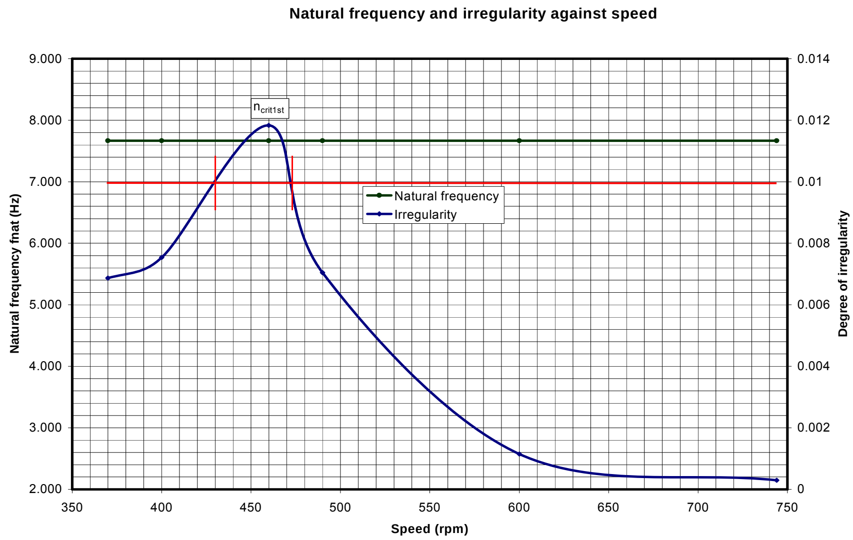
Critical speed (example highly flexible coupling)



- Campbell diagram
- Highly flexible coupling → rubber discs
- Low natural frequency
- First excitation frequency can meet the natural frequency at low speed
- $n_{critical}$ is at 460rpm
- Amplitude of first order can be too high

Torsion analysis and Coupling

Degree of irregularity (example highly flexible coupling)

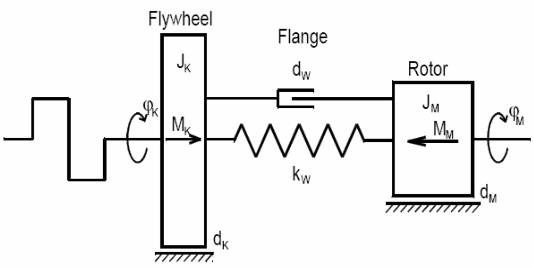


- Due to resonance between 420 and 472 rpm the irregularity is higher than 1/100, the critical speed is in that region
- Possibly this region of speed should be avoided (controlled in the frequency converter)

Torsion analysis and Coupling

Reaction of the torsion system on mass ratio

- Due to the mass ratio a drive (including a flywheel) reacts more sensitive to the excitation coming from motor side than from compressor side
- Generator operation (negative torque) can be problematic, due to sudden change of force direction
- Natural frequency of a system can be changed by changing the inertia of flywheel J_K or motor J_M or by changing the coupling rigidity



Example:

Power consumption of compressor 1400 kW, speed range 200 … 450 rpm

Momentum of inertia:

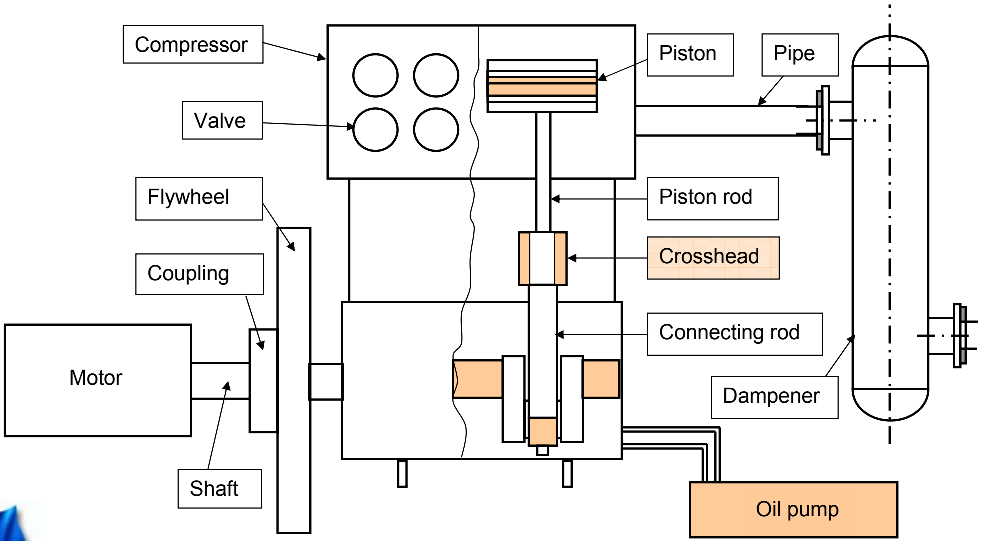
J_M = 470 kgm^2, J_K = 1325 kgm^2

•Equivalent model for an oscillatory torsion system: spring - absorber system

•Index K for compressor side,
Index W for the rigid coupling, flange
Index M for the motor side

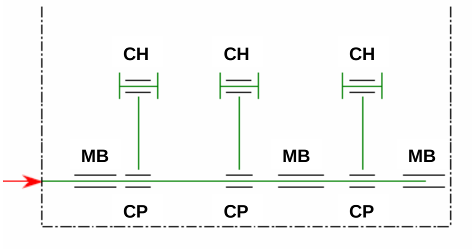
•Limitation in increasing the flywheel size

Lubrication



Lubrication

Lube oil system



Lubricated parts:

- Main bearings at crankshaft
- Crank pin
- Crosshead, cross head pin
- Piston and packing lubrication

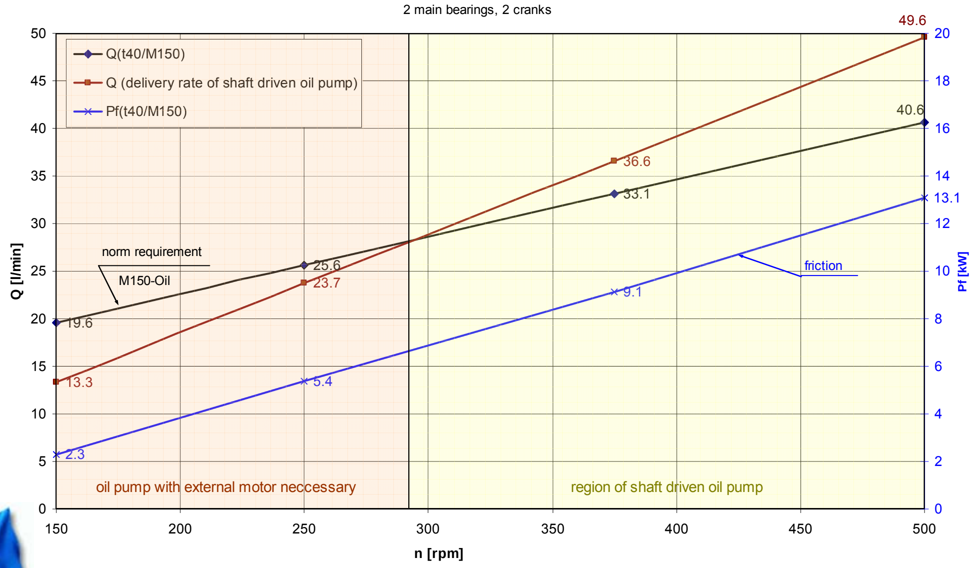
Main components of a lube oil system:

- Lube oil pump
  (shaft driven or with external motor)
- Bypass and bypass valves, safety valves (for oil pressure regulation)
- Lube oil cooler
- Possibly lube oil heater
- Lube oil filter

•Cross head slide block and cross head pin

Lubrication

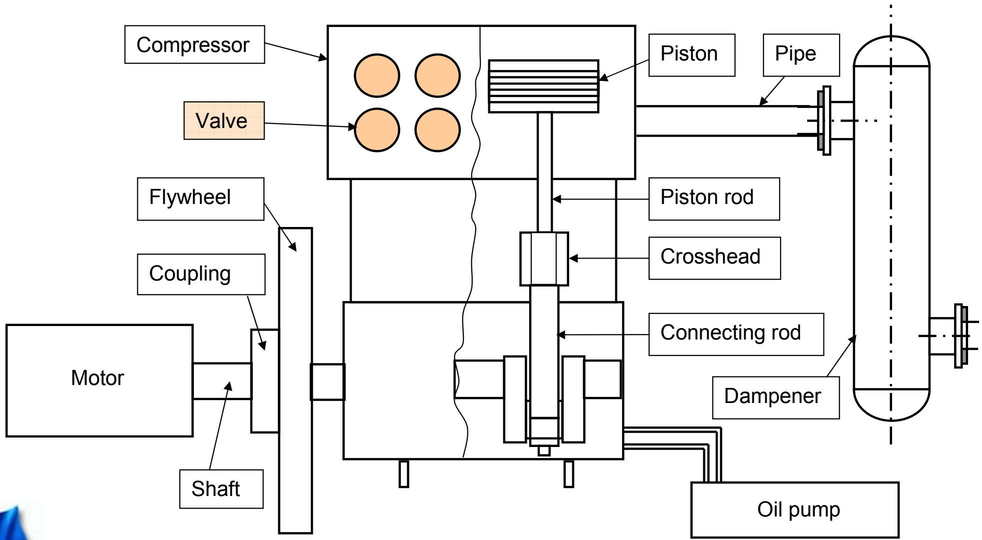
Lube oil requirement for slide bearings



•Friction Pf [in kW] is proportional to speed (as well as to force, diameter of bearing and kinetic coefficient of friction μ)
•Total lube oil requirement Q [in l/min]
- •depends on oil pressure, bearing clearance, speed
- •as well as temperature and viscosity, thus on oil type

•Clarifications:
- •M150 → ISO Viscosity Grade
- •t40 → Temperatures in °C

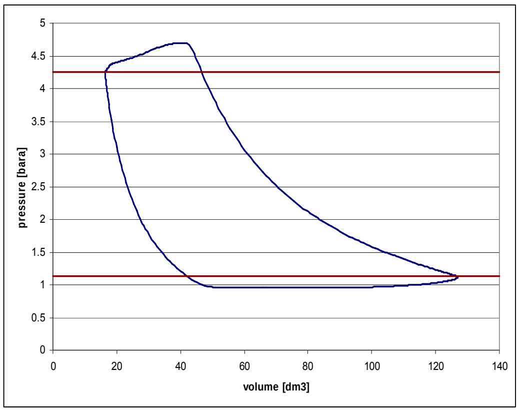
Compressor Valves



Compressor Valves

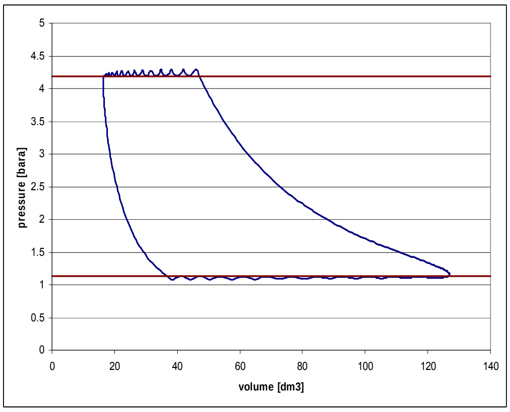
Pressure characteristic in the compression chamber

max. speed 500 rpm



Upper dead centre 90°; lower dead centre 270°

speed 100 rpm



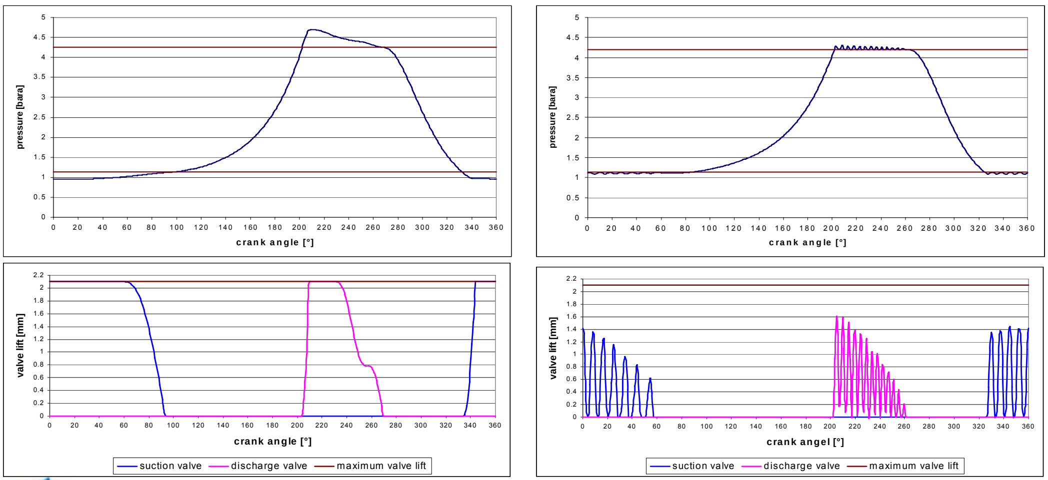
Upper dead centre 90°; lower dead centre 270°

- Lines of suction and discharge pressure
- Over-compression at maximum speed → power losses
- Valve fluttering at low speed → valve damage

Compressor Valves

Pressure characteristic & Valve lift vs. Crank angle

max. speed 500 rpm / ***speed 100 rpm***



Upper dead centre 90°; lower dead centre 270° *Upper dead centre 90°; lower dead centre 270°*

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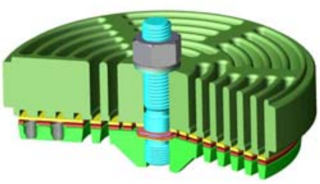
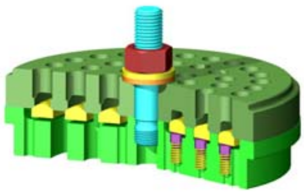
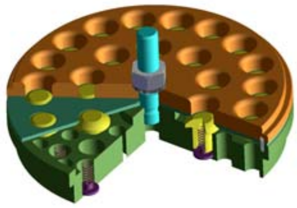
•Example: plate valve, maximum valve lift 2.1 mm (line in valve lift diagrams)
•Valve design at maximum speed
  •Gas velocity in valve 33 m/s (valve stage 1)
  •Power losses quite high at maximum speed 14.2% (power losses of all valves compared with power consumption)
•Valve fluttering at minimum speed
  •Gas velocity in valve 7 m/s (valve stage 1)
  •Power losses very low at minimum speed 2.1% (power losses of all valves compared with power consumption)

Compressor Valves

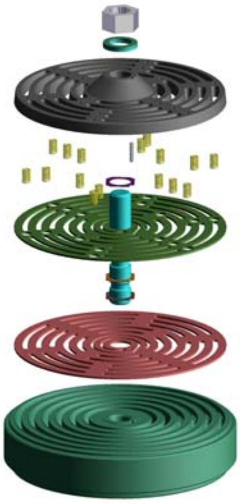
Different valve types similar problems

Plate Valves	Ring Type Valves	Valves with Poppets
		
		

- Design the valve springs at maximum speed
- Prevent problems at minimum speed for different valve types

Compressor Valves

Details on Plate Valves



- Prevention of fluttering at low speed by:
  - •Belleville damper disc
  - •Variable valve lift
  - •Variable spring bias

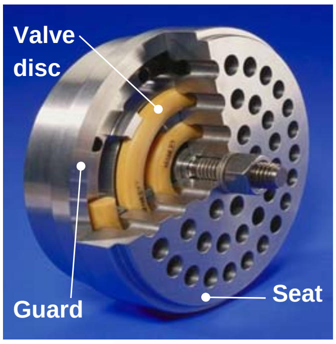
•Valve design on maximum speed
•At low speed problem: fluttering
→ Possible solutions from different valve manufacturers

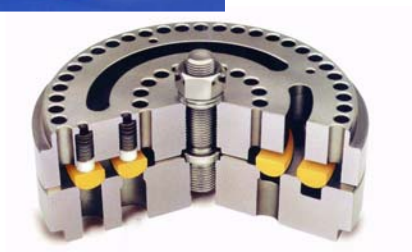
- •Belleville damper disc is conical and catches the valve plate in between the distance to maximum valve lift
- •Variable valve lift
- •Variable initial stress in the springs

Compressor Valves

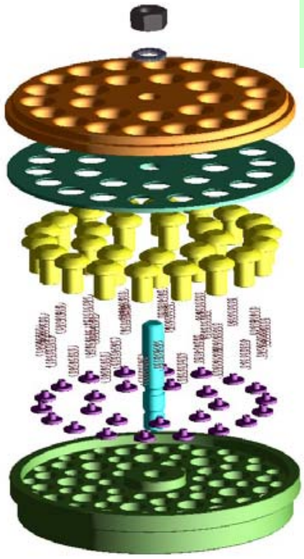
Details on Ring Type and Poppet Valves

Ring Type Valves





Valves with Poppets



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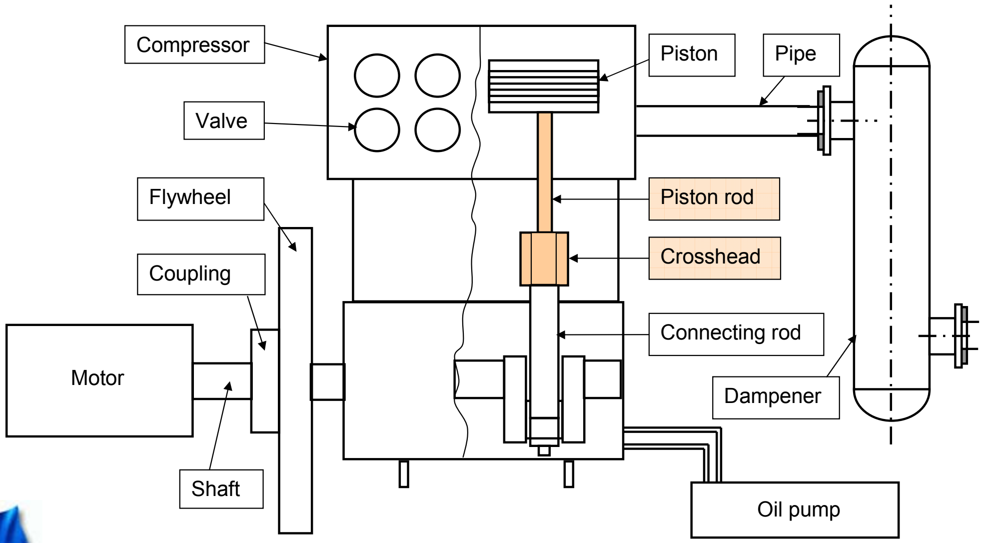
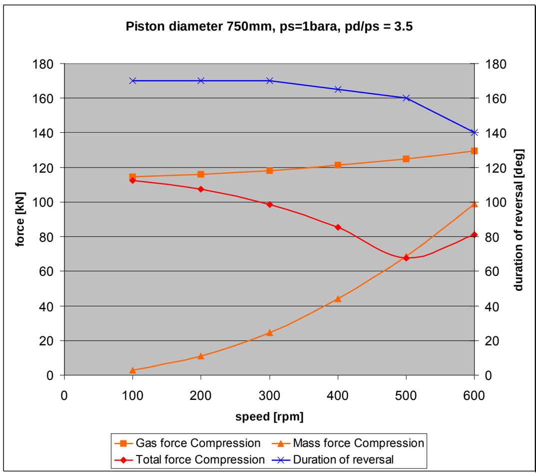
•When the gas velocity gets too small, some valve discs or poppets do not open (stochastically)
•Partial opening can be controlled by using defined, different spring sizes for the different concentric valve discs or the poppets in one valve.

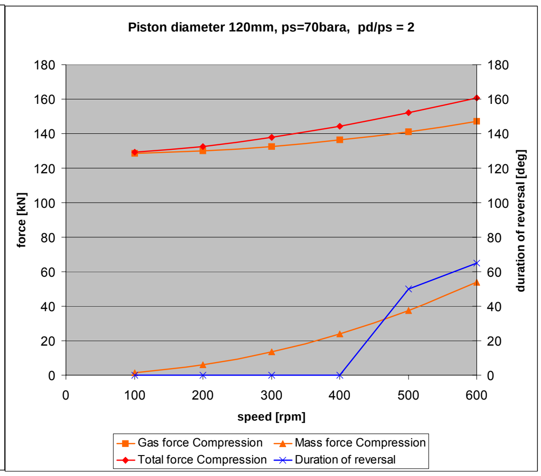
Combined rod loads and Duration of reversal



Combined rod loads and Duration of reversal

Example for large and small piston diameters





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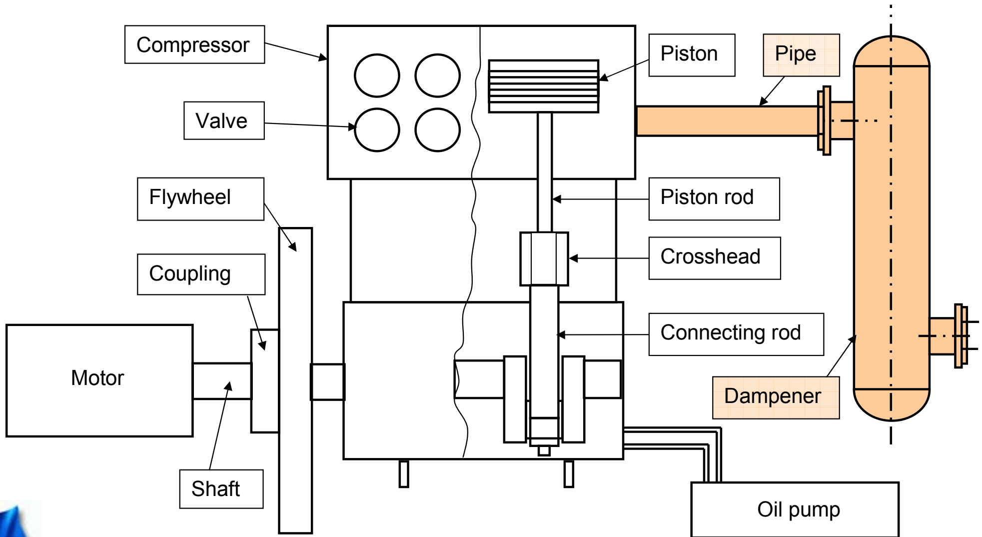
- •Large piston diameter
    - •Lower stage number (usually 1st stage)
    - •Low suction pressure level
    - •Quite high pressure ratio
- •Small piston diameter
    - •Higher stage
    - •Quite high suction pressure level
- •(example calculated with Ethylene)
- •Smaller mass forces with smaller speed (speed has a quadratic influence on force)
- •Reduction of gas forces due to lower gas velocities in the valves
- •Resulting total force dependent on vector addition of both forces (depends on crank angle)
- •Duration of reversal has to be controlled individually over speed range
    - •Can rise or be constant with lower speed
    - •Can fall and even drop down to zero with lower speed !
- •Difference between horizontal and vertical cylinders: influence of gravitational acceleration on the piston in common very small, can be neglected

Dampeners and Pipes



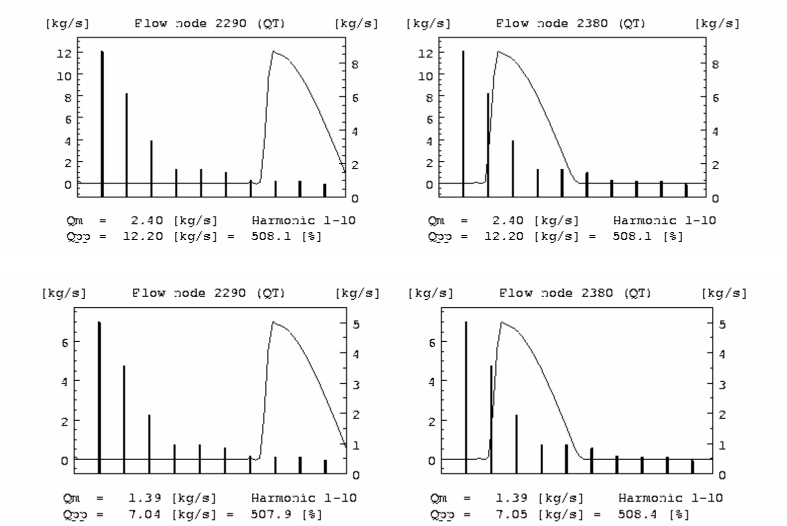
Dampeners and Pipes

Flow functions (flow pulse)

Cyl. 1, head end | **Cyl. 2, head end**

Max. speed 520 rpm

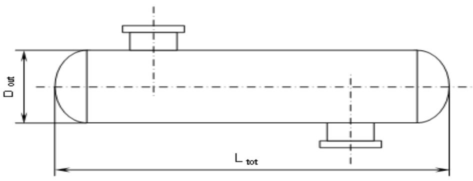
Speed 300 rpm



- Graphs
  - are print outs from the simulation program PULSIM
- reviewed nodes
  - from suction side,
  - two cylinders 1st stage head end (flow nodes 2290 and 2380),
  - (single acting machine)
- Two different speeds: 520 rpm and 300 rpm
- Curves in the graphs:
  - Time flow function for one crank revolution
  - Spectrum
- Evaluation at 520 rpm
  - 1 flow pulse per crank revolution (in degrees) from each cylinder
  - Mean 2.4 kg/s, Maximum 12 kg/s → 508% of mean
  - 10 orders (harmonics) are shown, first order is dominant order
- Evaluation at 300 rpm
  - Flow pulse smaller compared with 520rpm, (lower mass flow per crank revolution) but same shape
  - Same shape but different scale
  - Mean 1.4 kg/s, Maximum 7 kg/s → 508% of mean
  - first order is dominant order

Dampeners and Pipes

Dampener design



- Damper to be designed on maximum speed due to higher mass flow per crank revolution
- BUT: pulsation wave length shorter for higher speed → more pulses per time
- → Difference in required damper size can be small (comparing min and max speed)

Damper		S1	D1	S1	D1
Compressor speed	rpm	520		300	
No. of required dampers		1	1	1	1
Recommended damper volume	l	390.0	315.0	380.0	310.0
Sized according		PTP	PTP	PTP	PTP
Allowable PTP	%	2.00	2.00	2.00	2.00
Velocity of sound	m/s	335.6	390.0	336.7	388.6
Pulsation wave length	m	38.72	45.00	67.35	77.73
Design pressure, min. recom.	bar g	30.0	70.0	30.0	70.0
Damper outer diameter	mm	547	526	542	523
Damper wall thickness, min. recom.	mm	10	20	10	20
Damper length	mm	1879	1802	1863	1792

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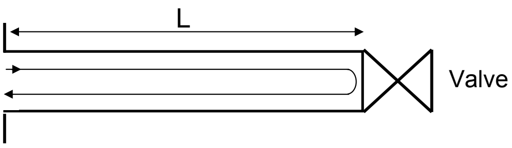
•Example: Recommended damper volumes to be within 2% ptp (peak to peak)
•Higher mass flow at max. speed → increasing dampener size
•More pulses per time (not per revolution) → Effect: can reduce the difference of the required size at min. and max. speed

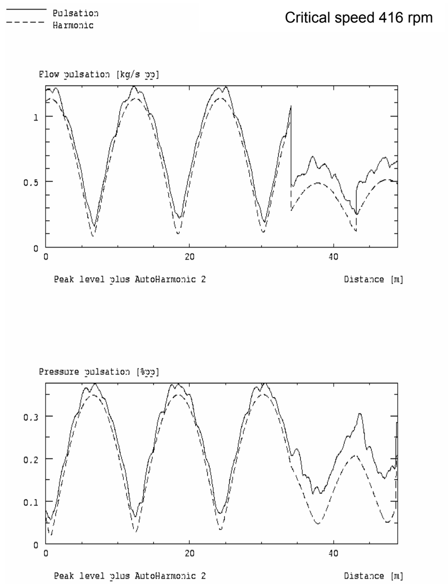
Dampeners and Pipes

Standing waves in pipes, acoustical resonance

Acoustical resonance



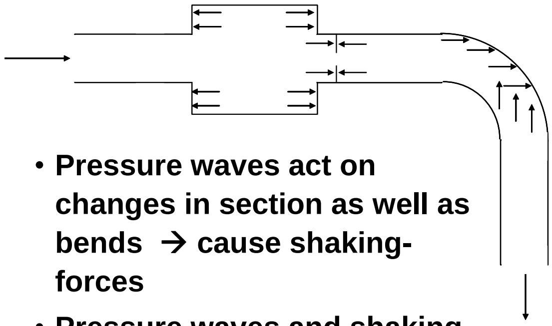
- **Length of a pipe and velocity of sound affect the frequency of the wave**
- **If the runtime of the wave is as long as the period of excitation, the system gets resonant (at a special speed) → high amplitudes**
- **At speed controlled compressors the complete range of speed has to be checked**



- Length of a pipe and velocity of sound affect the runtime of the wave (traveling time from open end to closed end and back)
- If the runtime of the wave is as long as the period of excitation (by compressor), we have resonance → high amplitudes → at a special speed
- Example: Critical speed 416 rpm

Dampeners and Pipes

Vibrations, mechanical resonance, shaking-forces



- **Pressure waves act on changes in section as well as bends → cause shaking-forces**
- **Pressure waves and shaking-forces work periodically during each crank revolution**
- **As a consequence vibrations are excited in the pipe system**

- **Critical speeds are found (existing resonance with natural frequency)**
- **For these speeds the maxima of following parameters are evaluated**
  - **Displacement**
  - **Velocity**
  - **Supporting force**
  - **Resulting stress**
- **Maxima at different nodes → evaluation at these nodes**

•Pressure waves effect changes in section as well as bends and cause shaking-forces
•Pressure waves and shaking-forces work periodically during each crank revolution
•As a consequence vibrations are excited in the pipe system including all parts (bottles, valves, flanges, supports)
•For these speeds the maxima of following parameters are evaluated

•Displacement of the pipe (should be < 1 mm)
•Velocity of vibration (should be < 30 mm/s RMS Root Mean Square)
•Supporting force and moment
•Resulting Bending Stress (should be < 45 N/mm^2)

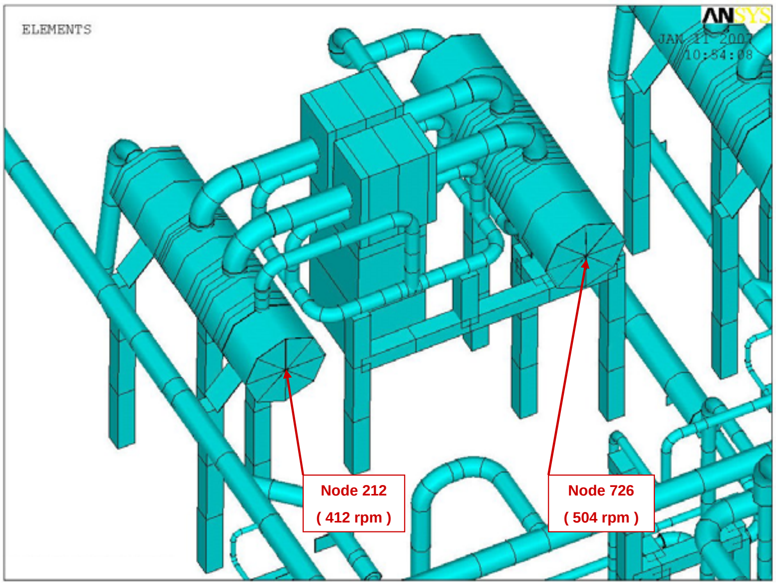
•Maxima at different nodes of the plant → evaluation at these nodes → graphs
(critical nodes of displacement and velocity are often the same)

Dampeners and Pipes

Vibrations - Displacement and Velocity

Maxima for displacement and velocity at various speeds:

- **412 rpm**
  - **Node 212**
  - **Suction dampener end**
  - **Displacement 0.038 mm**
  - **Velocity 8.50 mm/s**
- **504 rpm**
  - **Node 726**
  - **Discharge dampener end**
  - **Displacement 0.121 mm**
  - **Velocity 29.59 mm/s**



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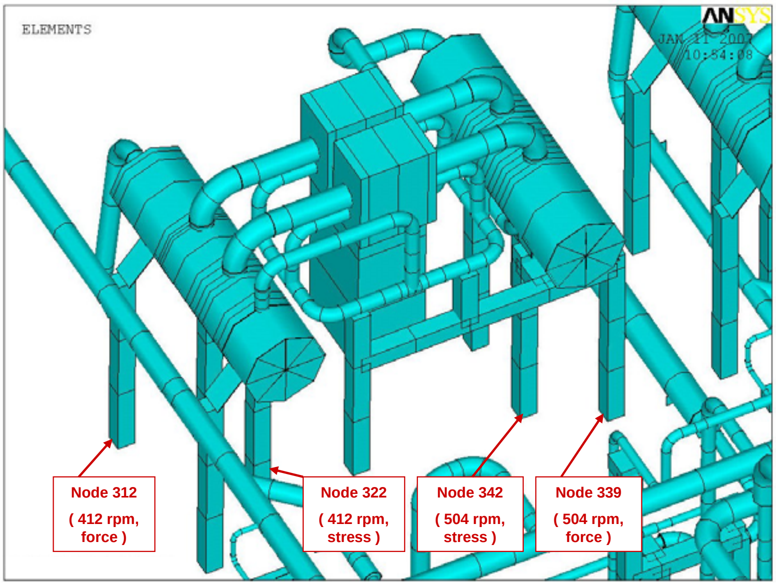
•Typical location for maximum displacement and velocity: dampener ends
•Maximum displacement → often means maximum speed (same node)
•The maxima for the different speeds are usually at different nodes → depending on resonance

Dampeners and Pipes

Vibrations - Supporting Force and Resulting Stress

Maxima for supporting force and stress at various speeds:

- **412 rpm**
  - **Maximum force at node 312: 584 N**
  - **Maximum stress at node 322: 6.54 N/mm²**
  - **Maxima at support feet of suction dampener**
- **504 rpm**
  - **Maximum force at node 339: 2926 N**
  - **Maximum stress at node 342: 4.63 N/mm²**
  - **Maxima at support feet of discharge dampener**



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- Typical location for maximum force and stress: support feet
- 412 rpm: Maxima at support of suction dampener (suction side with maxima in displacement and velocity for this speed)
- 504 rpm: Maxima at support of discharge dampener (discharge side with maxima in displacement and velocity for this speed)

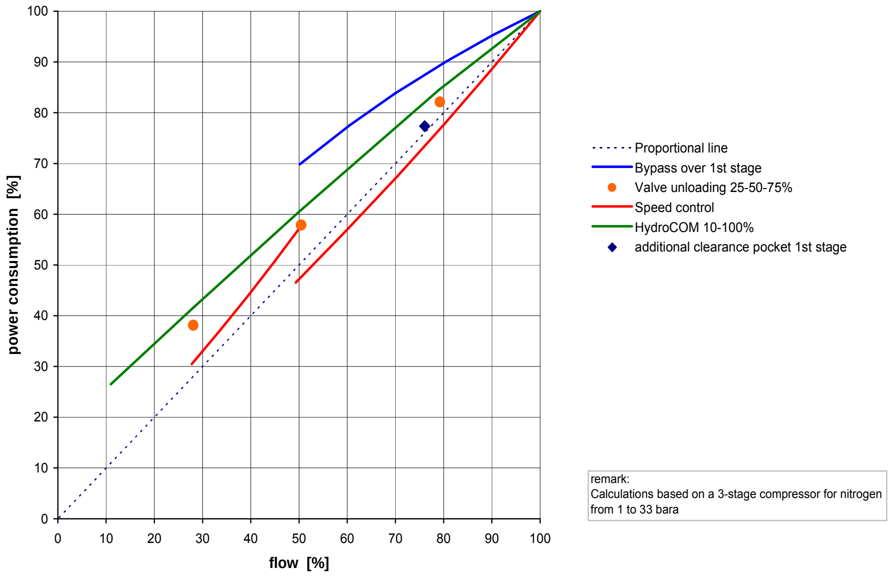
Summary - Speed control
Advantages and Disadvantages

- Step-less variable speed drive without additional equipment at the compressor, easy maintenance
- Pressure ratios and inter-stage pressures (p-V-diagrams in principle) remain unchanged
- Power consumption changes accordingly to flow and speed
- Energy efficient solution
- Less wear at piston and packing rings

- Electrical VSDs are expensive compared with fixed speed drives for the same power consumption
- Complete speed range has to be checked on critical speeds concerning pulsations, vibrations and torsion analyses → possibly some speeds have to be forbidden for longer operation
- Shaft driven lube oil systems limit the minimum speed → external lube oil systems can be necessary
- Combined rod loads have to be checked also at low speeds, due to the mass force reduction
- Not possible for multi-stream compressors

Summary - Speed control

Speed control compared with various capacity controls



•Speed control is the most energy efficient solution (better than the proportional line, saving power by less friction and less valve losses)
•Speed control can be combined with 50% valve unloading

Speed control for reciprocating compressors