Reciprocating Compressor Capacity Control

Bypass Control

Cylinder Unloading

Clearance Volume Control

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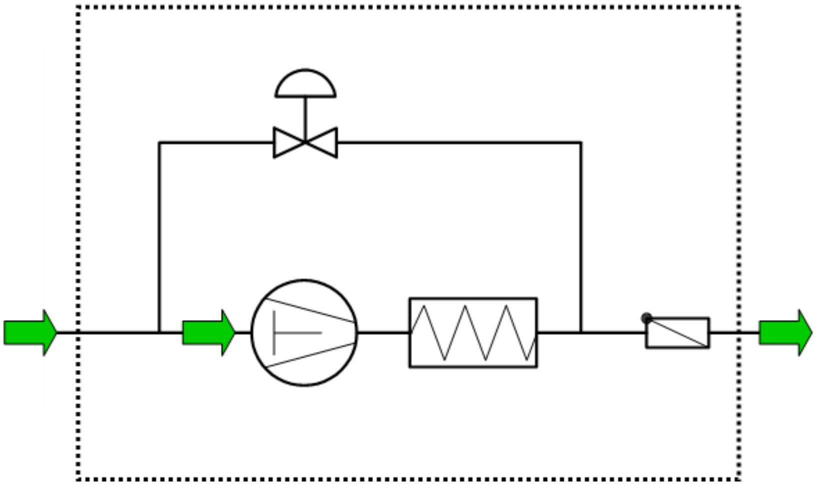
BYPASS CONTROL SYSTEMS

- OVERALL BYPASS
- STAGE BYPASS
- COMBINATIONS

BYPASS CONTROL SYSTEMS

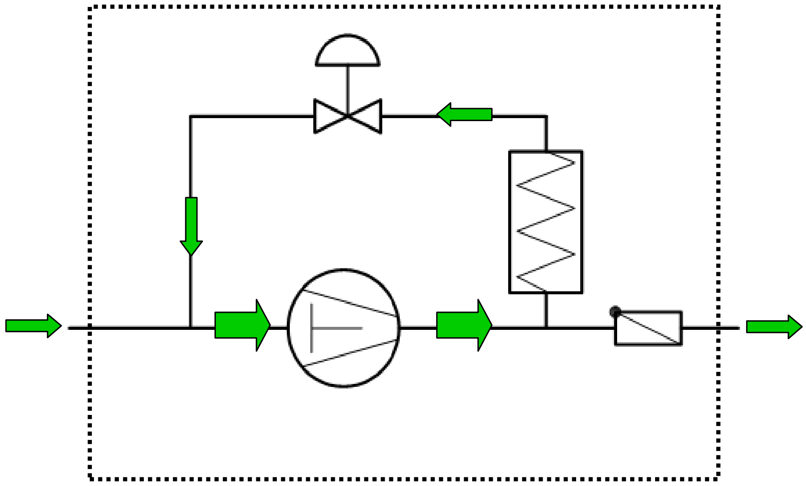
OVERALL BYPASS



Compressor with aftercooler and bypass system.

BYPASS CONTROL SYSTEMS

OVERALL BYPASS WITH EXTRA COOLER



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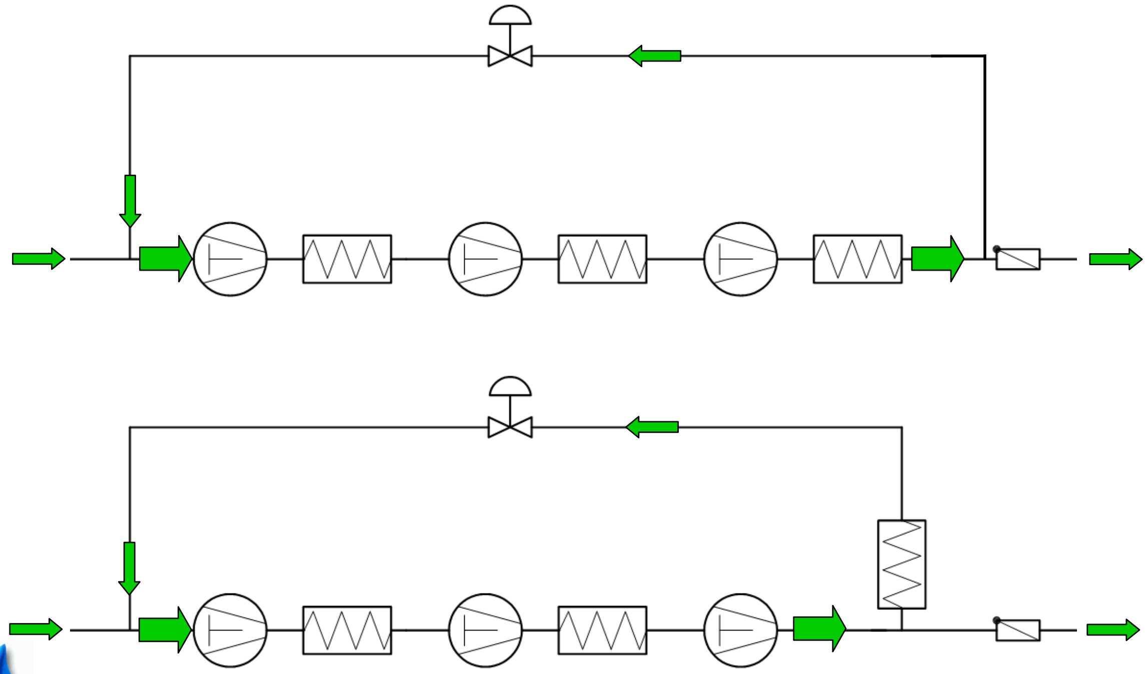
Single stage compressor without aftercooler, with bypass cooler for avoiding too high stage temperatures.

BYPASS CONTROL SYSTEMS

OVERALL BYPASS FOR MULTI-STAGE COMPRESSORS



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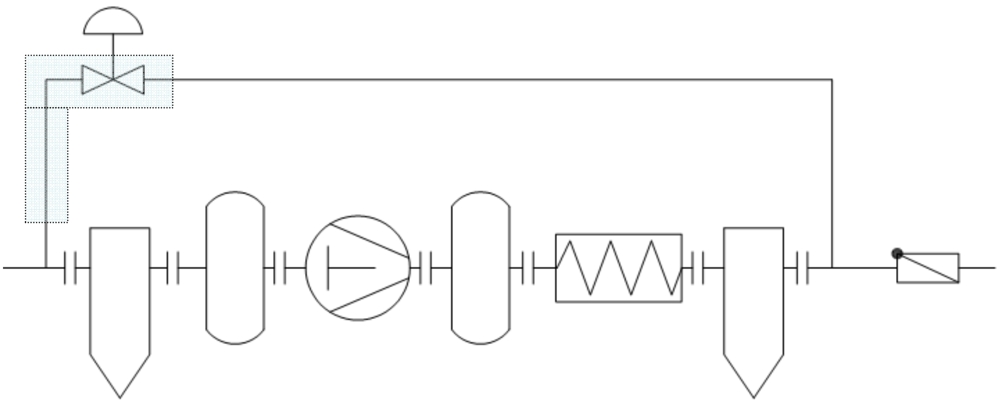
Multi-stage compressors with overall bypass

BYPASS CONTROL SYSTEMS

OVERALL BYPASS: ARRANGEMENT



The bypass line branches downstream of the last active vessel (pulsation damper, cooler, separator – whichever is last) and re-enters the inlet line upstream of the first active element.

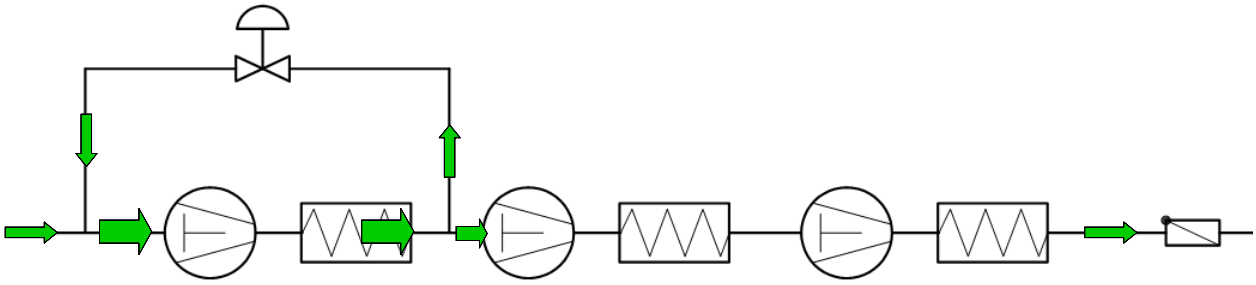
This is to avoid the recirculation of liquids, dust, and possible debris often encountered during plant commissioning.

The line dimensions shall be such that the gas velocity is kept below 3 times the respective main line gas speed.

Around the bypass valve and its downstream pipe, adequate space should be provided for insulation against noise and freezing.

BYPASS CONTROL SYSTEMS

1ST STAGE BYPASS



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Three-stage compressor with bypass around the first stage only.

This is useful for

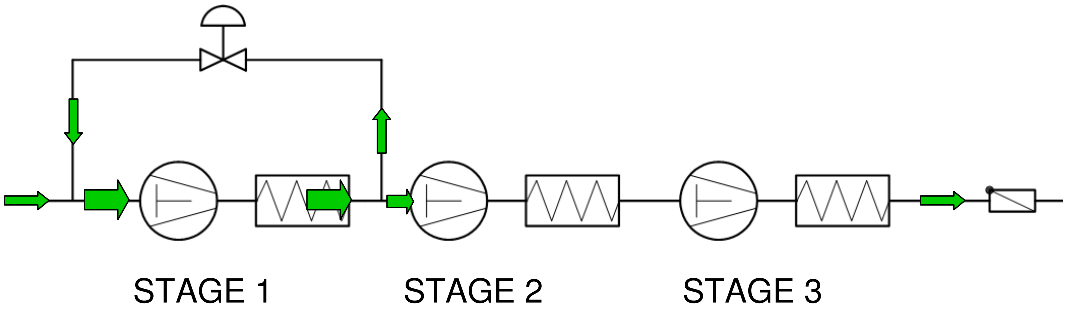
- **stepless capacity control down to ~ 70 %**
- **capacity adjustment at variable inlet pressures.**

Details on the next sheets

Bypass Example 1:

3-Stage H_2 Compressor, Constant Inlet Pressure, Variable Demand

Bypass Around 1st Stage



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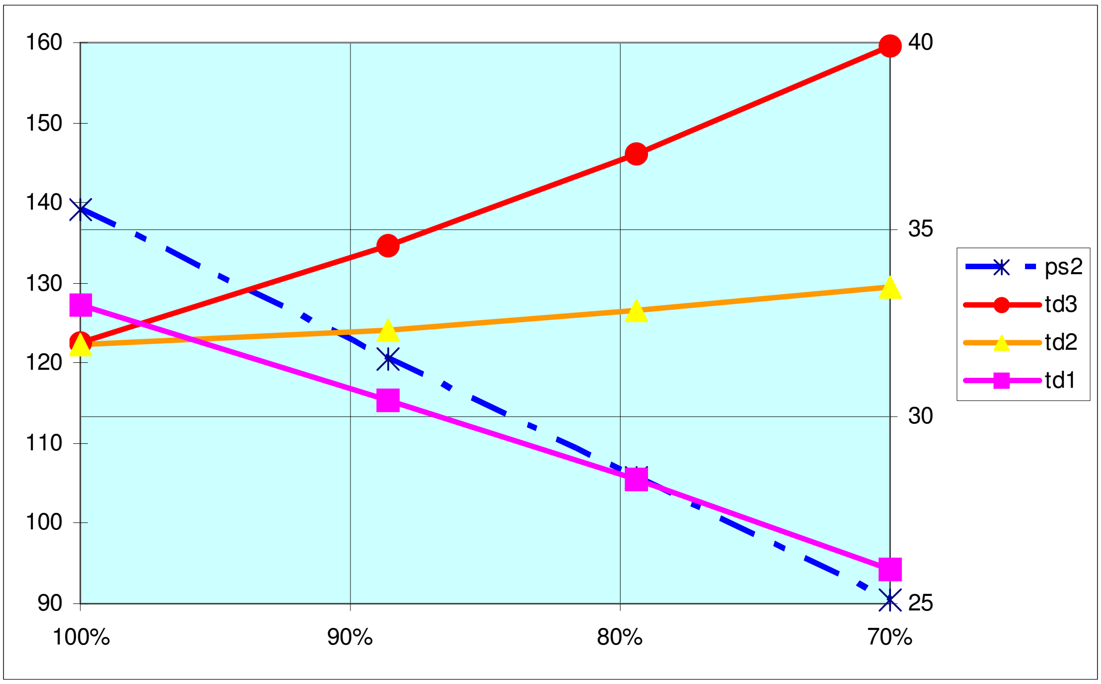
The effects of bypassing the first stage for capacity contral at constant inlet pressure are shown in terms of

- **control range**
- **effects on pressures and temperatures**
- **specific power consumption and efficiency.**

1ST STAGE BYPASS: Stage Temperatures & 2nd Stage Suction Pressure vs. Capacity



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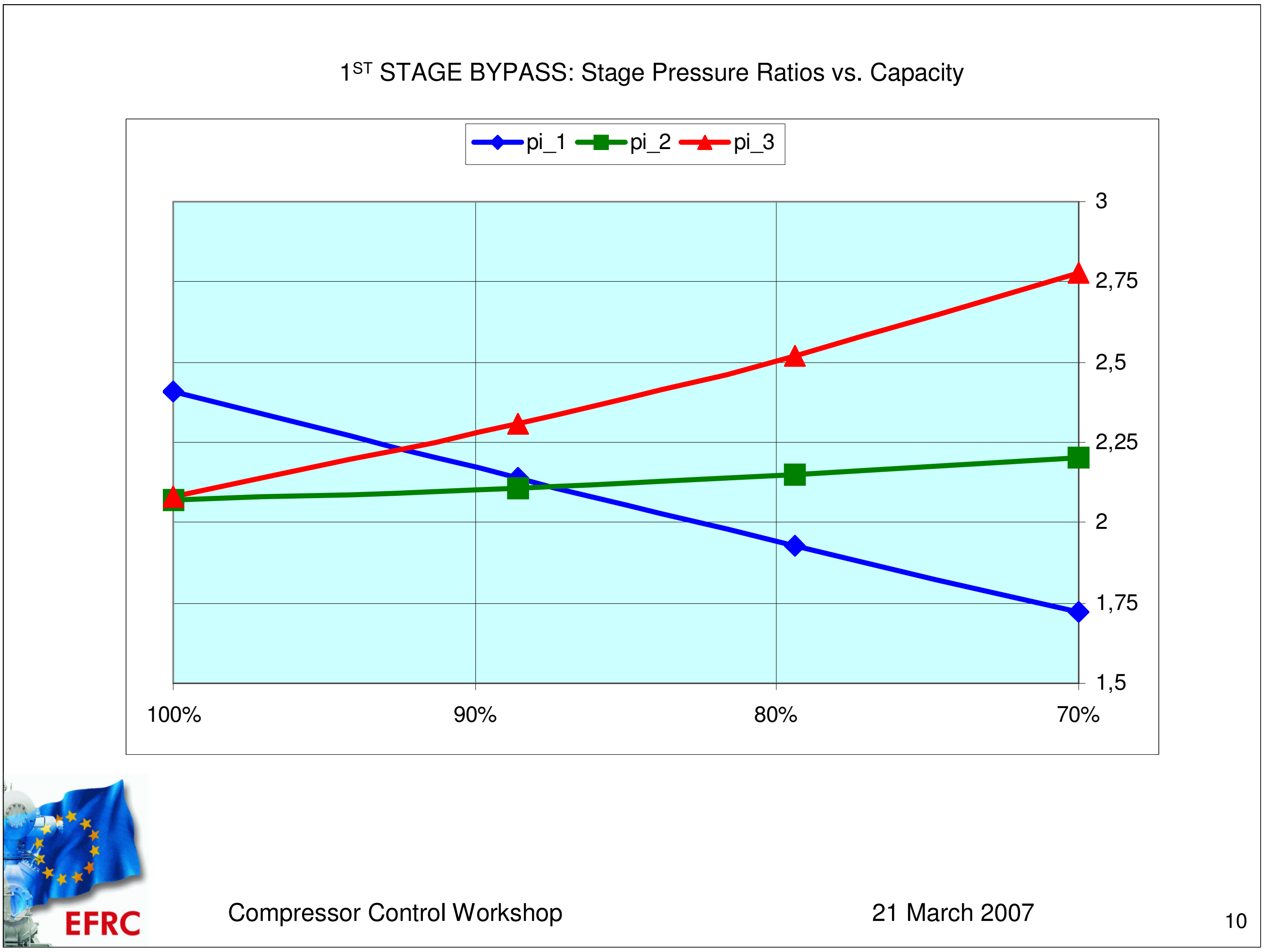
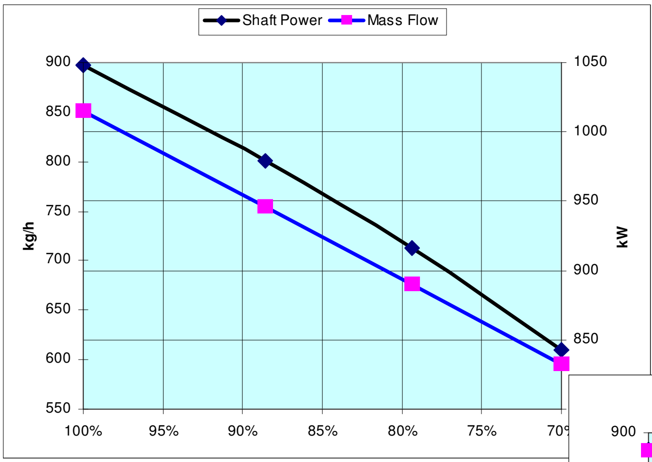
By recirculating part of the first stage capacity, the inlet pressure of the second stage and thereby the total discharge capacity is reduced.

Simultaneously the pressure ratios of the following stages increase, resulting in higher temperatures especially of the final stage.



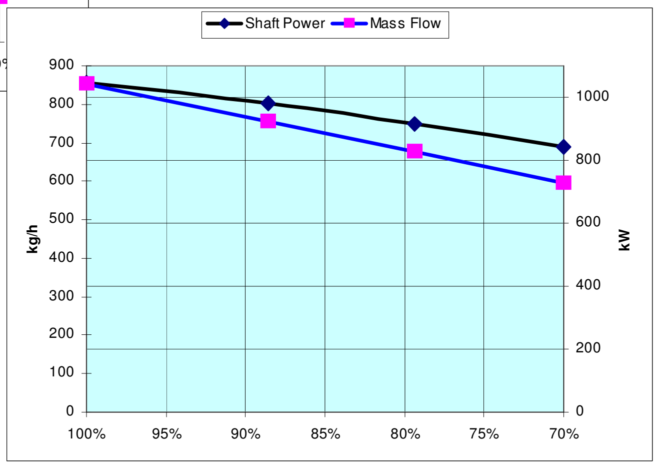
The same effects in terms of pressure ratio per stage.

The widest control range can be achieved if the design pressure ratios at full load are distributed in a decreasing pattern, from highest possible in the first stage to lowest in the last stage.



1ST STAGE BYPASS:

Shaft Power and Mass Flow



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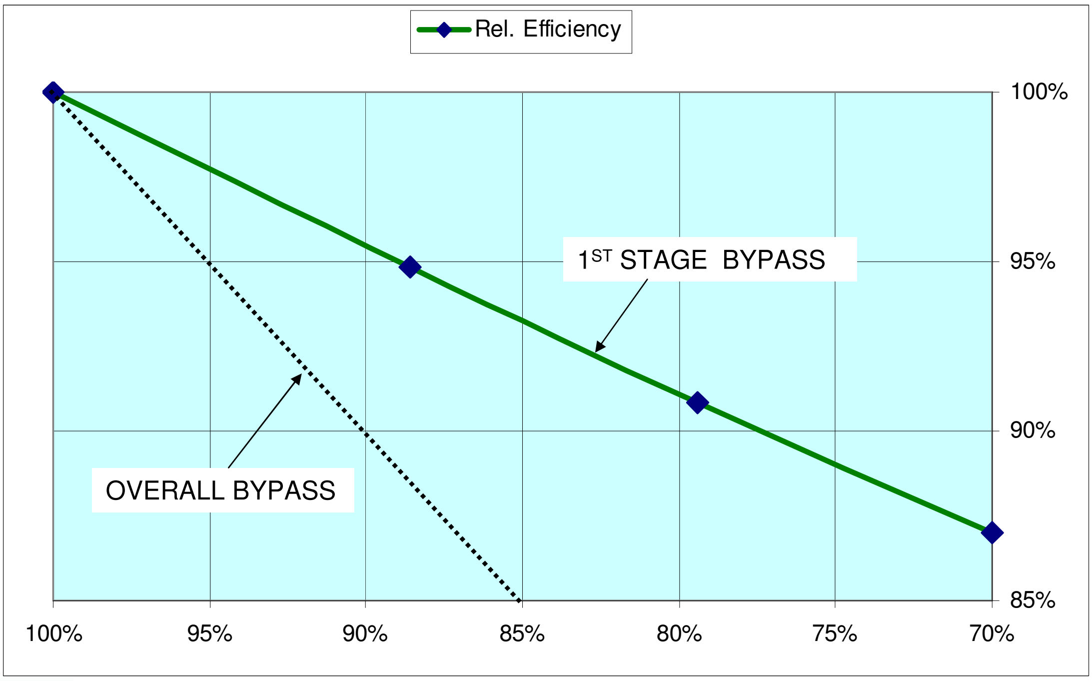
Both diagrams show the capacity and the absorbed power vs. the relative capacity in percent.

The left diagram might induce the conclusion that the absorbed power decreases quicker than the capacity – this is of course not so.

The right diagram with non-suppressed zero ordinates shows this clearly.

1ST STAGE BYPASS: Efficieny Decreases With Relative Capacity



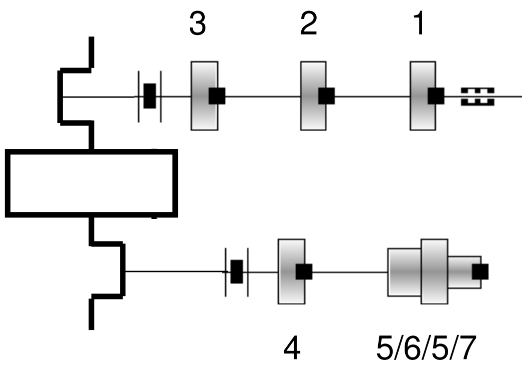
In this sample case the energetic efficiency drops down to 87 % at 70% capacity, while an overall bypass would result in a 70 % efficiency only (dotted line).

For the given conditions, this means a power saving of 17 % , i. e. more than 170 kW.

Bypass Example 2:

7-Stage CO Compressor, Variable Inlet Pressure, Constant Demand

Bypass Around 1st Stage



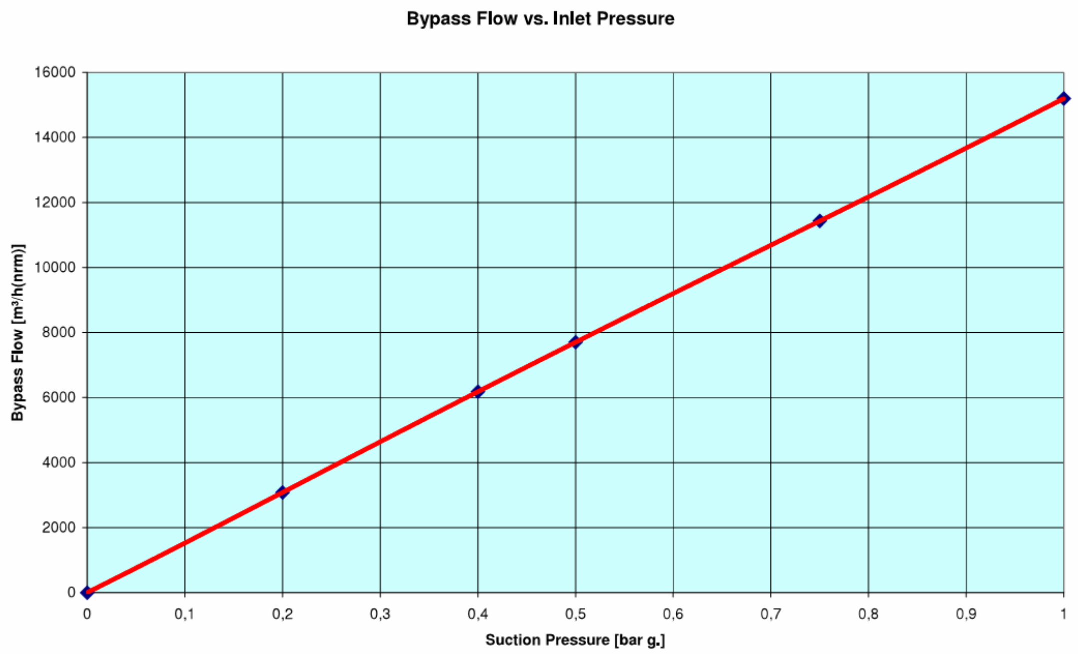
This sample shows an old-fashioned „Einheitsmaschine".

Due to changed gas supply conditions the formerly quite constant inlet pressure could now vary from 1.0 to 2.0 bar abs.

Normally, this machine is controlled by a stepless reverse flow system; this study had to assume worst conditions, i. e. no control system other than a spring loaded automatic bypass valve around the first stage.

For a constant demand, only the inlet conditions of the second stage must be kept constant.

Bypass Example 2: Valve Flow Quantity



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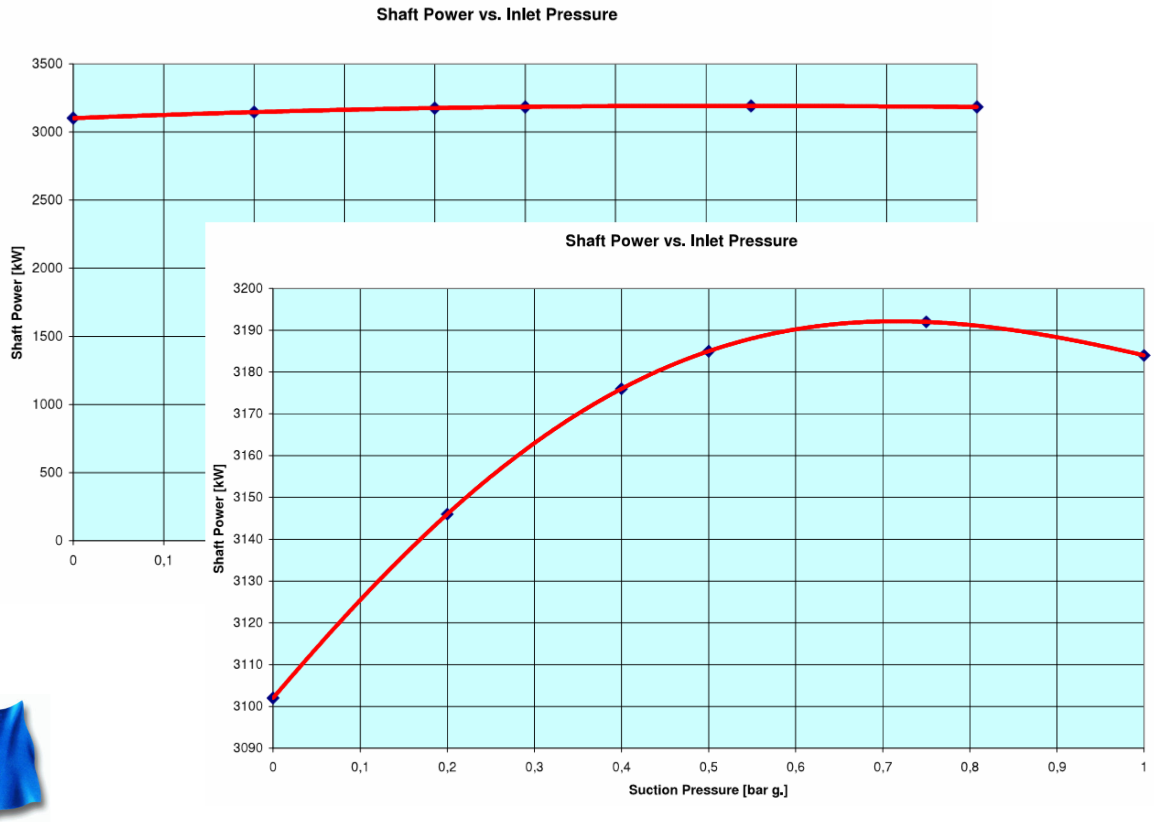
Here the bypass valve flow, nearly proportional to the inlet pressure 1st stage, is shown.

Since at twice the inlet pressure the first stage capacity is more than doubled, the bypass flow gets higher than the total capacity.

(Without a functioning control, the frame and the driver would immediately fail at approx. 10% inlet pressure rise)

Bypass Example 2: Absorbed Shaft Power



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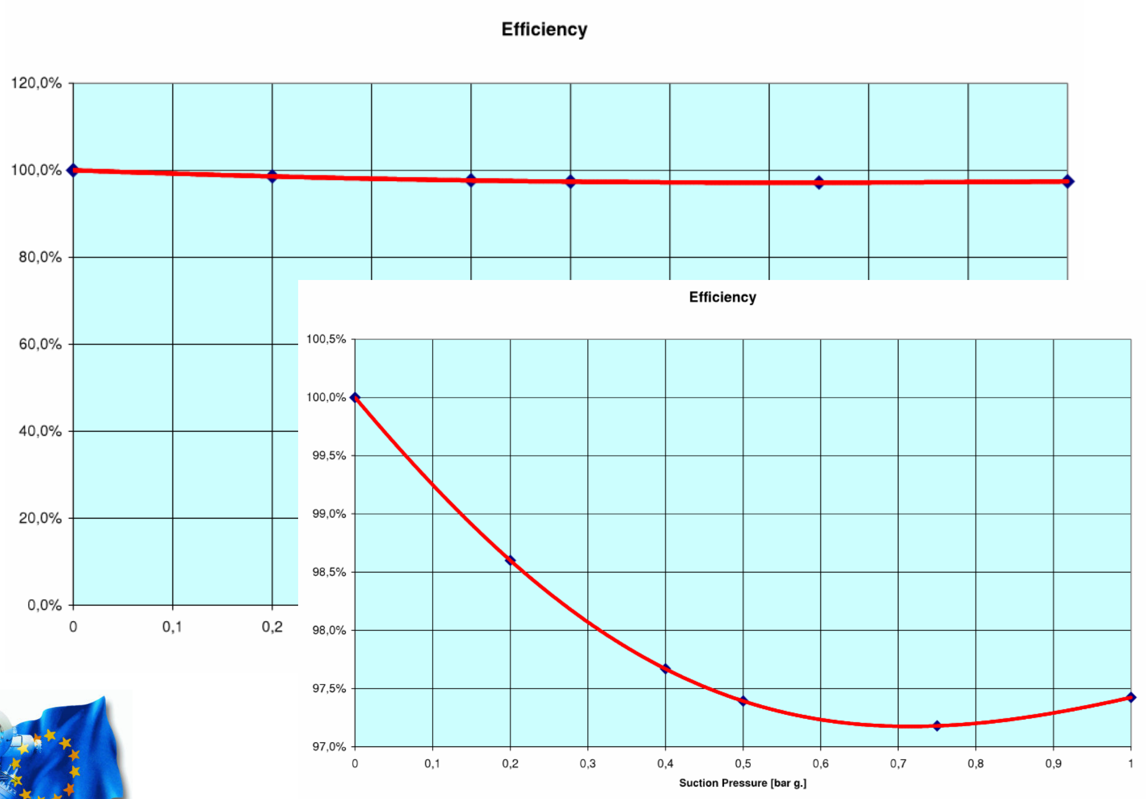
With a seven-stage machine, the effects of the increased inlet pressure are rather small, resulting in a power increase of less than three percent.

Bypass Example 2: Efficiency



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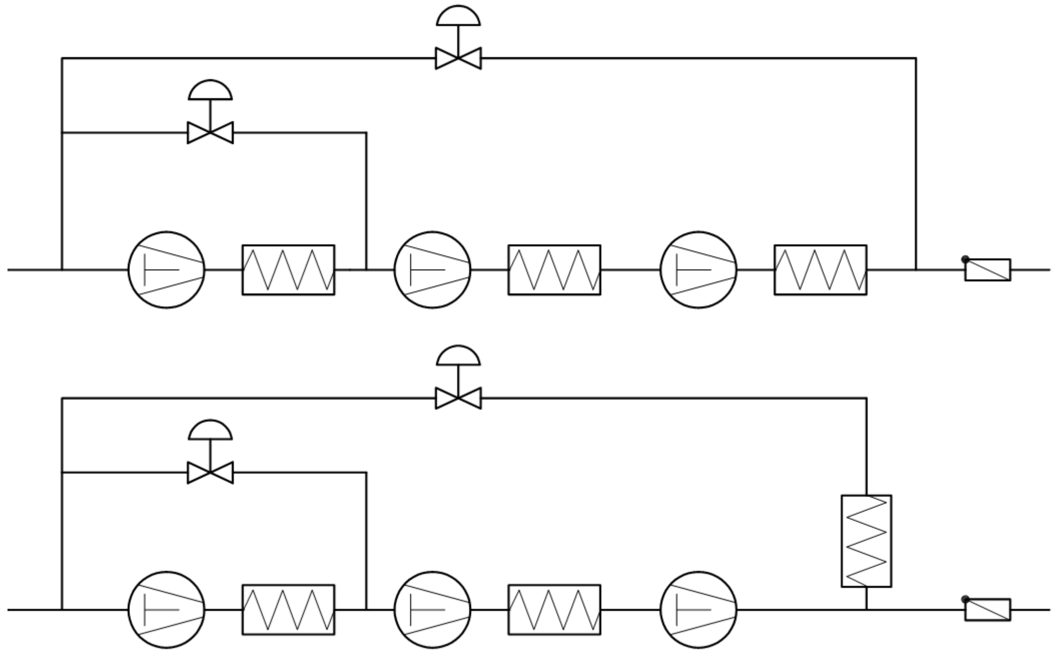
This is also reflected in these diagrams.

A few remarks here:

• The „efficiency“ is shown relative to the power consumption at the lowest specified inlet pressure, not relative to any ideal process.

• For maintaining power consumption and capacity at increased suction pressure, the method „suction throttling“, i.e. controlling the inlet pressure of the first stage by means of a throttling valve, would also be possible. The size of such a valve had to be rather big, since the design inlet pressure of this machine was atmospheric.

BYPASS COMBINATIONS



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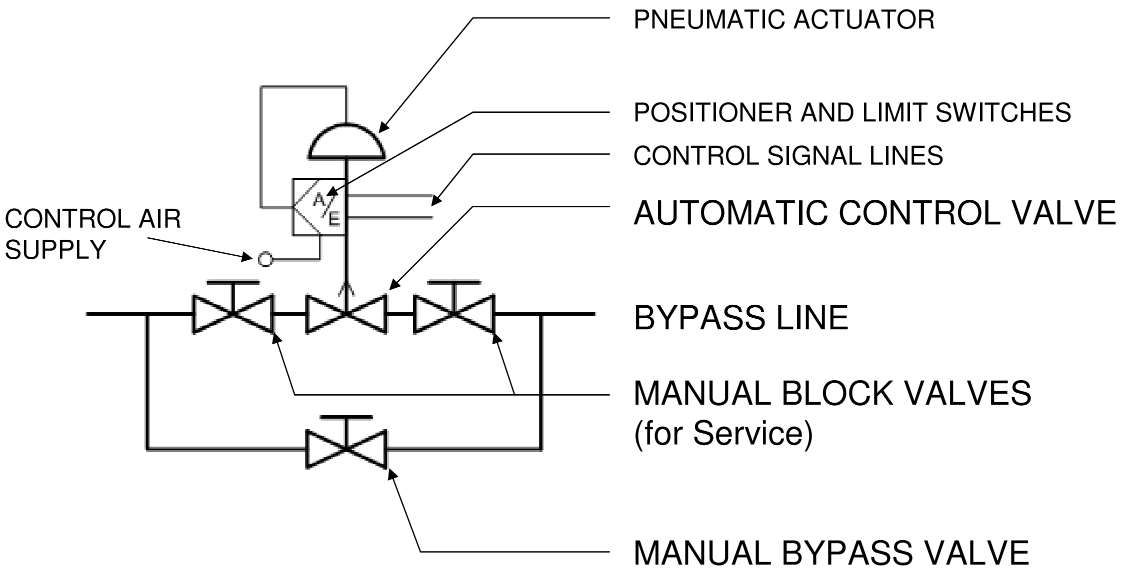
For extending the control range beyond that of a first stage bypass, an additional overall control bypass can be used.

For optimum efficiency, the first stage bypass should always have priority, the overall bypass only opened if the limiting parameters for the first stage bypass (often: high end temperature last stage) are reached.

This arrangement is often also used for unloaded start-up.

With machines having more than two stages, both the bypass valves should be fully open for start-up, to reduce the build-up of interstage pressure due to the flow resistance of the higher stages,

TYPICAL BYPASS VALVE GROUP



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While the previous sheets showed the bypass valves as simplified single items, the real valve arrangement is usually more complex,

To achieve stable control behavior, the automatic valve is equipped with a positioner and limit switches.

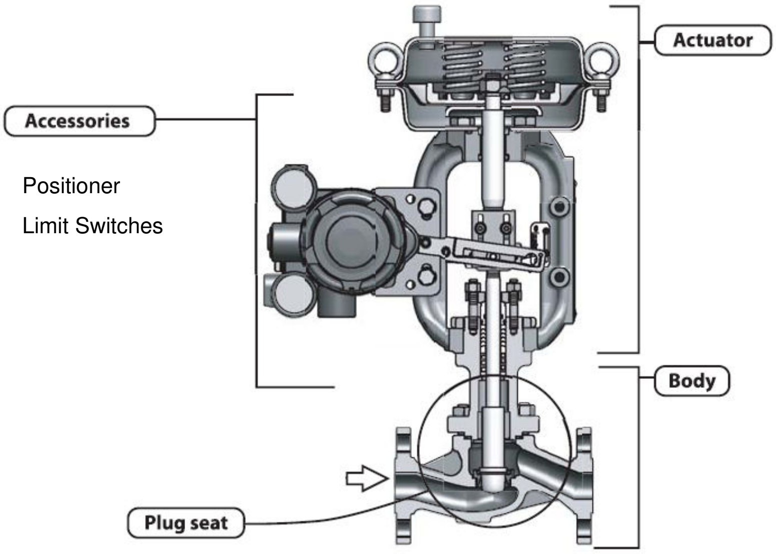
The isolating valves are used for blocking off the automatic control valve for maintenance. Their flow resistance, especially that of the downstream valve, must be carefully engineered.

To keep the compressor unit functional during this, a manual bypass valve is arranged parallel to the automatic one.

Even on this level of detail, drain valves, purge connections, heat tracing, and noise insulation are not shown but must be considered.

Since in most cases the pressure ratio across the bypass valve is supercritical, the flow speed in the valve is equal to the sound velocity, causing noise levels much higher than the compressor if no precautions are taken, e.g. special cage designs instead of a simple plug valve, and external insulation of valve and downstream piping.

TYPICAL CONTROL VALVE ASSEMBLY



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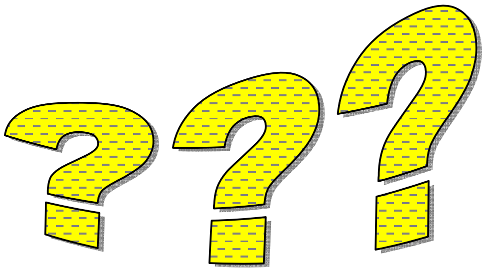
This sketch shows the major components of a pneumatic control valve in „normally closed“ execution.

Details regarding the flow characteristic, the packing, the seat and plug, and many more can be found e.g. at

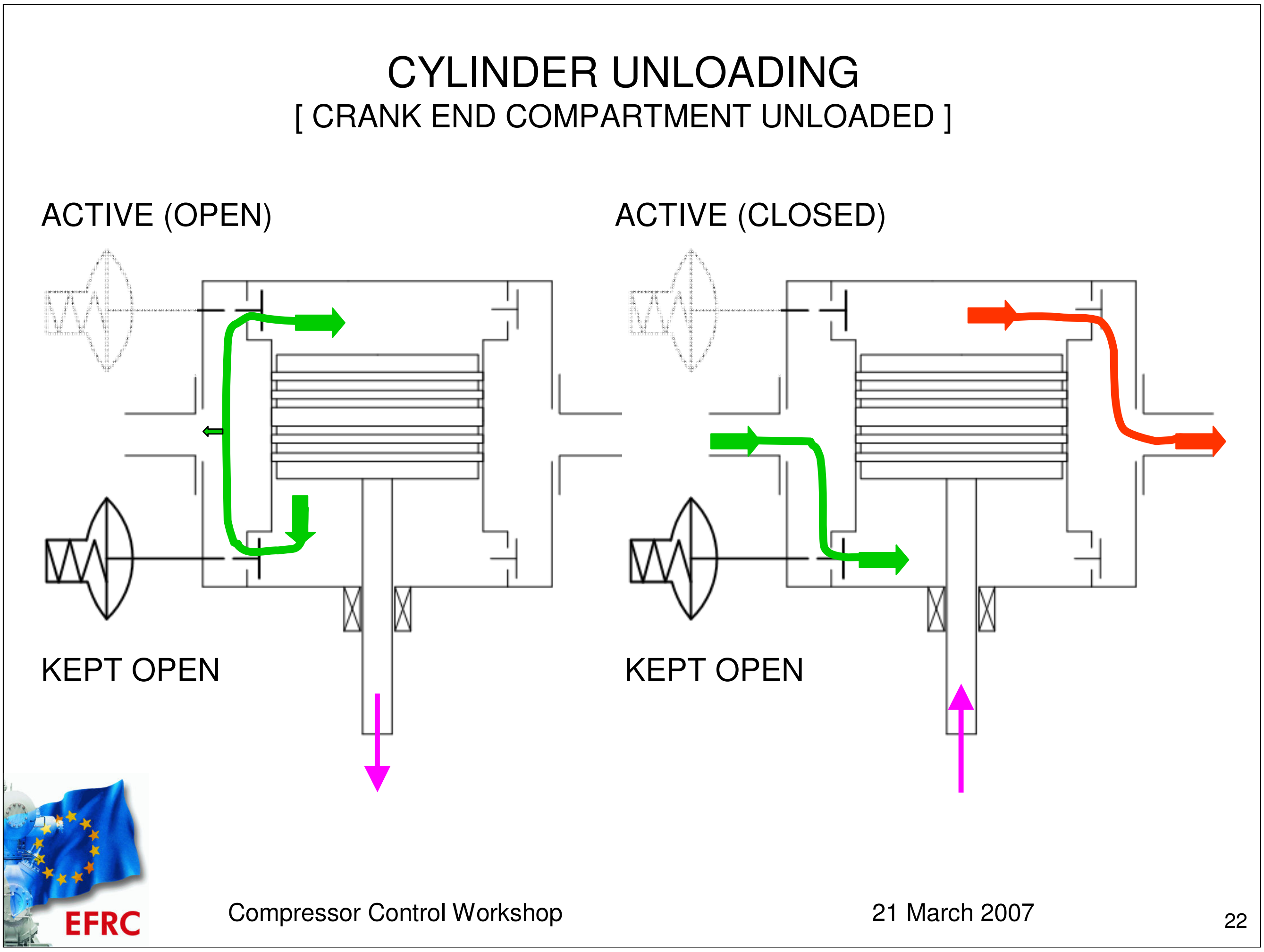
http://www.emersonprocess.com/fisher/

BYPASS CONTROL



CYLINDER UNLOADING

- FUNCTION PRINCIPLE
- HARDWARE
- EXAMPLES



This sketch shows the operation of a double-acting cylinder with unloaded crank-end compartment:

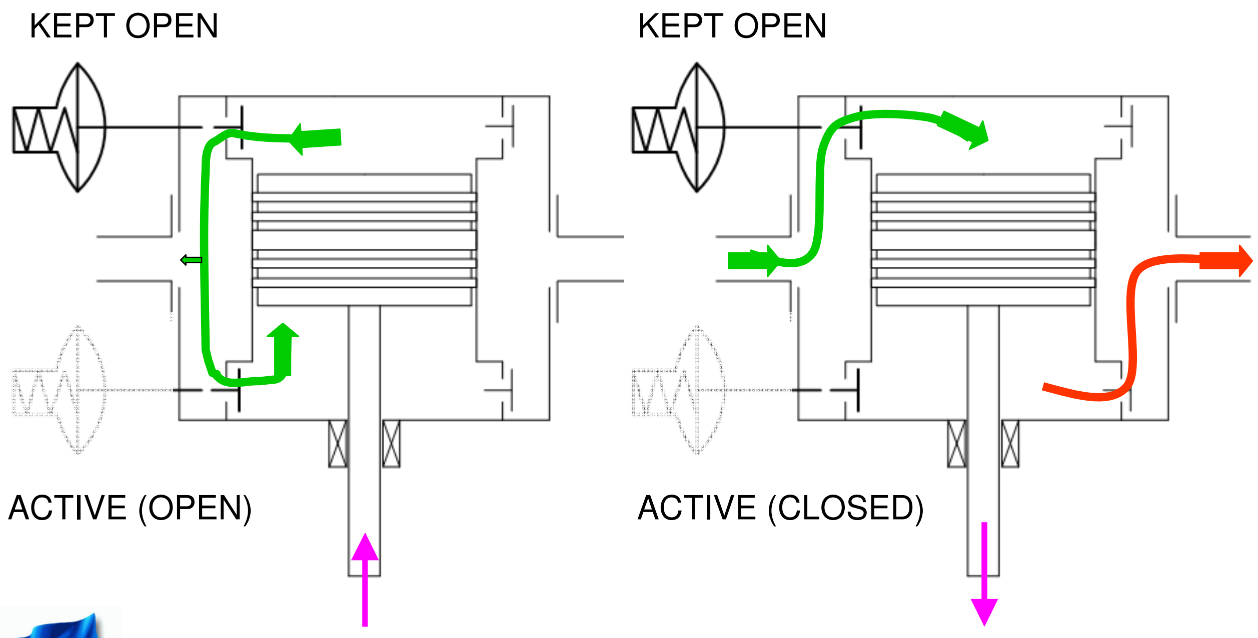
During the downward motion of the piston, the gas is moved from the crank-end compartment through the kept-open suction valves to the head end compartment.

The upward movement then compresses the gas in the head-end compartment until it is pushed out through the HE discharge valve. Simultaneously, the CE compartment fills with fresh gas from the inlet line.

At the beginning of the HE compression, the gas has passed not only the HE suction valve, but already two times – once in, once out – the kept open CE suction valve.

CYLINDER UNLOADING

[HEAD END COMPARTMENT UNLOADED]



The process is identical when the head end suction valves (or ports) are kept open,

In both cases, there is a small backflow into the suction line during the inlet stroke of the active compartment,

CYLINDER UNLOADING

- FINGER TYPE UNLOADER
- PLUG TYPE UNLOADER
- PORT UNLOADER

CYLINDER UNLOADING

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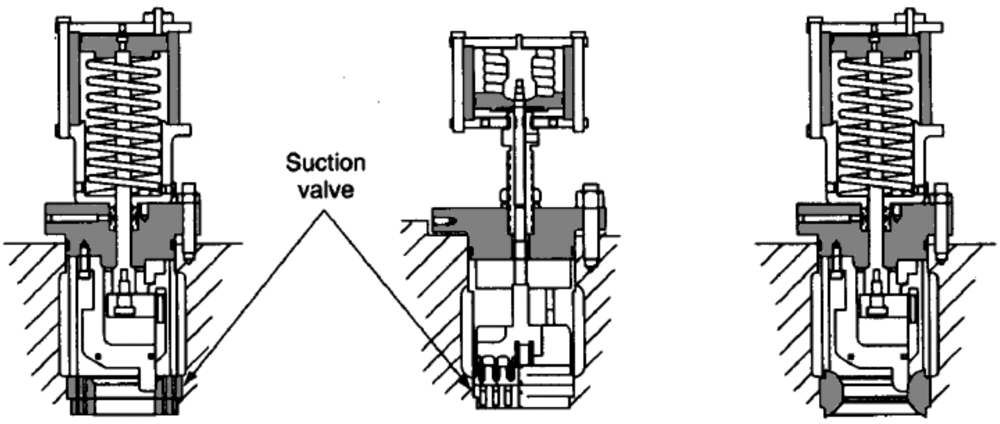
This picture shows part of a single stage NEA compressor with four vertical cylinders in line, each cylinder equipped with two suction valve unloaders.

Discounting permanent no-load operation, the nominal control steps are 100% - 87.5% - 75% - 62.5% - 50% .

CYLINDER UNLOADING

BASIC SKETCHES [from API 618]



Plug Type Unloader | Finger Type Unloader | Port or Clearance Pocket Unloader

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These API schematics show the three main types of unloaders,

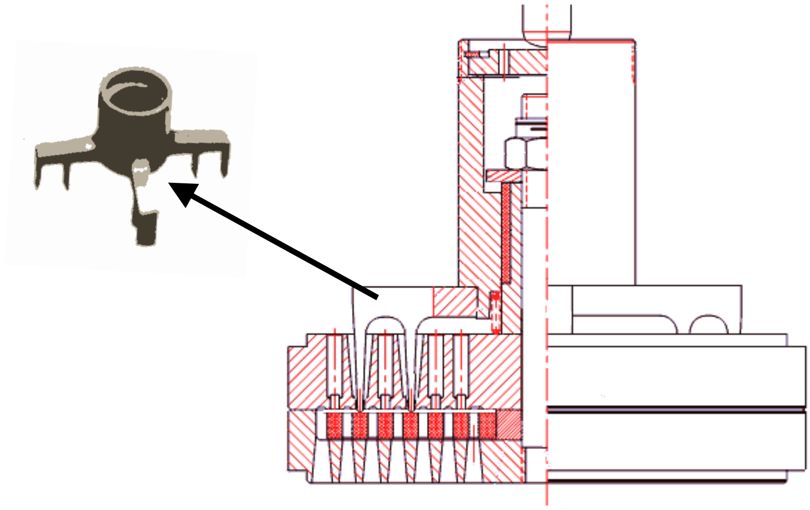
with the preferred spring orientations toward unloading, i. e. the control pressure for the pneumatic actuators <u>loads</u> the respective compartment.

While the finger type unloader is suitable for suction valves only, the port and plug unloaders can – in special cases – also be installed on the discharge side.

The port unloader is mainly used if the cylinders have an otherwise unused valve port, as is often the case when a standard cylinder with more than one valve per side is used for a light gas like hydrogen. In this case the necessary valve area can be provided with one valve only, the other port free for a port unloader.

Suction Valve Unloading:

Plate Valve with Finger Unloader



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This sectional sketch shows a typical combination of a plate valve with a finger unloader as an assembly unit.

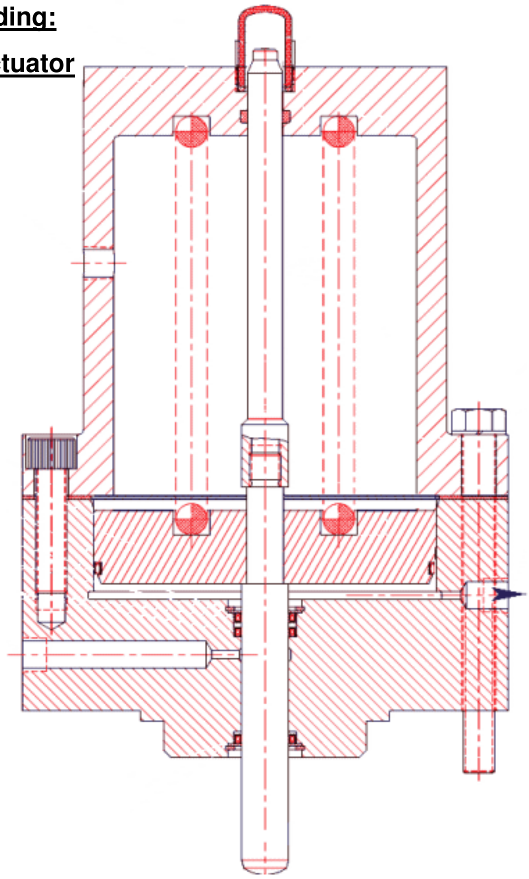
By exerting a suitable force on the top of the unloader assembly, this is pushed down against the forces of the central return spring and of the closing (and damper) springs in the valve guard, such forcing the valve plate against the guard by the fingers that reach through the openings in the valve seat.

In unloaded position the actuator force must always be sufficient to overcome all the closing forces, especially the dynamic gas force on the valve plate.

Usually, the axial gas passages in the valve seat are substantially larger than the lift area of the valve. Therefore, the increase in flow resistance by the delicate fingers slighty obstructing the grid area is very low.

Suction Valve Unloading:

Pneumatic Piston Actuator



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This pneumatic piston actuator was also shown on the photo.

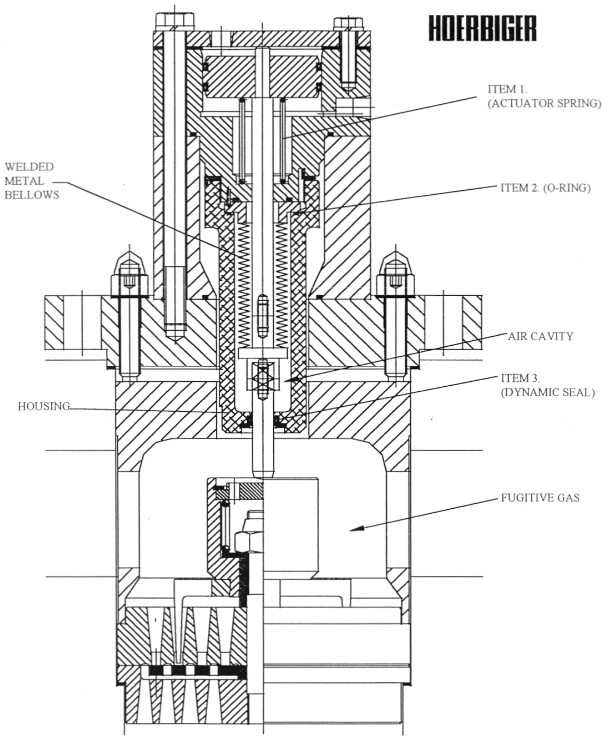
It fulfills the API recommendations in most respects:

The unloading force is provided by a spring, bringing the compressor in unloaded condition in case of control pressure failure, and it advertises its state through the transparent cap on top of the housing.

The tightness of the seals can be verified by connecting the vent between them to a flowmeter or a bubble glas, such avoiding substantial losses of control air or process gas by leaking stem seals.

Suction Valve Unloading:

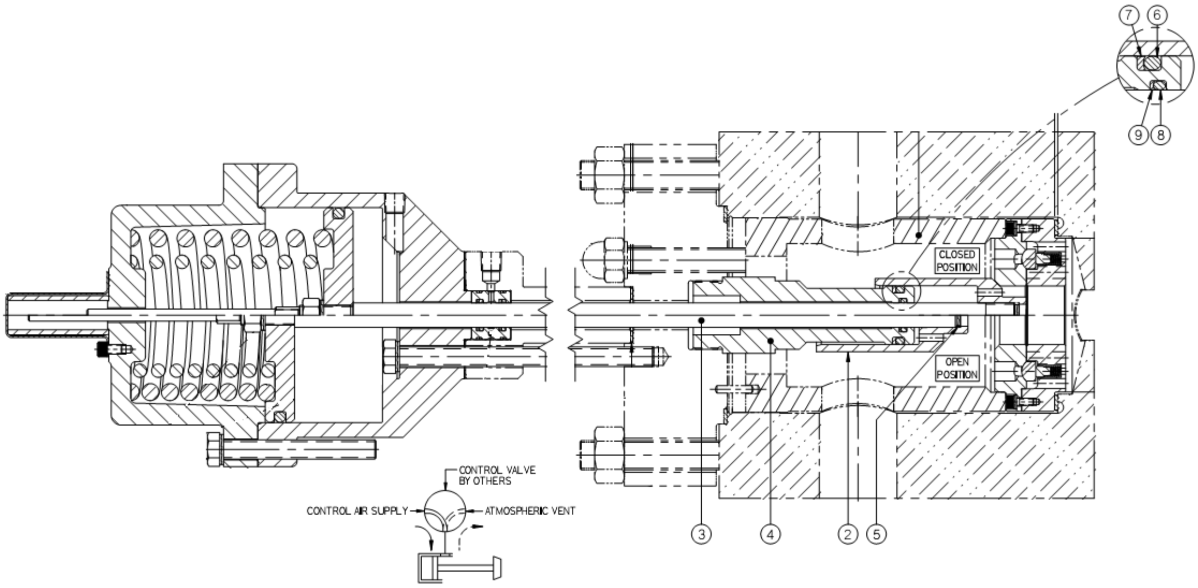
Plate Valve with Finger Unloader, Pneumatic Piston Actuator, Bellows Seal



This shows a similar design, the main difference being the bellows seal for the stem, with an additional protective shroud, resulting in zero leakage.

Suction Valve Unloading:

Ring Type Valve with Plug Unloader, Pneumatic Piston Actuator **[Cook Manley]**

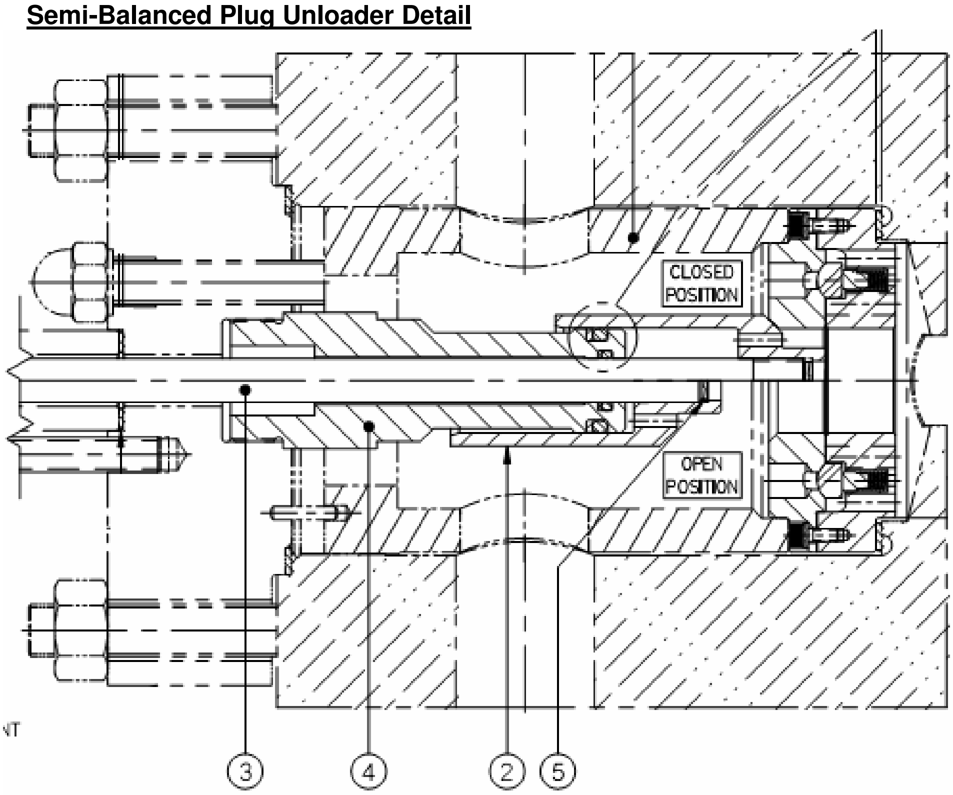


This assembly consists of a ring-type suction valve with central plug seat, a plug assembly, and a pneumatic piston actuator.

In this case the spring keeps the port closed, and pressure is needed for unloading.

Suction Valve Unloading:

Semi-Balanced Plug Unloader Detail



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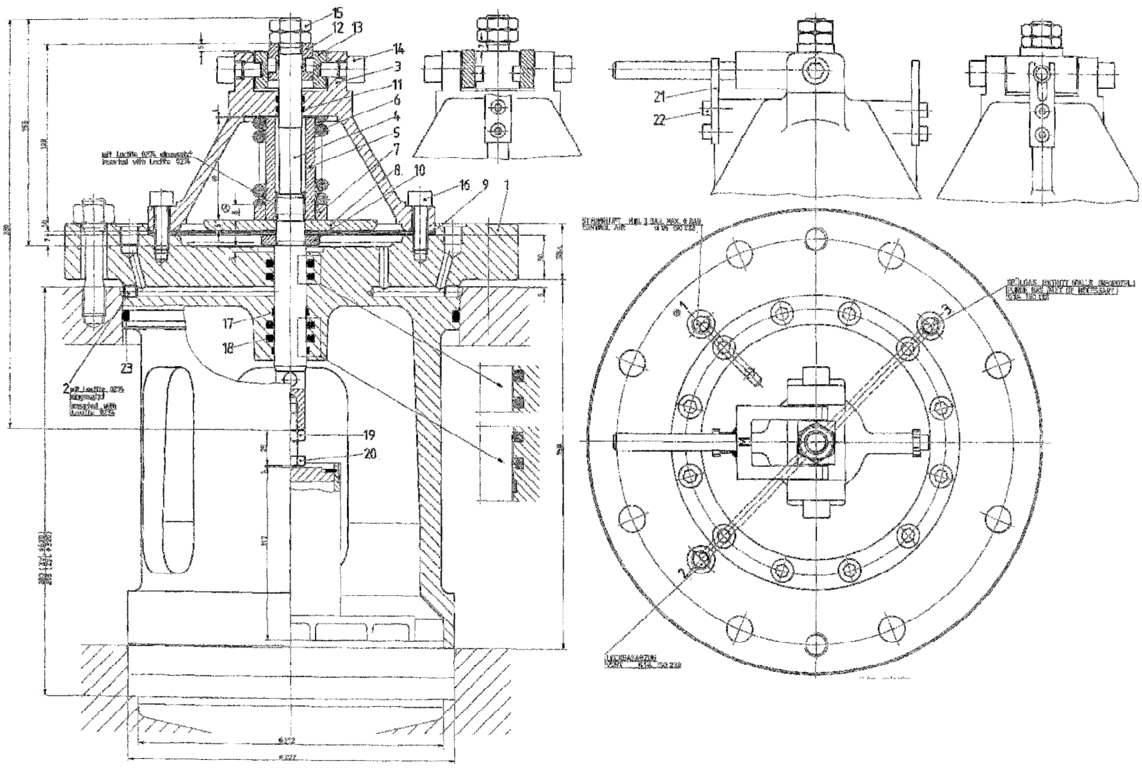
The force necessary for keeping the port closed is low due to the semi-balanced design of the plug assembly.

Suction Valve Unloading: Diaphragm Type Actuator Integral With Valve Cover



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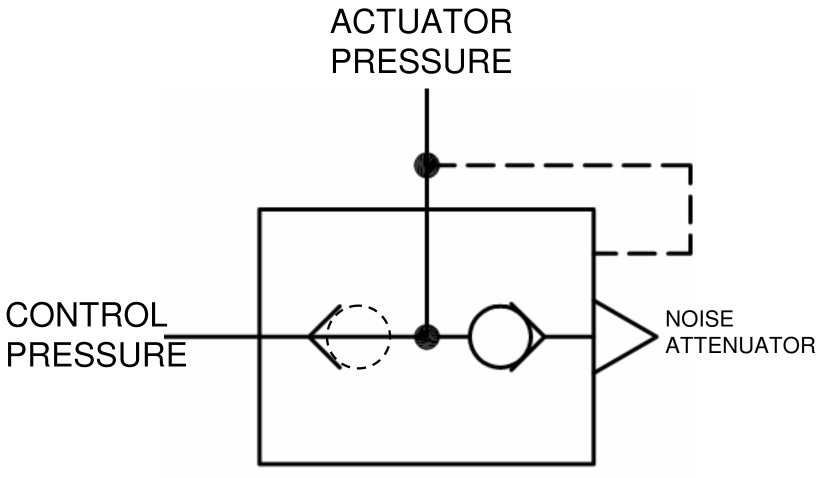
This (rather old) design shows a diaphragm actuator for a standard finger type unloader:

- **The housing is part of the valve cover**
- **The system unloads by spring force at control pressure failure**
- **The control state is indicated by the stem end**
- **A manual override is provided, using a lever and an eccenter mechanism.**
- **The stem is guided by PTFE sleeves, and sealed by stepseals.**
- **The seal condition can be monitored by the vent flow.**

(I designed this in the 1970s)

Valve Actuator Control Detail:

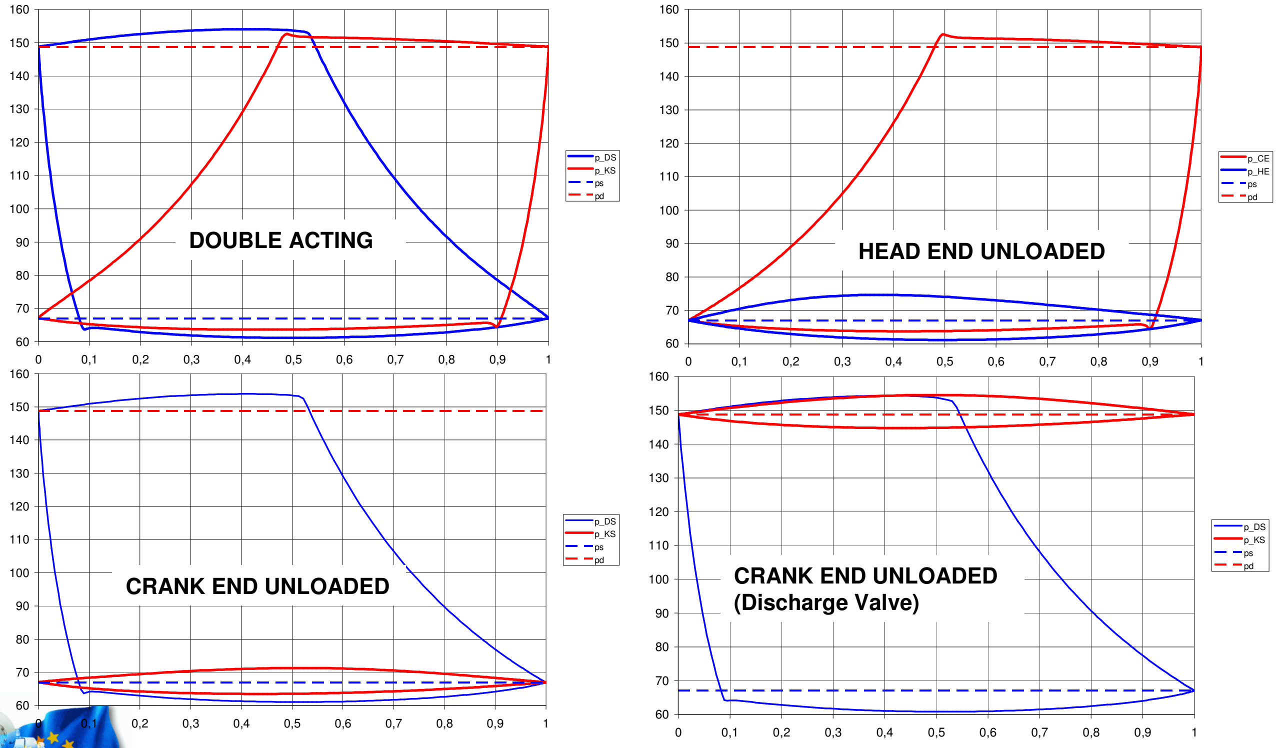
Quick Vent Valve



To achieve an immediate response, so to avoid intermediate actuator positions over several strokes, a quick vent valve is recommended.

This is especially important if the solenoid valves controlling the actuator pressure are arranged far from the valves.

Suction Valve Unloading: Typical Thermodynamic Effects
Indicated Pressures



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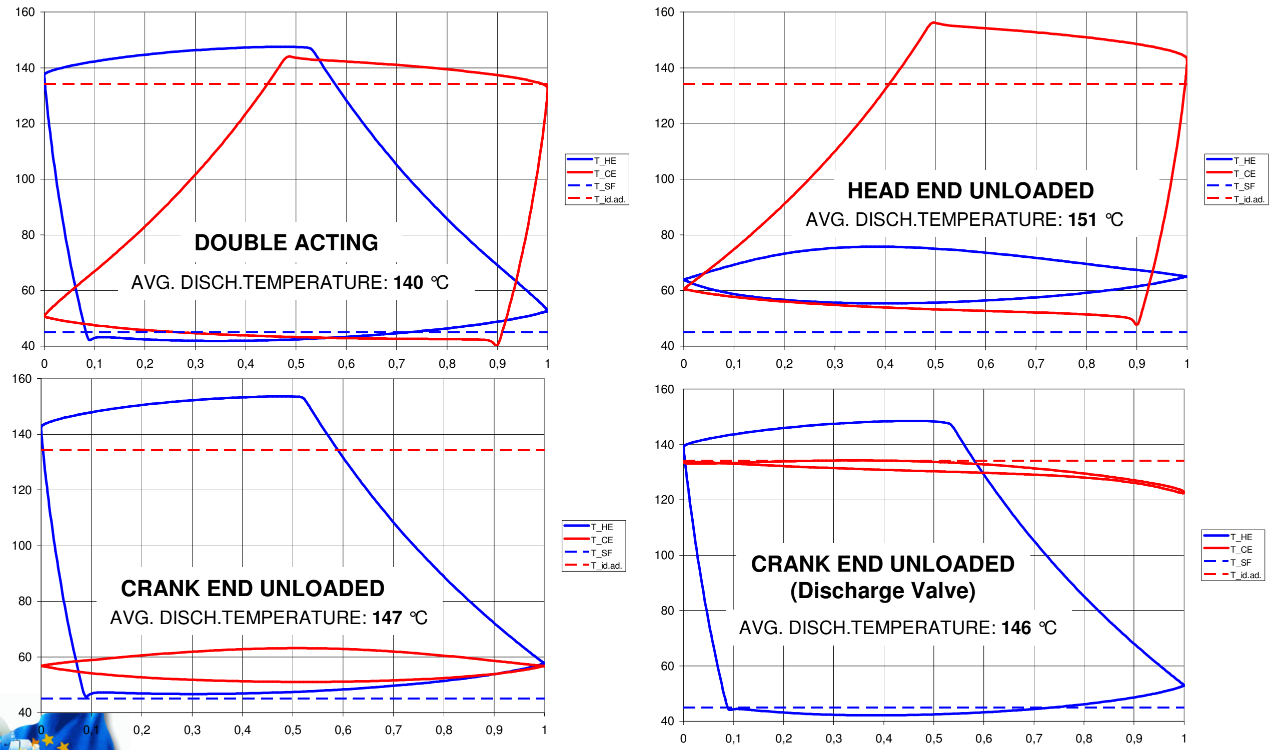
This and the following sheets show results from a proprietary simulation program.

The PV graphs show the typical fishbelly shapes.

Suspension

<u>Suction Valve Unloading: Typical Thermodynamic Effects</u>
<u>Indicated Temperatures</u>



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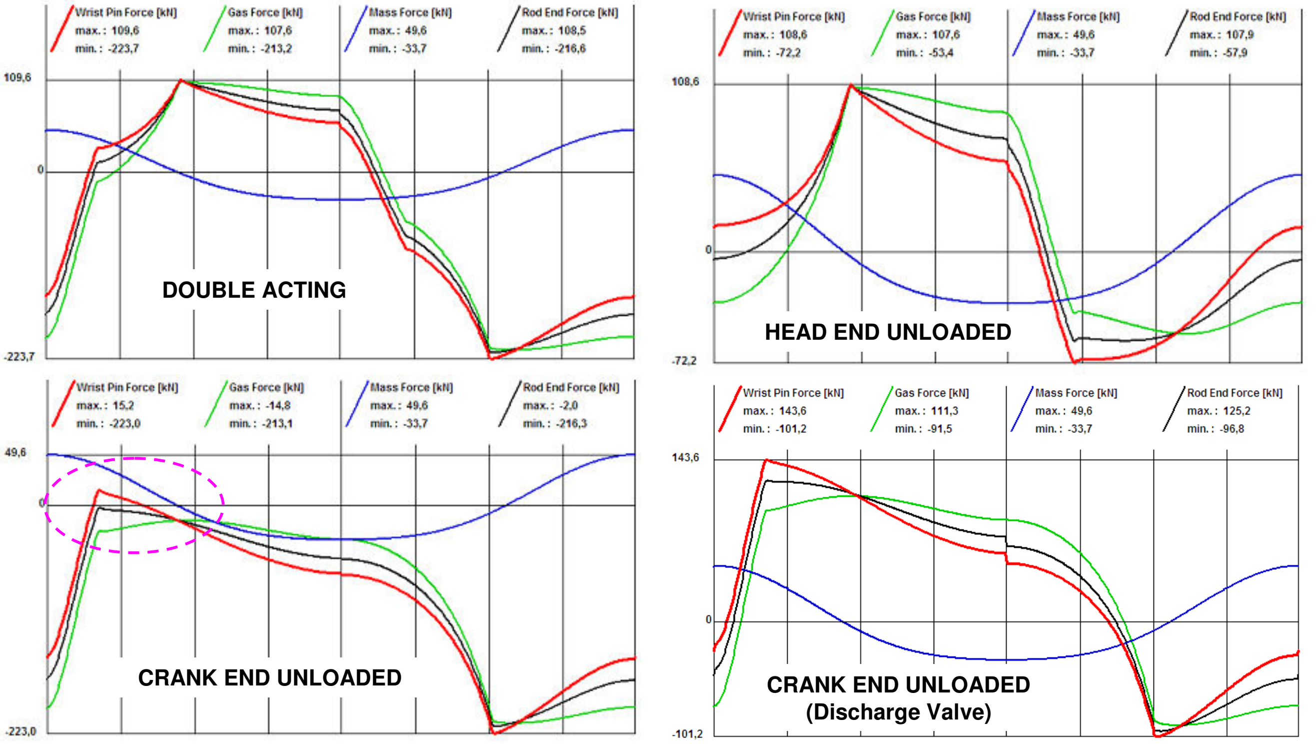
Due to the multiple passes of the gas through the unloaded suction valves, the discharge temperatures in part-load condition are increased correspondingly.

Altogether not surprisingly, the cylinder with the unloaded <u>discharge</u> valve shows the smallest temperature rise, due to the high temperature difference between the gas and and the cooled cylinder walls in the unloaded compartment.

Suction Valve Unloading: Typical Mechanical Effects

Wrist Pin Forces



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In most cases, switching from double to single acting will not cause problems with the rod load reversal.

Also in this case, the API limits (3% of the opposite load over 15° crank angle) are not exceeded.

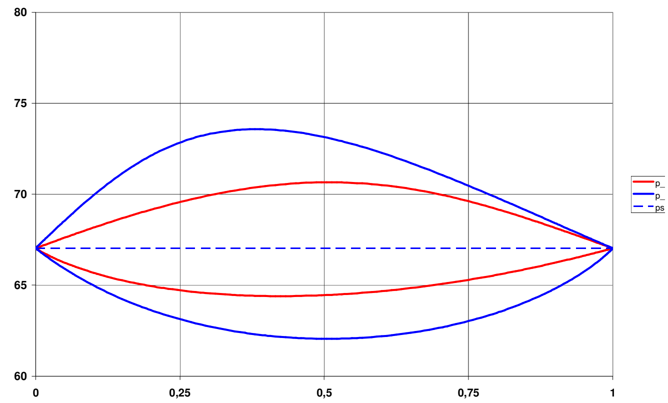
Nevertheless, the load case 3 (lower left) may easily deteriorate by a small change of the pressure level, e.g. due to wear of the piston rings.

In such cases, the unloading of the <u>discharge</u> side by means of a plug unloader should be considered.

Suction Valve Unloading: Typical Thermodynamic Effects

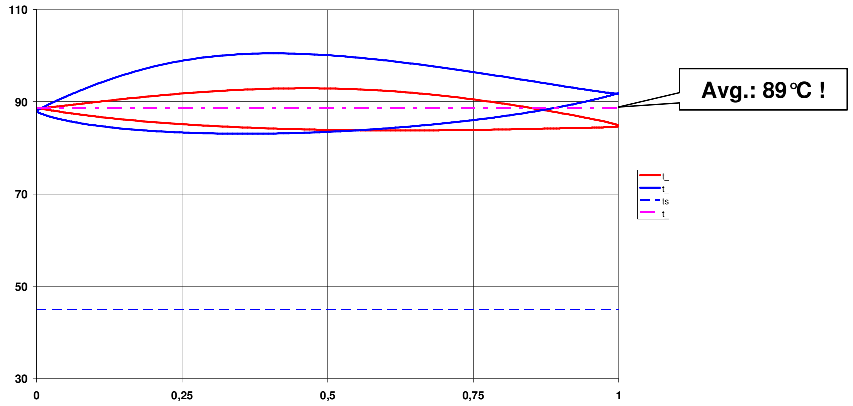
Indicated Pressures

[IDLE]



Indicated Temperatures

[IDLE]



If a cylinder is switched to no load, i.e. head and crank end unloaded over a longer period, the heat generated by piston and guide ring friction and by the continuous valve losses can be transferred only through the cylinder walls into the cooling water jacket.

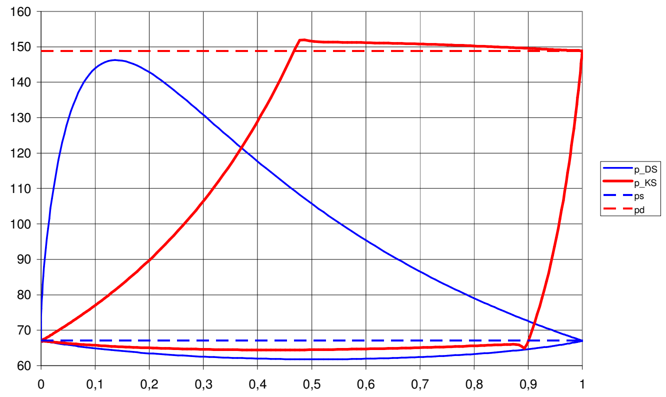
This example was calculated using Wosny's heat transfer functions. (The sample case is the same hydrogen machine as before.)

While in this case the equilibrium temperature is still moderate, in other cases with higher relative valve losses the cylinder may over-heat. This is the more probable the larger the cylinder is, (the heat exchange surface increasing with the square of the size, the contained gas with the 3rd power).

Suction Valve Unloading: Typical Thermodynamic Effects

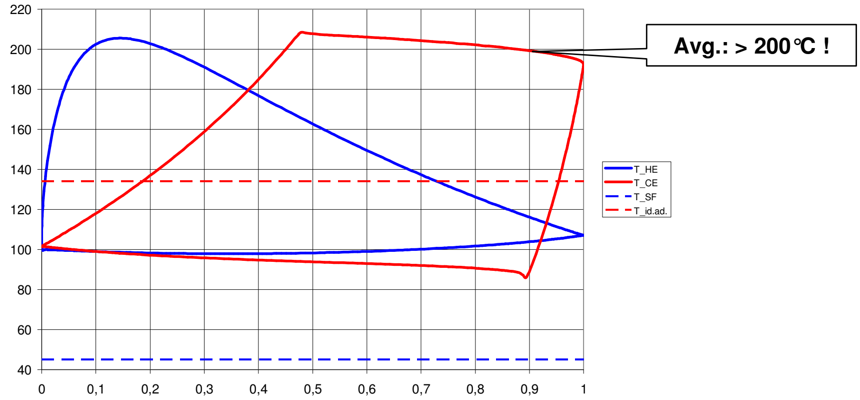
Indicated Pressures

[HE valve only partially kept open]



Indicated Temperatures

[idem]



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If it happens that the unloaded port or valve does not fully open, the flow resistance increases reciprocal to the sqare of the reduced area.

This can lead to a drastically increased temperature level, causing damages as shown on the next sheet.

Suction Valve Unloading: Typical Thermodynamic Effects

Observed Result [Valve with Nylon Rings}

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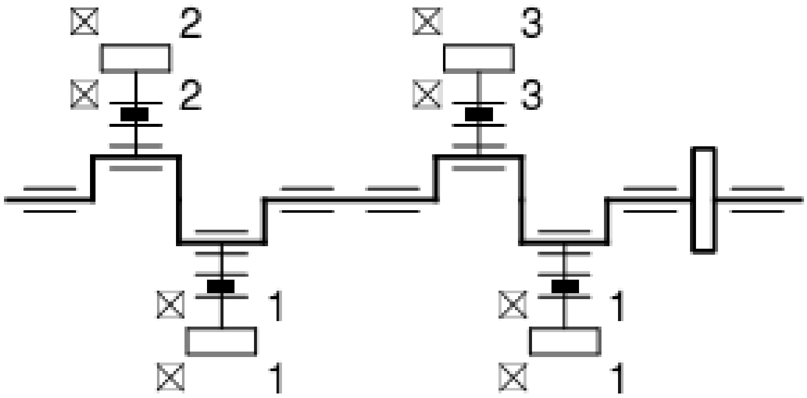
Ring type valve damaged due to plug unloader malfunction

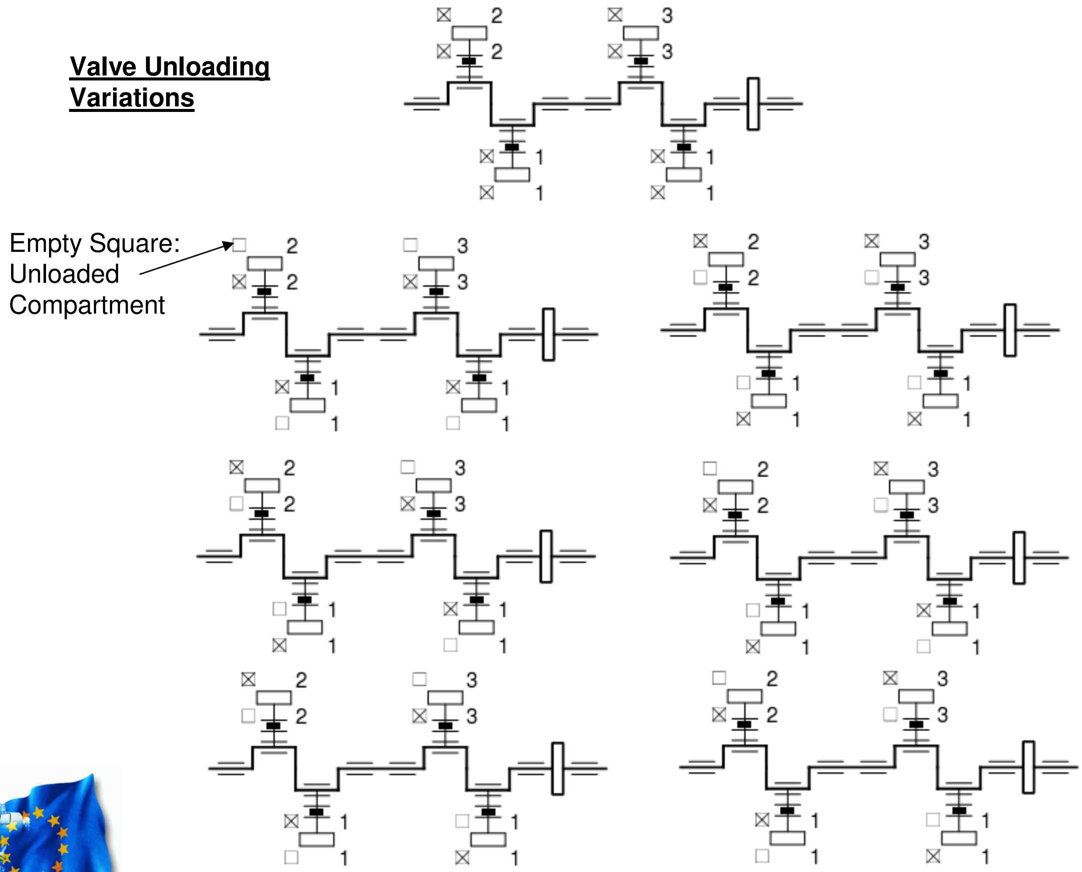
Valve Unloading: Sample Cylinder Arrangement

3 STAGE, 4 CRANK BALANCED OPPOSED COMPRESSOR



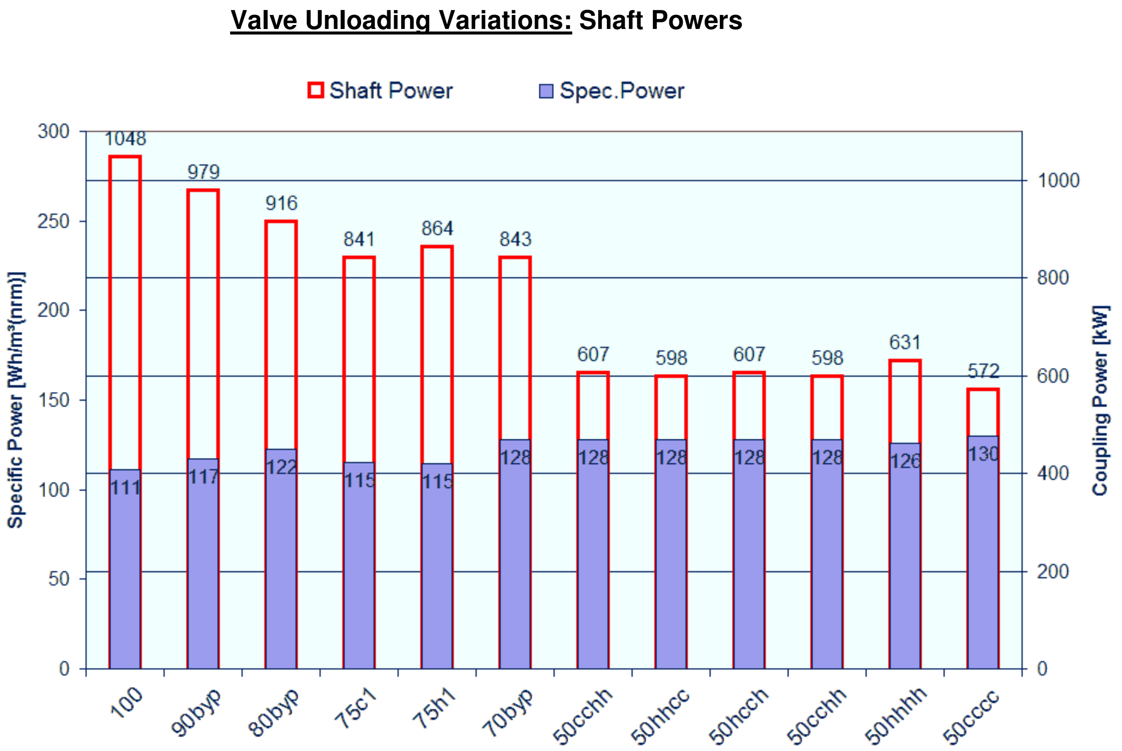
This sample shows the ideal case of a three-stage compressor with double acting cylinders only, each equipped with two suction valve unloaders.

Valve Unloading Variations



To achieve a part load near 50 % capacity, a total of sixteen combinations of cylinder unloading is possible, with different effects on

- **capacity**
- **crankshaft torque and torsional excitation**
- **crankshaft bending and journal bearing loads.**



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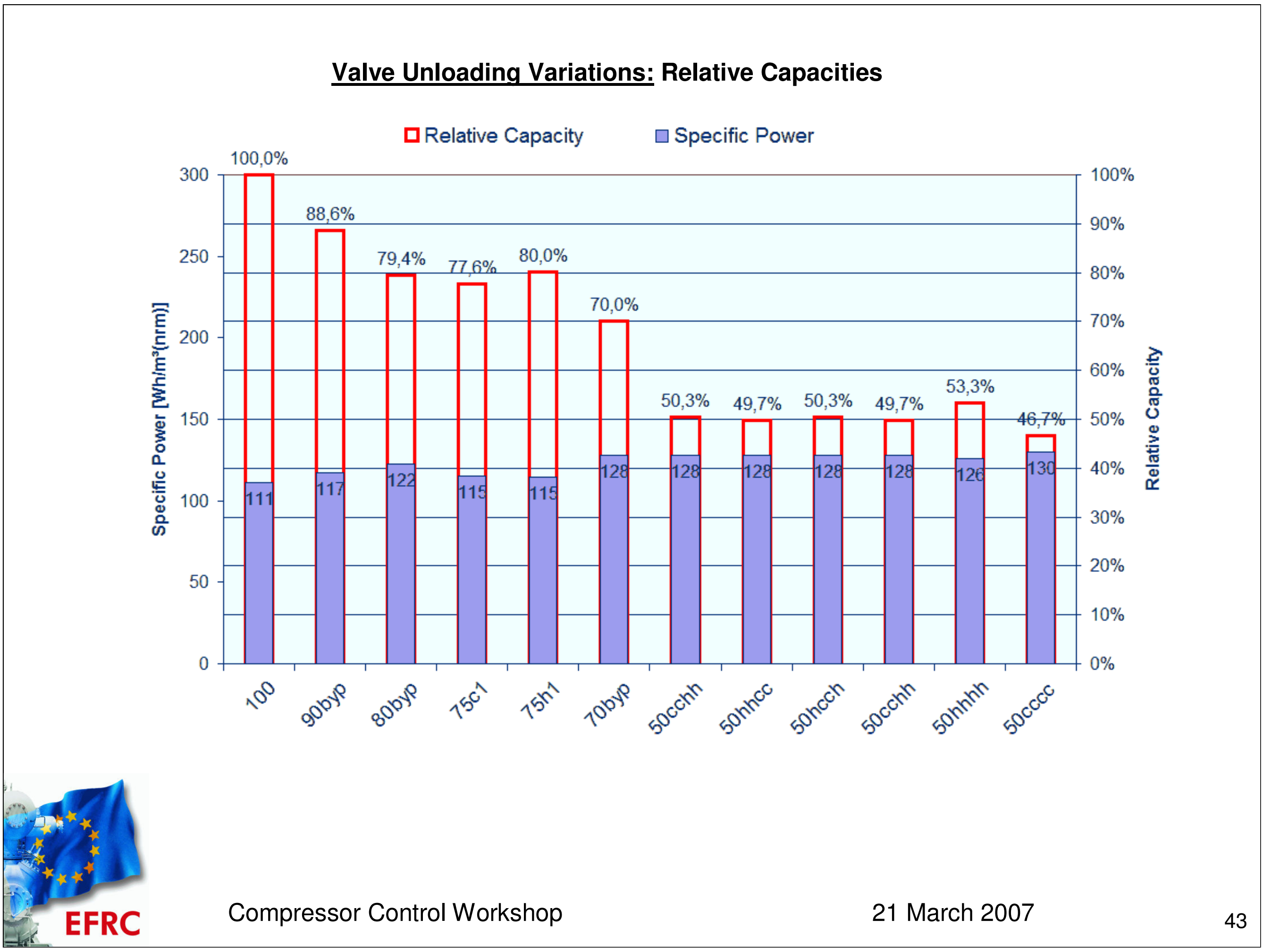
The diagram shows the absorbed and the specific powers for

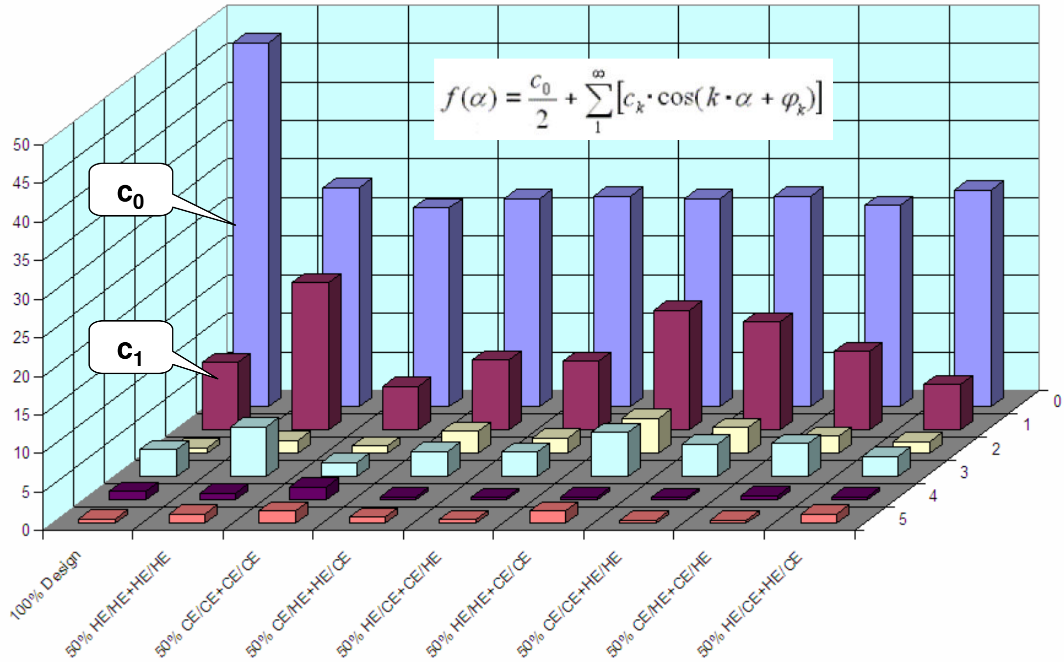
- full load,
- some first stage bypass control points,
- two 75 % load cases by unloading only one of the four first stage compartments, and
- six variations of 50 % unloaded operation.



This is a similar diagram, showing the relative capacities.

The last two bars show the extremes (all crank ends or all head ends unloaded)

Valve Unloading Variations: Shaft Torque Fourier Spectra



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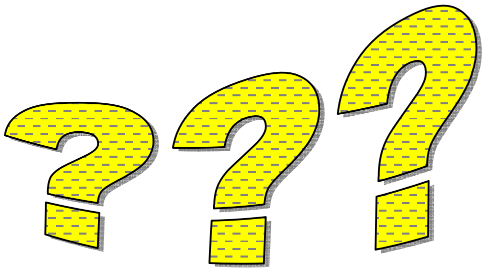
Beside full load, eight variations of 50% load were investigated,

The consequences of the selected control modes must be carefully evaluated, mainly in terms of

- **Shaft speed irregularity and effects on the electrical power supply**
- **Crankshaft fatigue safety (bending and torsion)**
- **Torsional and bending vibrations of the crankshaft**
- **Maximum crankshaft bending and resulting misalignment of the journals/bearing shells**

CYLINDER UNLOADING



CLEARANCE VOLUME CONTROLS

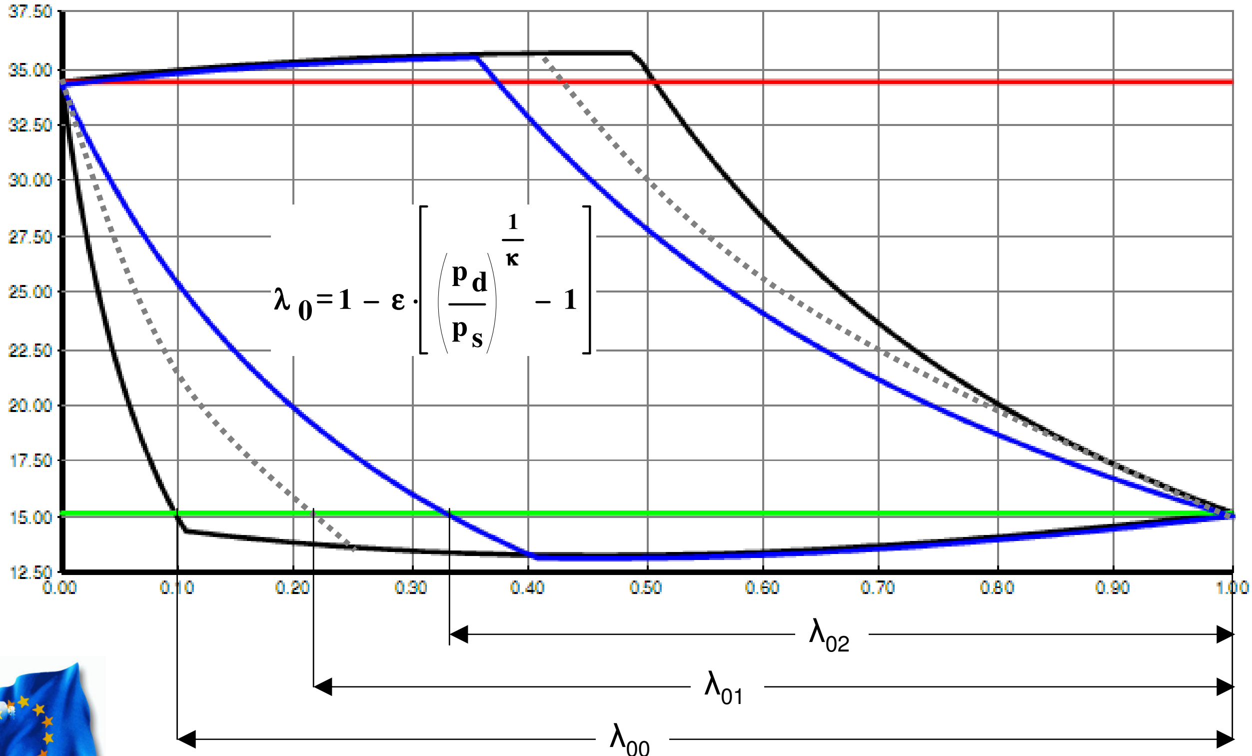
Three Major Applications:

- Adjust the capacity of a single compartment
- Vary the total capacity of a multi-stage compressor
  (constant inlet pressure, control 1st stage only)
- Keep the total capacity of a multi-stage compressor constant
  (variable inlet pressures, control 1st stage only)

The first item is generally to be performed with multistage compressors having both double and single acting cylinders, if the double acting stages are controlled by cylinder unloading.

Since this is not possible with a single acting cylinder, fixed clearance volumes can be provided instead.

CLEARANCE VOLUME CONTROLS: Thermodynamics

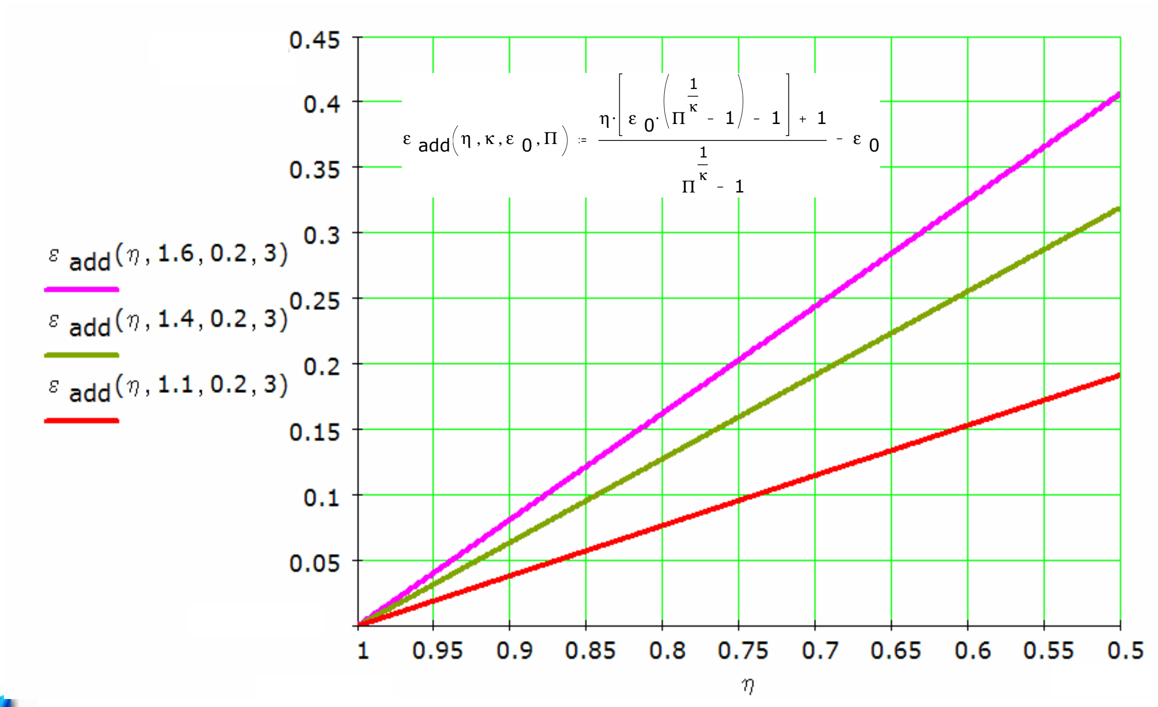


$$\lambda_0 = 1 - \varepsilon \cdot \left[\left(\frac{p_d}{p_s} \right)^{\frac{1}{\kappa}} - 1 \right]$$

This PV chart shows the effects of enlarged clearances for capacity reduction (at constant suction and discharge pressure).

CLEARANCE VOLUME CONTROLS: Thermodynamics

Necessary Additional Clearances for Capacity Reduction (Per Stage)



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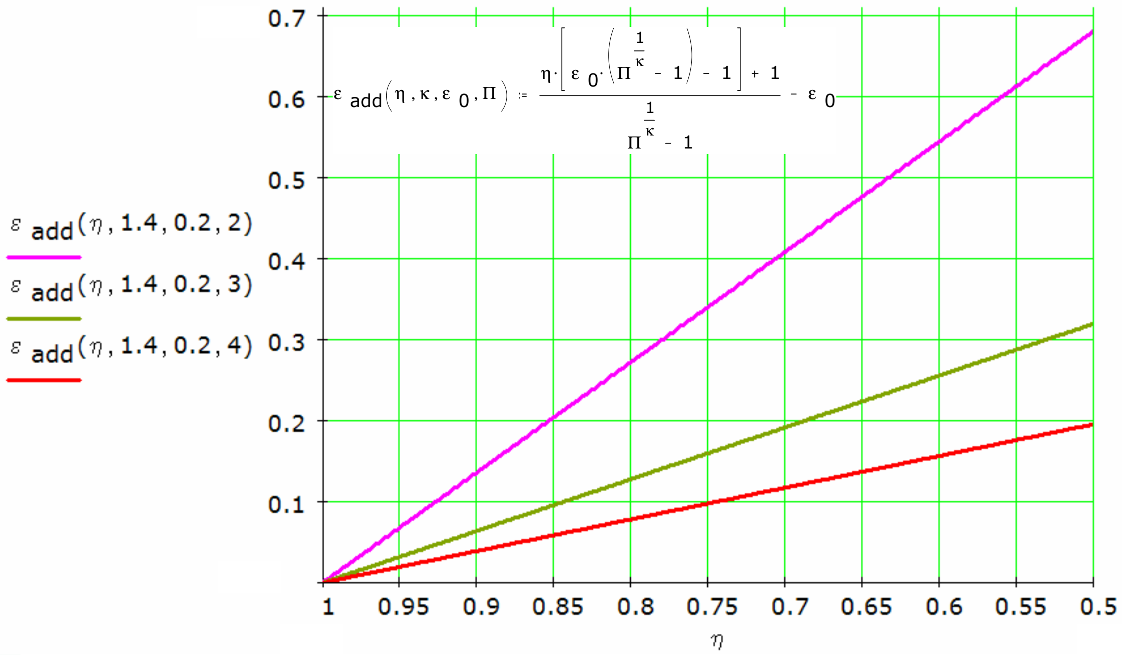
With constant inlet pressure and pressure ratio, the necessary additional clearance can be calculated from this proportional relation.

The graph shows the nessary additional clearance as parts of the compartment‘s swept volume over the relative capacity, here for an initial relative clearance of 0.2 (20% of the swept volume), the pressure ratio 3, constant inlet pressure, and isentropic exponents of 1.6, 1.4, and 1.1.

CLEARANCE VOLUME CONTROLS: Thermodynamics

Necessary Additional Clearances for Capacity Reduction (Per Stage)



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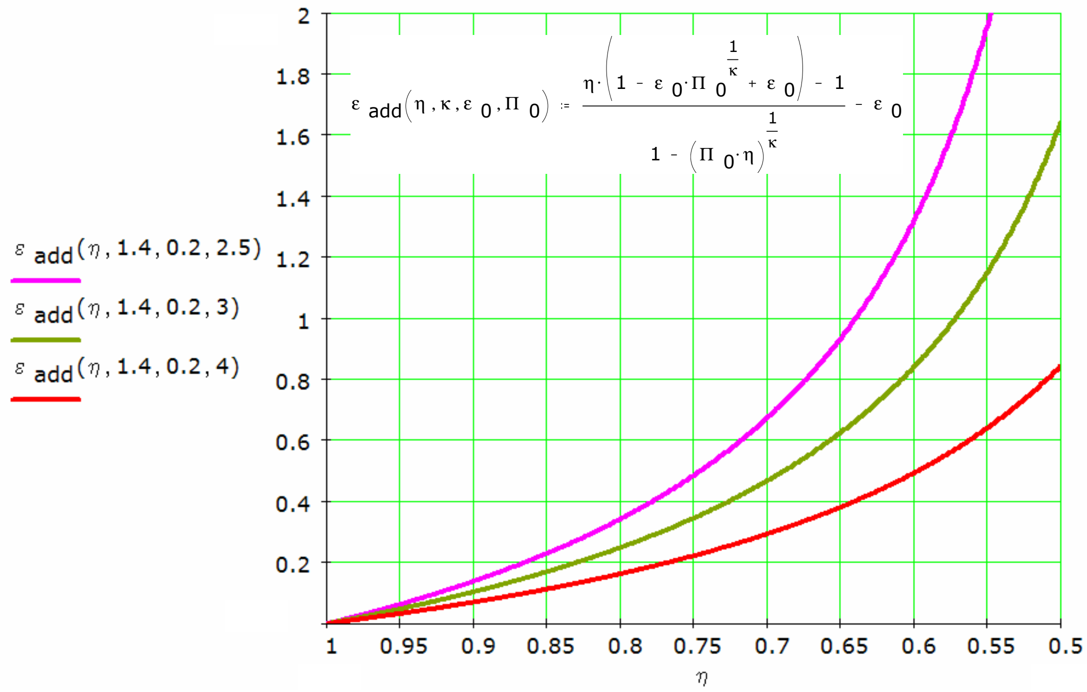
The same function, this time with constant isentropic exponent and pressure ratios 2, 3, and 4.

CLEARANCE VOLUME CONTROLS: Thermodynamics

Necessary Additional Clearances for Capacity Reduction (First Stage of Multi-Stage Compressor)



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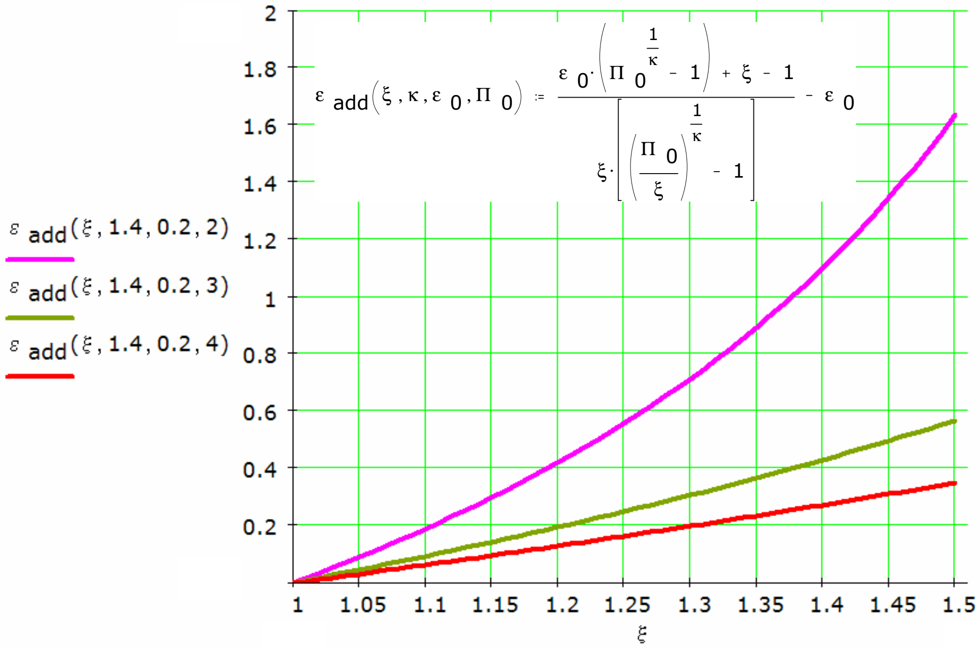
Similar to the first stage bypass control, here the capacity is governed by the inlet pressure 2nd stage.

Therefore the equation takes into account the proportionally reduced pressure ratio, leading to substantially higher additional clearance volumes.

CLEARANCE VOLUME CONTROLS: Thermodynamics

Necessary Additional Clearances for Inlet Pressure Variation

(First Stage of Multi-Stage Compressor, Constant Demand)



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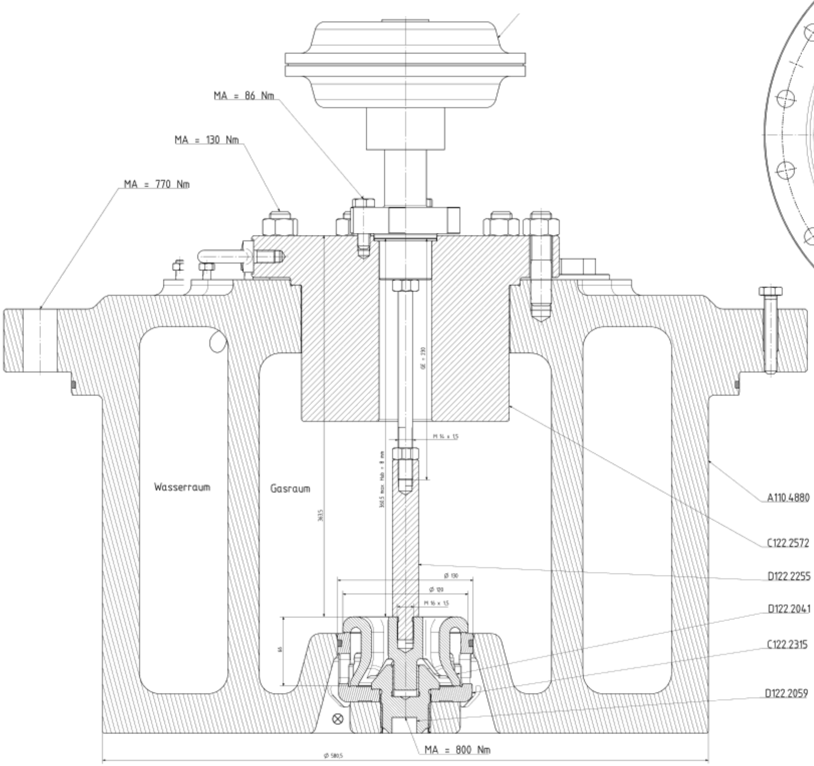
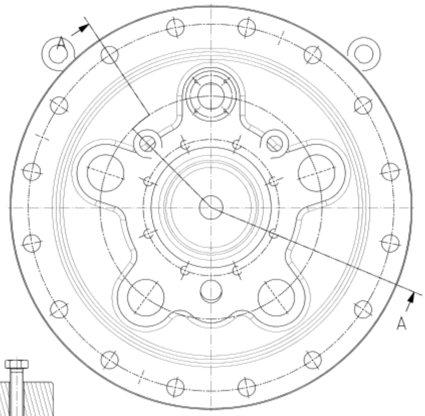
To achieve the same capacity at increasing suction pressure and constant end pressure first stage, the resulting extra clearance volume is shown as a function of the relative pressure increase ξ.

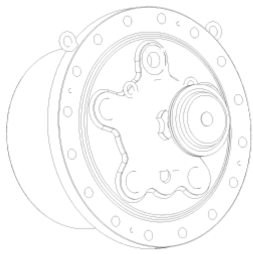
CLEARANCE VOLUME CONTROLS:

- Fixed Clearance Pocket with On/Off Valve
- Variable Volume Clearance Pocket
- Fixed Clearance Pocket with Controlled Valve

Fixed Clearance Pocket In Cylinder Cover [NEA]







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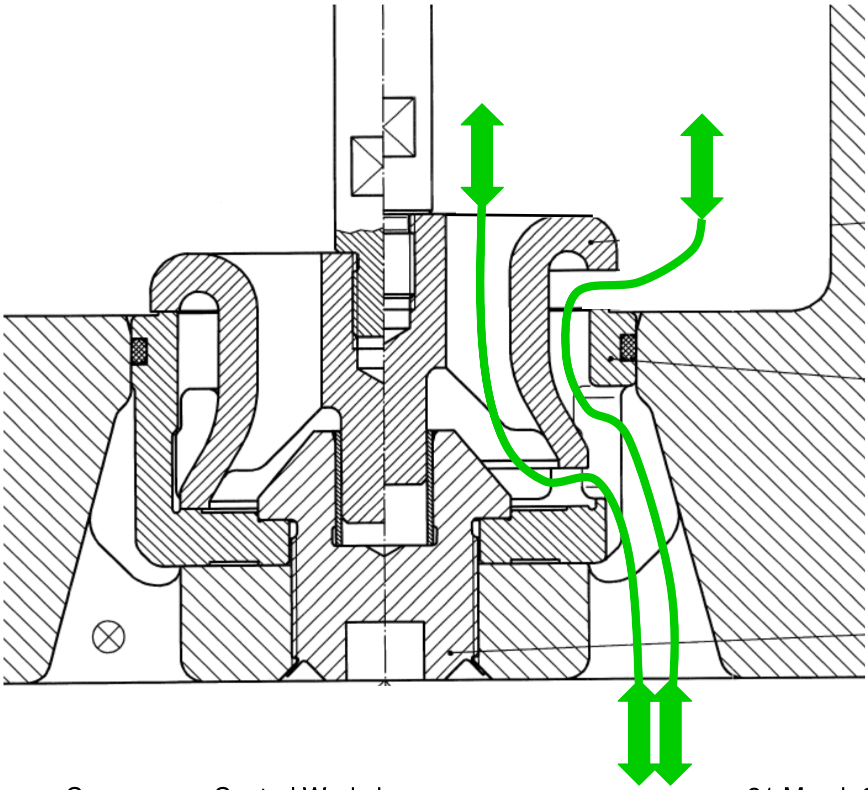
This shows the most common (because easiest designed) type of a fixed clearance pocket in the cylinder cover.

The special valve is kept in open/closed position by means of a pneumatic diaphragm cylinder,

Fixed Clearance Pocket: Semi-Balanced Breathing Valve

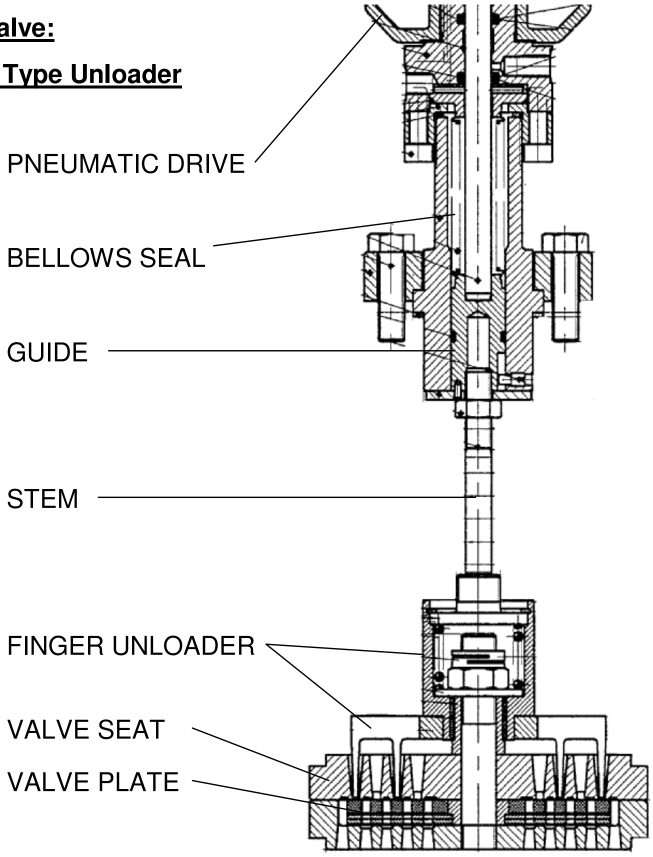


Here the double passages of the valve can be seen.

The semi-balanced plug design needs only small forces to keep the valve in position.

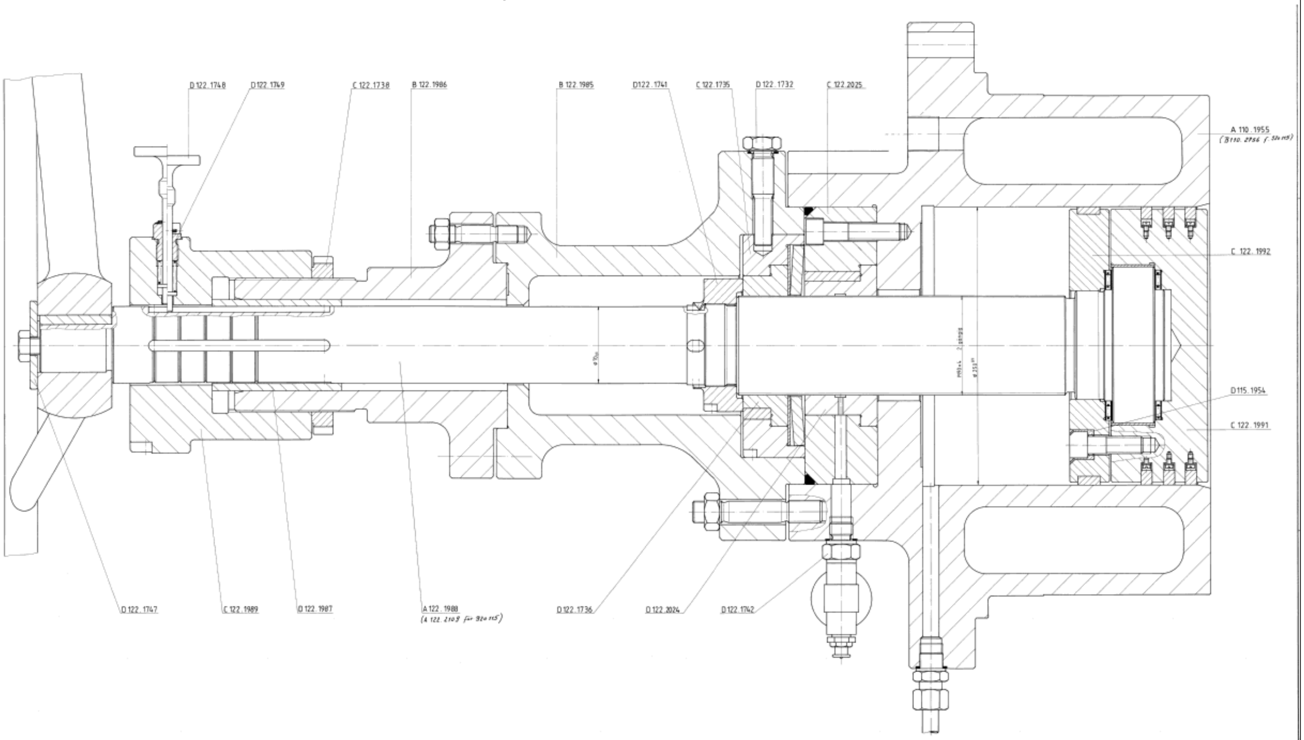
Fixed Clearance Pocket Valve:

Suction Valve with Finger Type Unloader



This assembly uses valve and diaphragm actuator identical with standard suction valve unloaders.

Variable Clearance Volume In Cylinder Cover, Piston- Controlled



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This piston type VVCP replaces – as the fixed clearance pocket shown previously – the head end cylinder cover.

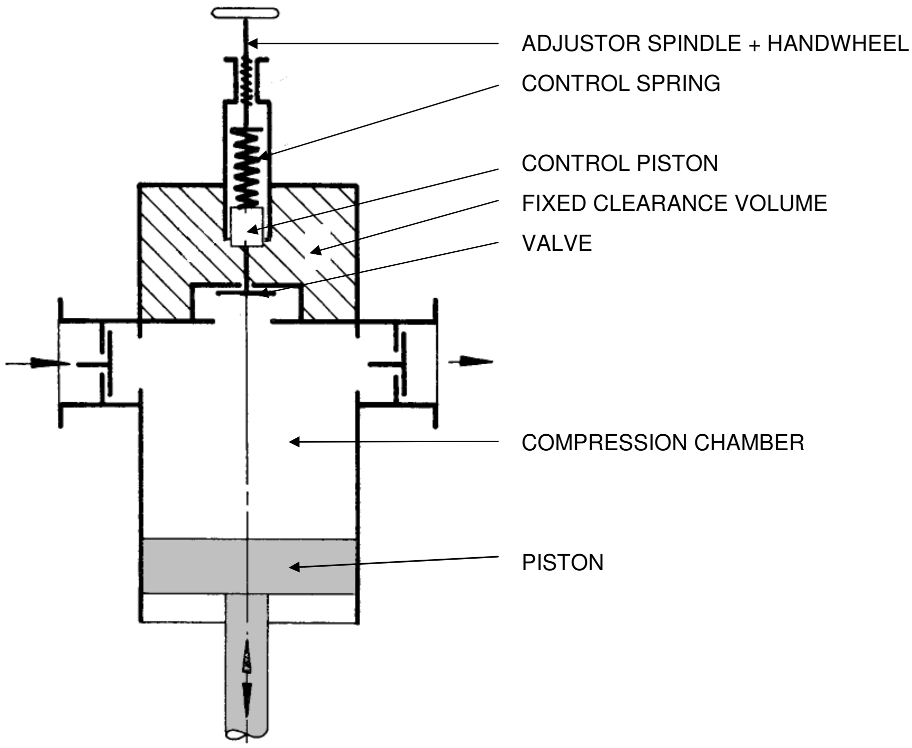
It allows he stepless adjustment of the additional clearance by changing the axial position of the control piston.

This is done manually by turning the hand wheel in steps of quarter revolutions.

The high dynamic forces exerted by the pulsating pressure in the compression chamber (right of the piston) call for a rugged design with ample safety regarding variations of operating conditions.

Fixed Clearance Volume with Stepless Opening Valve

Function Principle



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(This and the following sketches and diagrams were taken from a Hoerbiger leaflet.)

This rather elegant system controls the capacity by varying the clearance volume during compression and re-expansion.

The control piston is loaded by the internal pressure of the fixed clearance volume (hatched), and balanced by the control spring and/or external control pressure. During the suction stroke the valve is open, resulting in an enlarged clearance.

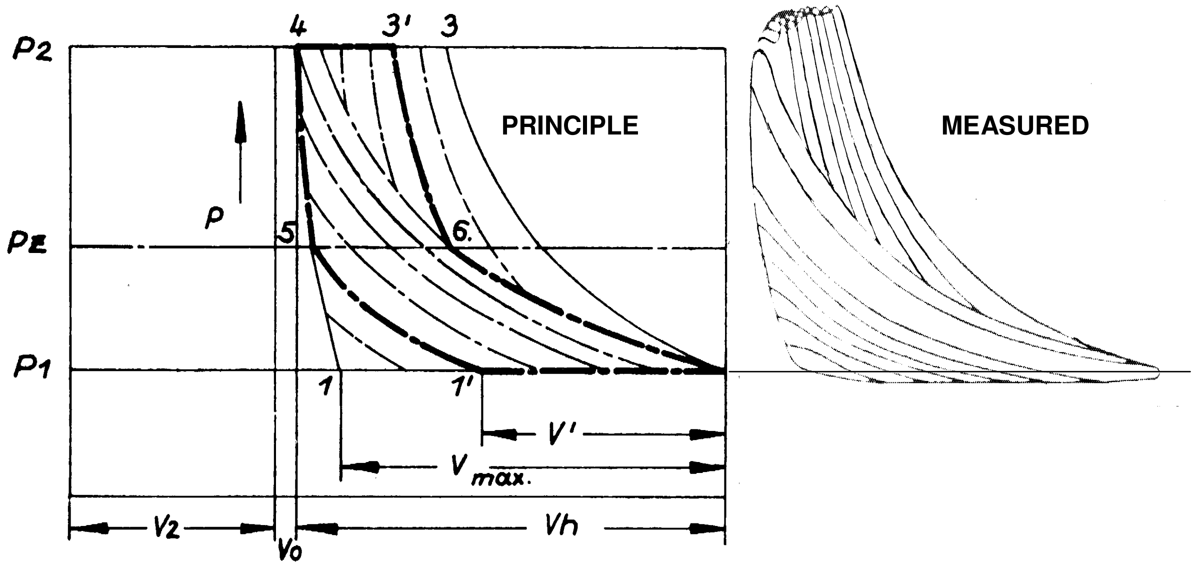
At the beginning of the compression stroke therefore the pressure increases simultaneously in the compression chamber and in the fixed clearance volume.

When during compression the pressure in the fixed clearance volume reaches a level defined by the control piston area and the preload on the valve, the valve closes, leaving the clearance volume with an intermediate pressure and the compression chamber with its own original clearance.

Compression and discharge now follows the ‚normal‘ path.

This adjustment of the clearance volume is automatically reverted during re-expansion, resulting in a smaller volumetric efficiency and capacity.

Fixed Clearance Volume with Stepless Opening Valve: PV Characteristic



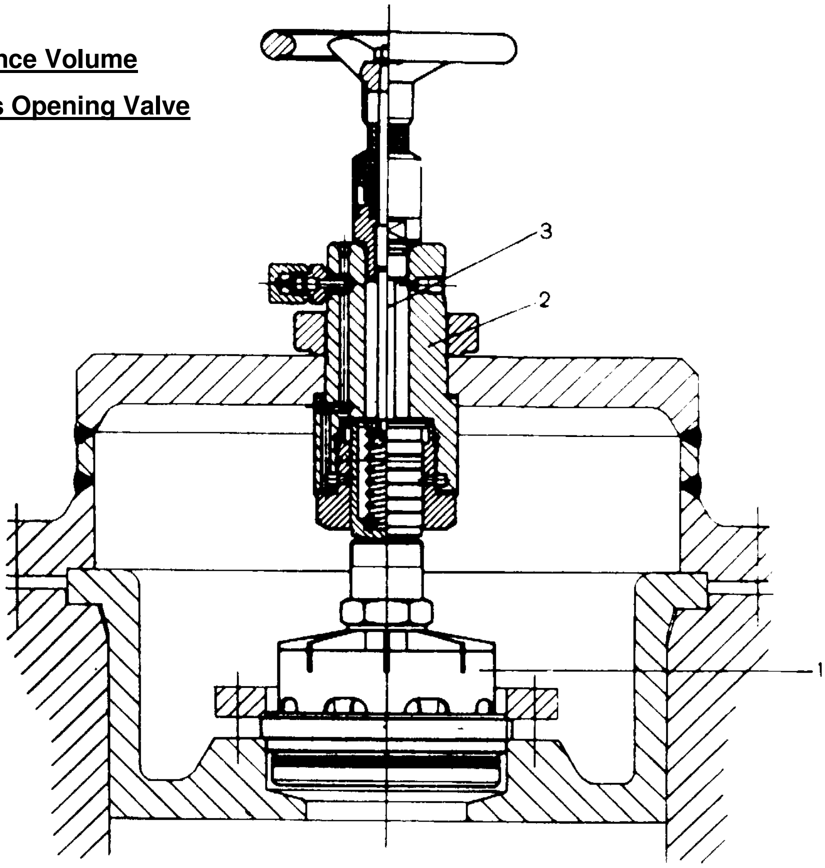
The higher the closing pressure, the lower the capity:

Simply by adjusting the preload on the control piston, either by adjusting the spring compression or by changing the auxiliary control pressure, capacities from 100% to 0% can be achieved.

Fixed Clearance Volume

with Stepless Opening Valve

[Hoerbiger]



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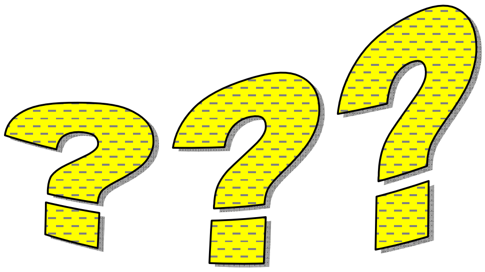
This sketch shows the execution with a finger-controlled suction valve, a control piston with bellows seal to the atmosphere, control spring with hand wheel adjustment, and an additional pressure connection for pneumatic adjustment of the closing pressure.

CLEARANCE VOLUME CONTROL



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