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Stepless Variable Capacity Control

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INTRODUCTION

This paper discusses two different methods of stepless variable capacity control systems for reciprocating compressors. When using reciprocating compressors it would be desirable if cylinder displacement could be varied infinitely in order to meet the momentary demands which may be dictated by various criteria:

- maintaining suction or discharge pressure within very close limits,
- quantity of gas required,
- quantity of gas available,
- horsepower available, etc.

These requirements are generally met today by installing additional clearance pockets, by unloading of cylinder ends, or by varying the speed of the compressor within certain limits. None of these systems, however, meet the real demand with sufficient accuracy.

REVERSE FLOW REGULATION

This system utilizes pneumatic pressure to hold the suction valve open during part of the compression stroke, thereby delivering part of the intake volume back into the suction line. The reverse flow of the gas through the suction valve causes a dynamic pressure on the valve plate, which increases with the piston speed and reaches its maximum at approximately mid-stroke. The valve plate is kept open by means of an unloader and a spring positioned between the unloader and the actuator. At the moment when the increasing dynamic pressure on the valve plate overcomes the springload, the suction valve closes, and the gas remaining in the cylinder is compressed and delivered through the discharge valve.

Fig. 1 shows the low mass pin unloader which is used in low pressure applications and high speed compressors. Through extensive research and experience gained in numerous installations in the field and the resulting theoretical derivations for valve plate designs and valve lifts, the life of regulated and unregulated valves is equal.

One of the new developments was a new valve design series with a separate movable outer part. This allows a delayed closing, caused by the unloader, of the inner portion only; whereby the outer part of the plate moves as in an unregulated valve (Fig. 2).

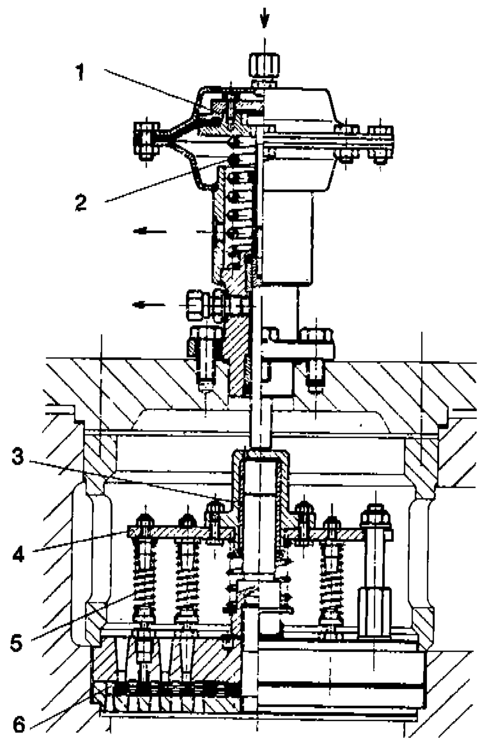

FIG. 1: Suction Valve with Unloader and Actuator for stepless reverse flow regulation
1 = Diaphragm Piston, 2 = Damping Spring,
3 = Pin Unloader, 4 = Pressure Plate,
5 = Unloader Springs, 6 = Valve Plate

The dynamic pressure on the valve plate during the reverse flow is proportional to the density of the gas at intake conditions, a flow coefficient of the valve, and the square of the gas velocity through the valve.

$$F \cdot c = fe \cdot V$$

$$V = \frac{F \cdot c}{fe}$$

F = piston area

c = piston speed

fe = valve lift area

V = gas velocity through valve

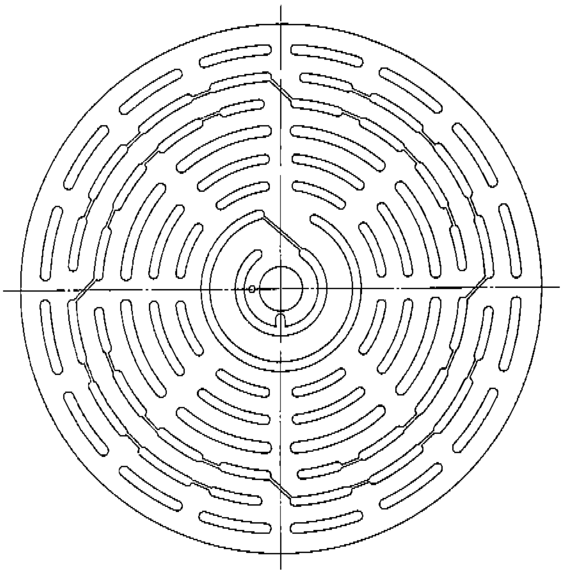

FIG. 2: Partially regulated valve plate

The maximum piston speed (Fig. 3) determines the limit for this type of capacity regulation, since an increase of the force acting on the valve plate keeps the valve open through the whole discharge stroke. Depending on the cylinder end (head- or crankend), the maximum piston speed is reached before or after mid-stroke.

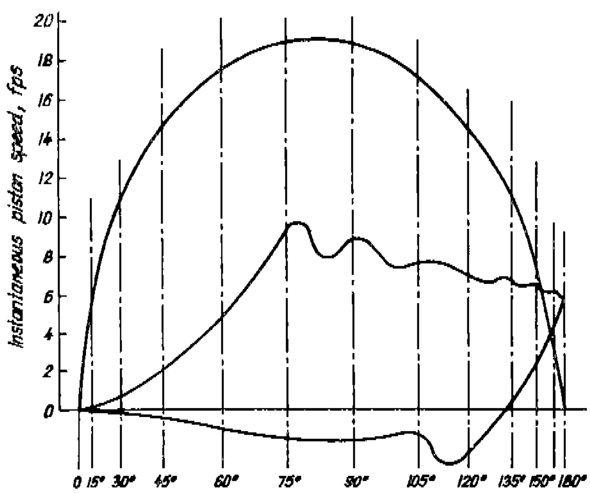

FIG. 3: Instantaneous compressor-piston velocity for a given crank position having a 5:1 connecting rod to crank throw and a 12-in. stroke, at 360RPM

The stepless capacity regulation can therefore be activated between the beginning of the discharge stroke and the point of maximum piston speed. The intake volume which is smaller than the full piston stroke by the volumetric efficiency, can be reduced by more than 50 percent. The regulation range is therefore:

$$R = \frac{50}{Ev}$$

For example: $Ev = .8 \qquad R = \frac{50}{.8} = 62.5\%$

Hence, it is theoretically possible to vary the capacity steplessly from 100% - 37.5%. On medium and high speed compressors this span is greater since the dynamic pressure curve is out of phase with the piston speed curve. On double-acting cylinders the regulation range can be increased by unloading the crankend completely at 50 percent capacity, the headend is then regulated again over its full range, which is:

$$RD = \frac{50}{Ev \cdot 2} \quad KD$$

KD is a correction factor from Fig. 4, which compensates for the ratio of crankthrow to the connecting rod length. Figs. 5 and 6 show a theoretical and an actual pressure-volume diagram, respectively, of a cylinder end equipped with a stepless reverse flow regulation. The enclosed area of a PV-diagram is proportional to the energy consumed for compression work, therefore, these diagrams also demonstrate the savings of horsepower.

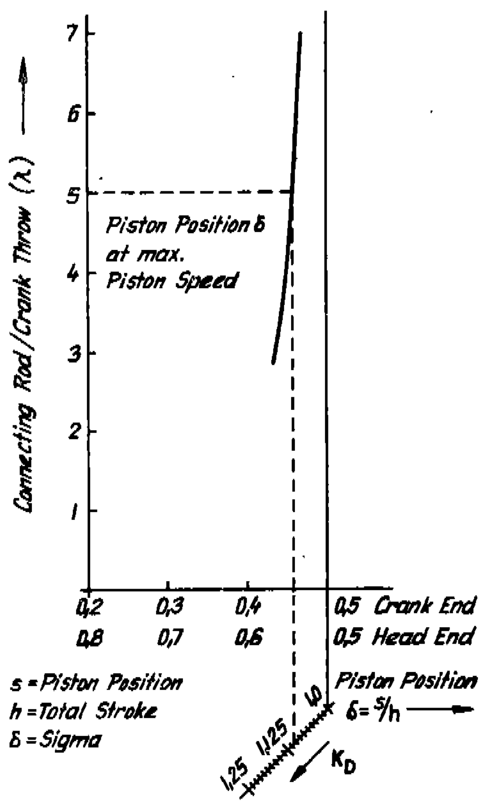

FIG. 4: Correction factor for ratio of connecting rod length to crank throw (λ = 5, KD = 1.11)

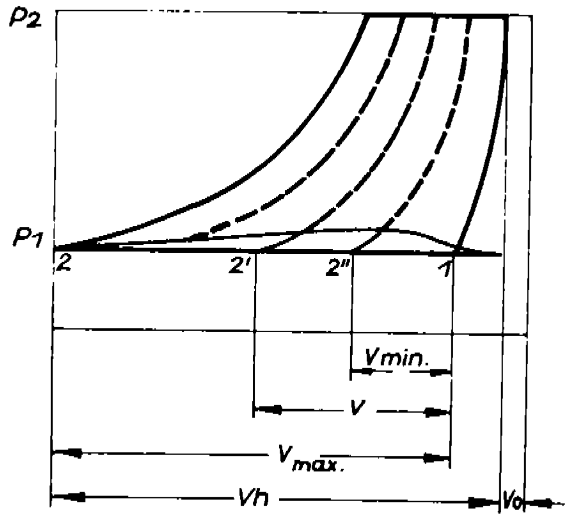

FIG. 5: Theoretical PV-diagram for reverse flow regulation; P1 = suction pressure, P2 = discharge pressure, VH = piston displacement, V_{max} = actual capacity, V_{min} = min. capacity with reverse flow regulation, Vo = clearance;
1 = suction valve opens, 2 = suction valve closes at full capacity, 2' and 2" = suction valve closes at partial capacity

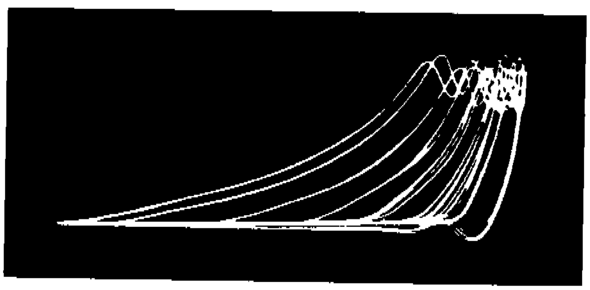

FIG. 6: Actual PV-diagram for reverse flow regulation

Stepless Regulation from 100% - 0%

By adding a fixed clearance volume to lower the volumetric efficiency of the headend to Ev = .5, a stepless variable capacity control from 100% - 0% can be accomplished. Fig. 7 shows the PV-diagram for this arrangement.

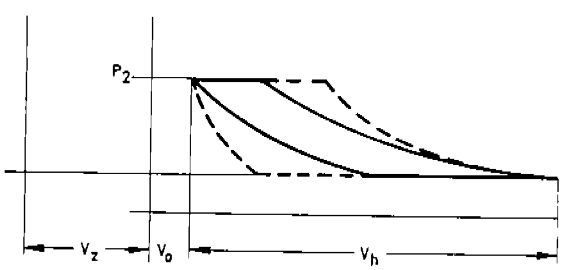

FIG. 7: PV-diagram for EV = .5
Vo = clearance volume
Vz = clearance pocket volume

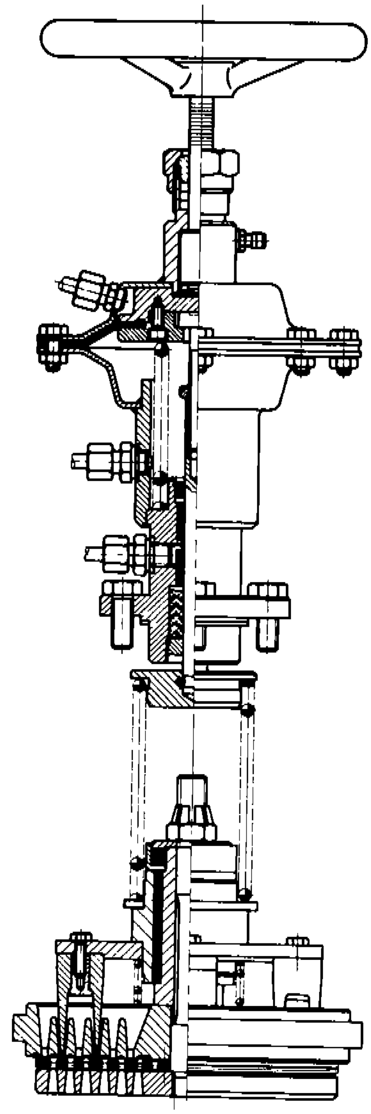

FIG. 10: Suction valve with unloader and actuator for stepless reverse flow regulation on non-lube compressors

The necessary unloading force is generated in a diaphragm actuator by a variable, but steady, pneumatic pressure. The unloading pressure determines the closing time of the suction valve and has to be varied according to the capacity required. Fig. 8 shows the necessary unloader force (which is proportional to the air pressure on the diaphragm) in relation to the capacity of the compressor.

For a capacity range of 100% - 60%, this diagram shows sufficient linearity to assure adequate performance of the system. If the regulation span is expanded to 100% - 40%, a special pressure limiter has to be installed which stretches the portion from 60% - 40% so it can be used on the actuator (Fig. 9).

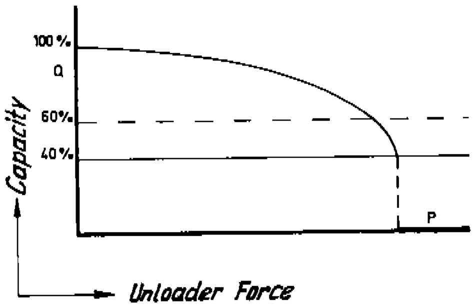

FIG. 8: Diagram unloader capacity/force

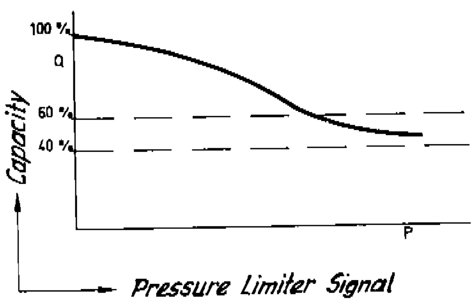

FIG. 9: Diagram capacity/pressure limiter signal

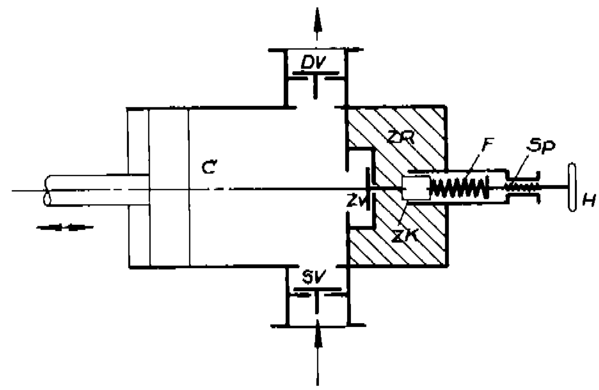

FIG. 11: Schematic of a stepless clearance volume regulation; C = piston, SV = suction valve, DV = discharge valve, ZR = clearance pocket, ZV = clearance pocket valve, ZK = unloader piston, F = adjustable spring

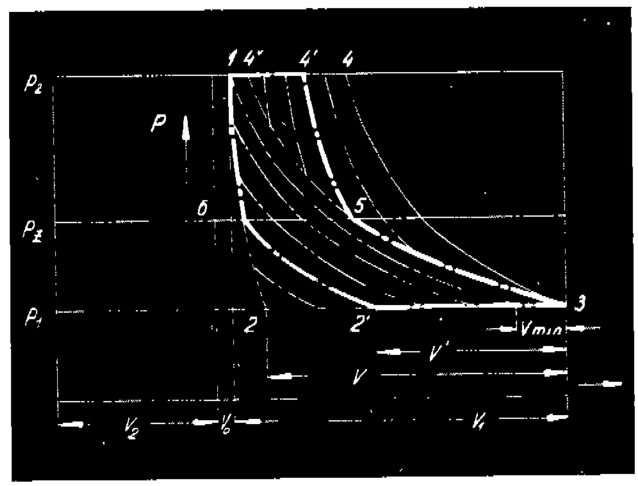

FIG. 12: Theoretical diagram of a clearance volume regulation

CLEARANCE VOLUME REGULATION

Clearance pocket regulations for capacity control are widely used. Most installations equipped with clearance pockets can separate those pockets from the cylinder with a plug type valve. In essence the volumetric efficiency is lowered and so is the output of a compressor. The quantity of gas accepted by a clearance pocket can be determined not only by its size, but also by the pressure inside the pocket. The pressure can be varied if the clearance pocket is separated from the cylinder at a certain point of the discharge stroke of the piston. Fig. 11 shows this type of arrangement. A clearance pocket valve (a regular suction valve) is installed between the clearance pocket and the cylinder. An unloader which actuates the valve plate is positioned over this valve and an unloader piston pushes against the actuator. The unloader piston can be actuated by means of a spring or by pneumatic pressure. This actuating load has to be varied depending on the required capacity.

Figs. 12 and 13 show a pressure-volume diagram of a stepless clearance volume regulation. The clearance pocket valve is kept open (pts. 3-5, Fig. 12) by the unloader during the compression stroke until the gas pressure in the clearance pocket acting on the unloader piston overcomes the preload of the spring (pt. 5, Fig. 12). At this time the valve closes and the remaining piston stroke is used to further compress (pts. 5-4', Fig. 12) and discharge the gas into the discharge line (pts. 4'-1, Fig. 12). During the suction stroke when the cylinder pressure has fallen below the pressure in the clearance pocket (pt. pz, Fig. 12) the regulating valve opens again allowing the gas trapped in the clearance pocket to flow back into the cylinder (pt. 6-2', Fig. 12). The intake volume will be reduced by the volume of this gas. By progressively altering the force which keeps the regulating valve open, any intermediate capacity between full load and no load can thus be obtained with infinite variation, depending on proper sizing of the clearance pocket.

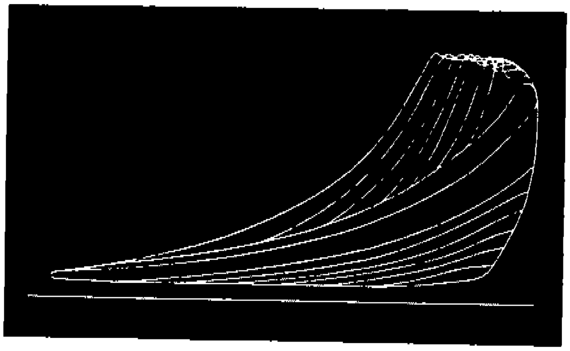

FIG. 13: Actual indicator diagram of an additional clearance volume regulation with limited regulating range

Fig. 14 shows an interesting possibility of accommodating the required clearance volume for a single-acting cylinder directly into the suction duct. The regular suction valve is used as the regulating (clearance pocket) valve, and the clearance pocket is defined by the volume between the check valve and the suction valve.

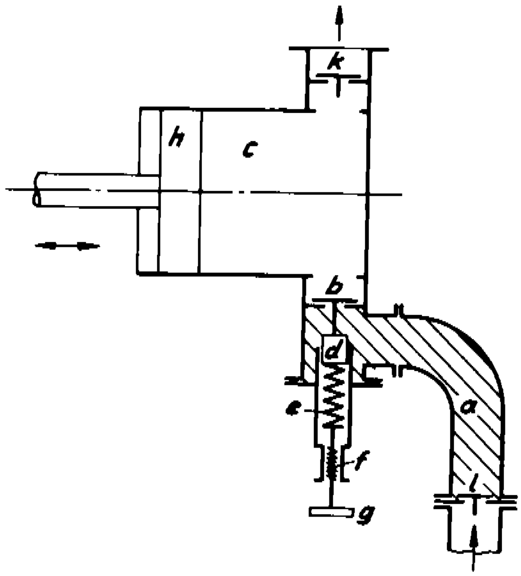

FIG. 14: Additional clearance volume regulation with direct manual operation on the machine and clearance pocket being part of the suction duct. a = additional clearance volume, b = regulating valve, d = unloader piston, e = spring, f = spindle, g = handwheel, i = suction valve, k = discharge valve

Required Size of the Clearance Pocket

The size of the clearance pocket is determined by the range of capacity regulation, fixed clearance volume, compression ratio, and the type of gas compressed.

$$R = 1 - \frac{Vmin}{V}$$

$$\rho o = \frac{Vo}{VH}$$

$$Vz = VH \cdot R(\rho'z - \rho o)$$

R = regulation range
Vmin = minimum capacity
V = maximum capacity
ρo = fixed clearance volume in %
Vo = fixed clearance volume
VH = piston displacement
Vz = required clearance volume

$\rho'z$ can be found in Fig. 15, whereby m = exponent of polytrope.

For constant pressure ratios the clearance volume can also be determined by:

$$Vz = VH \cdot R\left[\frac{1}{\left(\frac{p2}{p1}\right)^{\frac{1}{m}} - 1} - \rho o\right]$$

If the pressure ratio changes during the regulating process, the following formula has to be used:

$$Vz = VH\,\frac{\rho o\left[\left(\frac{p2}{p1}\right)^{\frac{1}{m}} - \left(\frac{p4}{p3}\right)^{\frac{1}{m}}\right] + R\left\{1 - \rho o\left[\left(\frac{p2}{p1}\right)^{\frac{1}{m}} - 1\right]\right\}}{\left(\frac{p4}{p3}\right)^{\frac{1}{m}} - 1}$$

It can be seen that on large cylinders with small pressure ratios (below 2) and a wide regulation range, the required volume becomes rather large. On large single-acting cylinders it is therefore advisable to use the solution as shown in Fig. 14.

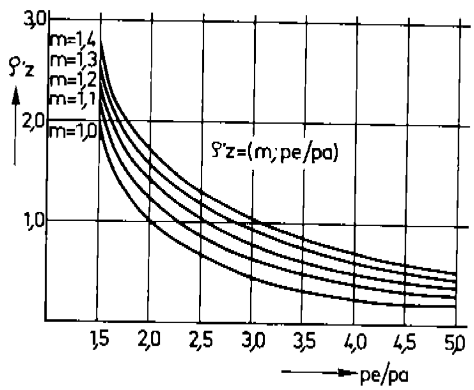

FIG. 15: Diagram for sizing of clearance pocket

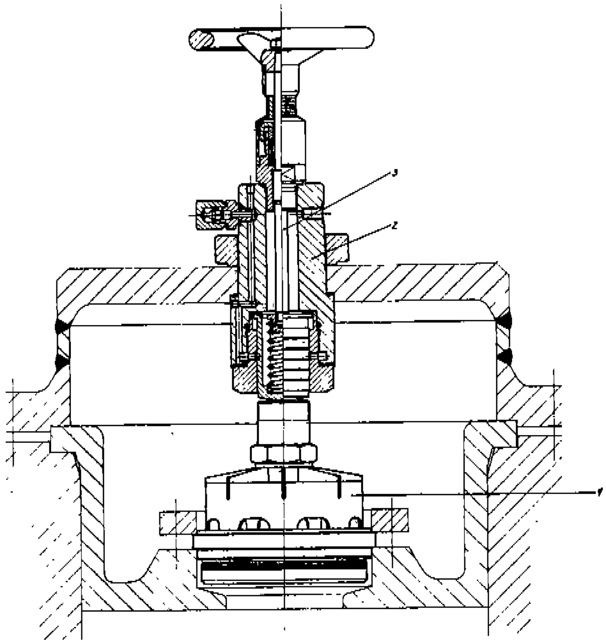

FIG. 16: Clearance volume regulation with servo-cylinder and additional screw spindle for manual operation:
Clearance pocket in the cylinder cover
1 = regulating valve with unloader, 2 = servo-cylinder with piston, 3 = screw spindle and hand wheel

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