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CONTINUOUS AND DISCONTINUOUS CAPACITY CONTROL FOR HIGH SPEED REFRIGERATION COMPRESSORS

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Abstract
The objective of this theoretical investigation was to find a capable and economical way of controlling high speed refrigeration compressors with a low number of cylinders.

Compressors with 4 cylinders or more allow control by cutting-off one or two cylinder blocks because the remaining blocks maintain a sufficient smoothness of low speed fluctuation. The type of regulation to be found in this work should therefore apply primarily to two- and three-cylinder-compressors.

Up to now the control of such compressors was performed only by external control-devices. These types of control do not affect the smoothness of low speed fluctuations, because they engage either before or after the compression. Application of external control affects all cylinders or only some of them with respect to an application before or after the compression. External controls are characterized by a high energy input. Therefore they are economically not reasonable. Examples for those are throttling of the gas in the suction line and a discharge to suction by-pass.

To be economically efficient an internal controller action affecting the process of compression is preferred. Such an internal controller action for low numbers of cylinders is supposed to affect all cylinders simultaneously in order to achieve a sufficient smoothness of low speed fluctuations.

1. INTRODUCTION
A refrigeration compressor is generally designed for maximum demand of refrigeration capacity. The target of the controlling process is to achieve the proper mass flow rate of refrigerant, which is needed to suffice the refrigeration demand under the current conditions of operation. The control of the mass flow rate always results in a reduction compared to the full load specification.

The most important requirements for a control method are the constance of the adjusted part load and an economical energy consumption. Problems in meeting all requirements simultaneously and completely are the reason for a great number of control methods developed. Especially for larger gas compressors large-scale control methods are worthwhile, because high investment and operating costs result in an earlier amortization.

Only some of these control methods are applied in refrigeration compressors, because they are generally of smaller size, so that the control device takes a considerable part of the overall costs. Growing efforts in energy conservation lead to an increasing demand for economically controllable refrigeration compressors. Depending on the valve type (ring valve or flapper valve) two different control methods dominate.

In the ring valve-type compressor a pair of single cylinders are unloaded by a suction lifter mechanism. Flapper valve-type compressors are controlled by a blocked suction port. Both control methods are applicable to compressors only with 4 or more cylinders, which do not have a negative affect on smoothness of low speed fluctuations upon cutting off cylinders. They cannot be applied for a low number of cylinders, because they result either in an increase of the discontinuity or in a zero delivery due to the cutting-off of one or two cylinders.

2. SURVEY AND CALCULATION OF POSSIBLE CONTROL METHODS FOR RECIPROCATING COMPRESSORS

With respect to the nature of variation of the flow rate delivered by the compressor the following control methods are considered:
- intermittent control by interrupting the delivery occasionally
- stepwise variation of the flow rate
- continuous variation of the flow rate

The following table lists a systematic classification of methods for controlling the flow rate of reciprocating compressors /1/.

2.1 External Control
2.1.1 Controlling by Affecting the Drive
There are two ways of controlling the delivery, which vary the flow rate by driving motor action: intermittently stopping the compressor and varying the speed.

2.1.1.1 Variation of Speed
The best and easiest way of controlling the flow rate delivered by a compressor is achieved by speed variation. It must be required that the variation does not result in an efficiency loss of the driving unit, which is possible by employing for example engines or direct-currentmotors.
Due to the constant speed of commonly used three-phase or two-phase motors the flow rate control for refrigeration compressors would require a frequency converter. Since there is an internal loss in frequency converters resulting in a decreased efficiency of the entire driving unit of approximately 10%, there are energetic losses although this method of speed variation is considered ideal. Furthermore there are still high costs for electrical control devices. Therefore such an application is presently not economically feasible.

Stepwise speed variation can be realized by motors with commutable poles. These motors generally have a worse efficiency than motors with constant speed. Varying the speed of a compressor, either continuously or stepwise results in a corresponding change in flow rate. Speed variation does not require changes in the compressor design.

Speed control is the most economic method for varying the flow rate of reciprocating compressors. Firstly friction losses decrease proportionally with the flow rate. Secondly a decrease of speed is associated with a decrease of the gas velocity in the valves and piping. This reduces the pressure losses and the indicated work per rotation. Thus the effective power demand of a compressor decreases more than the corresponding flow rate. In practice this sinking demand in specific power needed for the compressor is balanced by an increasing demand for the motor. Thus the resulting demand in specific power remains (nearly) constant with dropping speed.

The lower limit of part load is not set by the control system, but by the compressor and the drive unit. Dropping speed causes an increase in discontinuity. Therefore compressors with speed control mostly require a larger flywheel with a higher moment of inertia. Bearings supporting the crankshaft and connecting-rod have to be designed for the entire speed range. Furthermore it has to be considered that operation at certain low speeds causes fluttering of the valves while high speeds increase their impact load.

Inspite of the advantageous specific power requirement during partial loading, both infinitesimally variable and stepwise speed controls are at a disadvantage due to 1. the larger rotating masses, 2. less desirable starting conditions and 3. a lower efficiency during partial loading for the drive motor occurring from losses, if the occasion arises, from the frequency converter. An economical control range lies between 50 and 100% of the maximum flow rate /1,3/.

2.1.1.2 Intermittently Shutting the Compressor

---On and Off------------------------

This type of flow regulating is the most often used method in refrigeration compressors. It intermittently delivers either a no flow or a full flow condition, and can be reached in following 2 ways: A. By shutting down the drive motor and B. by uncoupling the motor from the compressor using a clutch.

The latter method is used frequently in air compressors and is conceivable in refrigeration compressors driven by an internal combustion engine, when the lowest turning speed is reached.

From an energetic standpoint the first method, shutting the drive motor on and off, is the best type of regulating, since an optimum efficiency prevails at full load and since no energy is consumed during shutdown. This is limited however to how often and in which manner the starting is executed. The wear on the aggregate increases also with higher switching frequencies. High starting electrical currents, high thermal loads on the regulating devices and an unfavorable ratio between increased current losses to the energy saved limits this type of on/off regulating system.

The coupling method between the drive motor and the compressor has the advantage that the starting torque and thus the current (peaks) are kept smaller. Its advantages, however, are the necessary and expensive clutch and the no-load losses during idling. As a result regulating the compressor through a clutch is not as economical as the on/off method, since no load work is wasted during zero flow in the compressor. However, this regulating method using a clutch is more economical due to the lock of the compressor's friction than all the following described methods, where the rotating compressor is brought into a no flow condition.

2.1.2 Regulating by Influencing the Compressor's Intake and Outtake

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By influencing the intake and outtake ports of the compressor it is possible to lower the flow rate. This method calls for either the intake being totally or partially closed or the outtake being totally or partially opened.

2.1.2.1 Throttling the Intake

The throttling of the intake according to Figure 1 can vary the flow rate uniformly. The more the intake is throttled, the more the specific compressor work rises as can be seen in the diagram (log p versus h) /4/. Consequently this method becomes less efficient with a dropping pressure ratio.

At higher pressure ratios the power does indeed drop with decreasing flow rates, but its useful range is unfortunately limited by the largest allowable output temperature. Therefore the useful range of the throttling is bound on 2 sides and becomes smaller, the larger the pressure ratio is at maximum capacity. This method to control the flow rate in an even

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continuous manner can be easily and cheaply constructed. Its largest drawback is an unfavorable high specific power demand during partial loading.

During partial load it must be considered, that a vacuum occurs in the suction line, which could contaminate the compressor with outside air. In spite of these handicaps this type of regulating is often employed. Throttle regulators are for refrigeration compressors commercially available.

2.1.2.2. Shutting Off the Intake

If the intake is completely shut off, the flow rate will be completely stopped and therefore the compressor will start to idle. This regulating works intermittently. After shutting the intake off the flow does not respond immediately to a full stop rather it drops to zero after a few revolutions. This lagging response comes from the time needed to evacuate the cavity between intake valve and the cutoff /5/.

This type of regulating is only used for air and gas compressors, so that for example the drive engine does not have to be turned off.

Gas leaking from an insufficiently seated shut off valve is the main drawback for this system, since due to the large pressure ratio, the exit temperature of the minute gas quantities has the danger of becoming too hot. In order to reduce the exit temperature as well as the idling work, the intake valve off regulating can be combined with a by-pass, which equalizes the low and high pressure sides of the circuit.

2.1.2.3. High to Low Side Equalization

Flow rate control by high to low side equalization can be achieved using two procedures. In the first one the by-pass valve is fully opened, and as a result the compressor begins to idle (Figure 2a). This regulating method can be either constructed as an external control or can be integrated into the compressor housing itself (see Figure 3b). The procedure is a member of the intermittent regulating class /6/.

In the second procedure the by-pass valve is only partially open and thus only a portion of the compressed gas is released to the suction side (Figure 2b). This is a stepless control method.

In both procedures a conducting pipe contained in a by-pass valve connects the high pressure side to the suction side.

Complete By-Passing

The by-pass with a fully opened valve is the simplest possibility to achieve idling and is mainly used in large high-capacity compressors to ease starting. This regulating does not cause low speed fluctuations of the machine, since the by-passing starts after the compression process in the cylinder and therefore no change in internal mass forces occur.

If the by-pass and its valve possess a sufficient cross-sectional area, then the discharge's pressure will be only slightly higher than the suction's. By-pass regulators are shown in Figure 3 for both internal and external designs.

This type of regulating can also be combined with the method of cutting off the intake, since circulation of the gas and thus its associated energy losses can be avoided.

It should be pointed out again, that the vacuum occurring in the cylinder can unfortunately lead to sucking air in the refrigeration system.

Throttled By-passing

By means of a throttled by-pass the flow rate can be uniformly reduced. The compressor power input remains throughout the entire operating flow range constant due to the unchanged internal compression. Therefore this type of regulating is quite uneconomical. First all pressures and temperatures remain unchanged during the compression and the rotating smoothness does not become worse.

On the other hand the suction conditions change and increased discharge temperatures of the compressed gas occurs. Therefore injection of liquid refrigerant into the suction line is necessary (Figure 2b).

This kind of capacity control is often applied to reciprocating compressors with a low number of cylinders.

2.2. Internal Capacity Control

2.2.1. Variation of the Compressed Gas Quantity

This type of capacity control reduces partially or stops totally the flow by returning the already drawn or compressed gas back into the suction manifold. The most frequently used method is to keep the intake valves open.

For this type of forced valve action a relatively complicated mechanism is required. Therefore, this is mainly found in larger compressors and is not represented at all in compressors having flapper valves.

Basically an analogous method on the discharge side is possible. Keeping the outlet valves open results in drawing back the expelled gas from the discharge manifold.

2.2.1. Controlling the Intake Valves

Capacity control by manipulating the intake valve is applied in the following ways: a) keeping it completely open, b) partially open it, c) open it up for part of the compression stroke. The first method does not compress or deliver gas to the discharge manifold. Consequently it deals in a type of intermittent capacity control. In contrast to this, both the other two methods allow stepless variable capacity control.

Fully opened intake valves are preferably used for an unloaded starting. Furthermore, this method is employed as an automatic capacity control in small and medium sized compressors (Figure 4). Here a piston or membrane being controlled through a regulator using a pressure medium adjusts the valve by means of a lever towards the valve stop. The technical expenditures in this design are relatively high.

The flow rate can be uniformly decreased by partially opening the intake valve. Since the largest percentage of gas flows back at the end of the compression during delivery it is only somewhat more economical than the capacity control using a by-pass.

A similar effect is achieved by prematurely shutting off the intake valve. A technical method is elaborated in /9/ to the effect, that the suction port is prematurely closed by means of a slider. The technical expenditures are unfortunately high.
In the third method of flow control, the inlet valves are held open for the entire intake stroke and for the beginning phase of the compression stroke. Consequently the gas can be expelled out of the cylinder at the beginning of the compression stroke (Figure 5a). The intake valves can be closed at a particular point during the compression stroke and the remaining gas in the cylinder is compressed. By changing the opening time of the intake valve, the compressed and circulated gas quantity can be varied in an economical way, since the consumed energy decreases nearly proportional to the flow rate /8/. The mechanical system for this regulation is so complicated, that it can only be realized for large gas compressors.

The discussed methods of regulating the inlet valve employ guided control devices. Mechanism for moving the valve plates can be classified as mechanical, electro-magnetic, hydraulic and pneumatic. Such hydraulic systems are relatively complicated and the disadvantage is their high inertia in the fluid upon periodic motion. Hydraulics can thereby not be used in high speed compressors. A pneumatic regulating system has in fact the simplest design but just as in the hydraulic system can only be realized for rotating speeds up to 600 rpm.

An internal self-acting control regulated by impact pressure is better for high speed refrigerant compressors than the intricate control systems. The intake valve can be held open during the compression stroke by means of either a variable spring force from an electrical coil, until due to the accelerating piston the impact pressure rises at a certain point enough to shut the suction valve off. The increase of the impact pressure ends at the piston's position where its speed is a maximum. As a result the self-acting control is limited, so that the closing of the valve can only take place before this point. By special design measures increasing the affect of the impact pressure, the limiting position can be shifted towards TDC, such that an uniform decrease of the capacity can be achieved up to approximately 40% /4/ (Figure 5b).

2.2.1.2...By-pass Ports in the Cylinder
In order to avoid the disadvantage of a reduced service life of a forced open suction valve, part of the transferred gas is returned to the suction manifold by by-pass valves located on the cylinder. There are two variations for this: a) actuated shut-off devices or b) waste gates, which are partially or fully opened.

This system of capacity control is employed mainly in gas compressors, but has been applied in the past also to refrigerant compressors according to Figure 6. It has relatively small energy losses and is not only economical but also extremely reliable, since there are no continuously moving parts. In view of this it is perfectly suited for high-speed compressors, where the regulating by keeping the suction valves open is not applicable due to technical problems. This method of internal back flow regulating is energetically slightly less favorable than the capacity control method of keeping the suction valve for part of the compression stroke open. The return ports are arranged in the cylinder in such a manner, that the remaining stroke constitutes about 50 to 60% of the total compression stroke after their opening.

2.2.1.3...Regulating the Discharge Valves
Capacity control can be achieved by regulating the discharge valve as well as by suction valve regulation. The flow rate is likewise adjusted by either completely or partially opening the discharge valve.

From an energetic point of view higher pressure losses occur for the fully opened suction valve during the entire intake stroke than for the fully opened discharge valve during the compression stroke. The higher pressure losses take place during idling, since the compressed gas must be expelled through the discharge port.

Partial opening of the discharge valve during part of the suction stroke is energetically more favorable than a fully opened suction valve during part of the compression stroke, since the work of re-expansion can be made useful /10/.

In a similar manner to the suction valve control, the opening of the discharge can either be a forced or self-acting control during part of the intake stroke. The compressed gas flows back into the cylinder during intake stroke. The re-expansion takes place after the discharge valve is closed and is taken full advantage through a delayed opening of the intake valve. Whereas the guided control design types can only be considered for large slow-speed gas compressors, the self-acting control designs are the only conceivable solutions for refrigerant compressors.

The self-acting delayed closing is caused by a spring or electro-magnetic force pressing the valve towards the valve stop until the pressure difference at the valve plate upon the upward motion of the piston overbalances the holding forces (Figure 7a, b). In consequence of increased pressure differences the closing force and thus the impact velocity, a determinant of the life expectancy, are larger for the regulated discharge than for the suction valve.

2.2.1.4... Blocked Suction
By blocking the entire cross section in front of the suction flapper valve the intake of gas is avoided (Figure 12). Expelling the gas into the discharge manifolds results in a compression work of approximately zero. Thus the power demand corresponds to the friction losses of the piston. Therefore a high efficiency of approximately 86% is obtained /5/.

2.2.2...Changing the Cylinder Volume
Here, the capacity of a compressor is regulated by adding an over-re-expansion volume to the cylinder volume. Each compressor cylinder has a so-called re-expansion volume, whose size is conditioned by the design and effects the volumetric efficiency and therefore the transferring quantity. The flow rate of a compressor can be reduced using the fact that after the compression stroke, the remaining gas in the clearance pocket re-expands holding the self-regulating valve flap longer closed.

The over-re-expansion chamber control method allows a stepwise change of the transferring quantity of the compressor or even infinitely variable using a
The simplified equation for the intake volume is expressed as

\[ V_s = V_g - V_0 \left( \frac{\pi}{\text{nex}} - 1 \right) \]  

(1)

when the effect of the temperature is not taken into consideration and the pressure coefficient is assumed to be constant.

Where: 
- \( V_s \) = Intake volume, 
- \( V_g \) = geom. stroke volume 
- \( V_0 \) = over-re-expansion g volume, 
- \( \pi \) = pressure ratio

The expression shows that \( \left( \frac{\pi}{\text{nex}} - 1 \right) \) of the volume of the over-re-expansion volume \( V_0 \) counteracts the stroke volume \( V_g \). That means, the larger the over-re-expansion volume \( V_0 \) is, the smaller the intake volume becomes.

Theoretical indicator diagrams can be seen in Figure 8a and 8b. Figure 8a shows an indicator diagram with the pressure ratio held constant. The dotted line and the solid line show respectively the entire process and the partially loaded process achieved by connecting the additional volume \( V_z \) to the cylinder. As a result the intake volume is reduced by the amount \( V_{ex,z} \).

Furthermore (for an energetic improvement) it is advantageous not to cool the re-expansion chamber, since this would cause the polytropic exponent to become larger than the isentropic. For this reason the cooled re-expansion chamber must have a larger volume and beyond that more power is required in account of irreversible heat transfer.

A good possibility to ensure against cooling is to design the re-expansion chamber in the cylinder head or in the valve plate. The process becomes somewhat more complicated when the pressure ratio is simultaneously changed. In this situation the expanded volume \( V_{ex,o} \) changes into \( V_{ex}',_o \). Therefore, as Figure 8b shows, the volume of the re-expansion chamber must be determined in such a manner, that the change in \( V_{ex,o} \) is taken into consideration for a desired reduction of the intake volume. Different pressure ratios occur frequently in refrigeration compressors due to the various external conditions (temperature of condensation). This means, that the over-re-expansion volume control has corresponding different partial loads.

### 2.2.2.1 Connecting an Over-Re-Expansion Volume of Constant Size

Stepwise capacity control is achieved by connecting one or more re-expansion chambers with constant volume to the cylinder room by means of a check valve. By this the capacity can be regulated from 100% to idling (see Figure 9a). Such a discontinuous control process is energetic favorable.

### 2.2.2.2 Over-Re-Expansion Volume of Variable Size

A stepless control of the flow rate can be achieved, when the re-expansion chamber is constructed out of a cylinder with an adjustable piston and therefore its volume is infinitely variable.

### 2.2.2.3 Over-Re-Expansion Volume of Constant Size, Connected For Part of the Compression Stroke

Even with a constant sized re-expansion chamber, a stepless even capacity control can be reached, if the time interval can be changed, during which the supplementary clearance pocket can be switched on or off within a compression stroke. This type of capacity control is also known as regulating through a re-compression chamber without complete compression.

The theoretic process of the control is shown in Figure 9b. Is the valve, connecting the over-re-expansion volume, opened the compression cycle curve starts to develop rather flat (points 1 to 5), according to the cylinder volume. Upon reaching a pressure that results in closing the connecting valve the compression continues to develop steeper from point 5 to point 6. From 6 to 3 the gas is expelled. During the subsequent suction stroke (point 3 to 7) the connecting valve remains closed. It opens (point 7) as soon as the pressure inside the re-expansion chamber overcomes the cylinder pressure. Subsequently the expansion curve (point 7 to 8) develops flat, due to the enlarged re-expansion. The effecting compression stroke starts (point 8) with an according delay.

With a sufficiently large re-expansion chamber the capacity can be uniformly regulated saving energy for a large range by linearly changing the control pressure. The regulating method is economical for a pressure ratio as small as \( \pi = 2.5 \). Below that the re-expansion chamber becomes too large. The timing of the cutoff valve between the re-expansion chamber and the cylinder can be brought about by using a draw piston. The draw piston is constantly influenced by the rising compression or falling expansion pressure as well as by an adjustable counter-force. The adjustment of the control pressure can take place by changing pre-stress in a spring or by using other automatic techniques.

It is conceivable, that the action of the compressor pistong could open and close the re-expansion chamber, which could provide a simple actuation for a small high-speed compressor. It is possible to combine the methods of internal by-passing through ports and of employing over-re-expansion volumes. By positioning the port in the cylinder wall, the re-expansion chamber can be self-actively opened and closed by the piston.

### 2.2.2.4 Changing the Stroke of the Piston

There are two possibilities to change the piston stroke which are: 1) shifting the position of the piston in relation to the cylinder and 2) shortening the piston stroke. A change in the displacement is achieved in both cases, so that an over-re-expansion control type occurs.

The displacement of the piston bearing requires a large-scale design and is therefore not practised. Reducing the stroke length thus enlarging the re-expansion volume is conceivable in compressors by using an eccentric crank shaft. Turning the eccentric would allow a relatively simple control.

From an energetic point of view this control method is comparable with a regulation by over-re-expansion chambers.
3. RESULTS

This paper dealt with a summary of literature concerned with capacity control of high-speed reciprocating compressors. The possibilities of capacity control were classified and valued for energetic consumption. Deviated from this the following table was created where increased energetic losses were responsible for the order of sequence (Table 2).

<table>
<thead>
<tr>
<th>Type of Capacity Control</th>
<th>Power Requirement for 60% Output (One Stage Compression)</th>
<th>Compression Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. speed regulation</td>
<td>60%</td>
<td>1,0</td>
</tr>
<tr>
<td>2. a) start stop regulation</td>
<td>63%</td>
<td>0,95</td>
</tr>
<tr>
<td>b) using over-re-expansion chambers</td>
<td>63%</td>
<td>0,95</td>
</tr>
<tr>
<td>c) variation of compressor stroke</td>
<td>63%</td>
<td>0,95</td>
</tr>
<tr>
<td>3. a) using over-re-expansion chamber of constant size</td>
<td>65%</td>
<td>0,92</td>
</tr>
<tr>
<td>b) using over-re-expansion chamber of constant size during a part of compression stroke</td>
<td>65%</td>
<td>0,92</td>
</tr>
<tr>
<td>4. discharge valve control</td>
<td>64%</td>
<td>0,91</td>
</tr>
<tr>
<td>5. a) suction valve control</td>
<td>70%</td>
<td>0,86</td>
</tr>
<tr>
<td>b) early suction valve closing</td>
<td>70%</td>
<td>0,86</td>
</tr>
<tr>
<td>c) blocked suction valve closing</td>
<td>70%</td>
<td>0,86</td>
</tr>
<tr>
<td>6. uncoupling the compressor</td>
<td>71%</td>
<td>0,85</td>
</tr>
<tr>
<td>7. a) internal by-passing</td>
<td>73%</td>
<td>0,82</td>
</tr>
<tr>
<td>b) cylinder-shut-off</td>
<td>73%</td>
<td>0,82</td>
</tr>
<tr>
<td>8. control of suction pressure</td>
<td>85%</td>
<td>0,71</td>
</tr>
<tr>
<td>9. compressor by-passing</td>
<td>100%</td>
<td>0,6</td>
</tr>
</tbody>
</table>

The most favorable capacity control in general is a steady control of compressor speed (1). The overall efficiency will be nearly n = 1. This is because the specific compression power will decrease at lower compressor speeds, and the efficiency of the machine will decrease as well. By this a certain compensation of the absolute values of efficiencies takes place. But this type of control cannot be favorable because of high investigations, especially when an electro motor drive was installed. Using motors with commutable poles will result in higher prices and lower efficiencies of the driving device.

In the sense of energetic criteria (start-stop-regulation (2a), use of over-re-expansion chambers (2b) and variation of compressor stroke (2c)) will be the next best type of capacity control. The start-stop-regulation will not be practicable because of bad accommodation. Furthermore the compressors must be run within non-regulated intervals and therefore higher friction and wear losses must be predicted. The other two types of capacity control (2b, 2c) can only be realized when special constructive expense is given.

Varying the re-expansion volume requires an extra piston which must be adjustable at different positions. A variation of compressor stroke can only be achieved by use of a special eccenter.

Using over-re-expansion chambers with constant volume (3a) or constant volume during a part of compression stroke (3b) therefore must be the most favorable type of control in the sense of energetic and economic criteria. A compressor efficiency of n = 0,92% can be stated for this type of capacity control. Influencing the discharge valve lift should be another suitable type of control. The internal energy of compressed gas can be used during re-expansion, but throttling losses in the valve area must be taken into account. The control of suction valve lift (5a, 5b) is less favorable because of higher energetic losses. As stated before throttling losses in the valve area will be given but no re-expansion work can be expected. A compressor efficiency of 86% should be predicted in this case. Nearly the same rates of efficiencies are to be expected when blocked suction valve capacity control is used (5c). Uncoupling the compressor will result in 85% efficiency.

The next best type of capacity control is the internal by-passing of gas (7a). This could be caused by special flow areas in the cylinder wall but higher throttling losses cannot be avoided (n = 82%). It would be better to combine such flow areas with a over-re-expansion chamber. In this case re-expansion work of compressed gas could be used furtheron and lower throttling losses were to be expected. This type of capacity control is well known from compressors of large size where over-re-expansion chambers of constant size are switched to the cylinder volume by special valves. Compressor efficiencies of 92% were reached in such constallations. Other methods as used normally for air compressors (control of suction pressure - n = 70% - and external by-passing - n = 60% -) are less favorable in the sense of energetic evaluation and therefore these methods are not discussed more detailed.

4. SUMMARY

Different types of capacity control were compared in this paper and an energetic evaluation of these methods was given. Only a few of these control methods can be applied to high speed compressors. No moving control systems are wanted besides the moving piston. Furthermore this study deals with compressors of small and medium size and therefore those regulation systems are not applicable which require complicated mechanical systems to influence the valve movement. Beyond that compressors with less than four cylinders were viewed. Therefore all cylinders must be regulated simultaneous and all those types of capacity control are not applicable which regulate only single cylinders as for instance this would be given in the case of suction valve blocking.

Taking into account these criteria the over-re-expansion method and the late discharge valve closing method /24/ will be the most favorable capacity control method for small refrigeration compressors with low number of cylinders in the sense of energetic criteria. Using over-re-expansion chambers of constant size will result in 92% compressor efficiency when 60% compressor output is given. Disadvantage of this capacity control method is a non-constant part load at different pressure ratios (Figure 8) and therefore a different characteristic must be expected for different working points.

Figure 11 shows different pV diagrams for several dead-volumes in the cylinder /23/. These results
were obtained from theoretical calculations of the working process. It can be stated that no pressure oscillations during suction and discharge period will occur. This type of capacity control which requires low energetic losses should be favoured at high compressor speeds.

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Over-Reexpansion as a Means of Compressor Capacity Modulation
Figure 1. Suction throttling, a) in a p,v-diagram, b) in a log p,h-diagram including schematic.

Figure 2. a) Wiring scheme for magnetic controlled bypass valve, b) log p,h-diagram for throttled equalization including injection.

Figure 3. a) Schematic of an external control device for equalization, b) schematic of an internal control device for equalization (Carrier system).

Figure 4. a) p,v-diagram for fully opened suction valves, b) design of a control device for keeping the suction valve open.

Figure 5. a) p,v-diagram for regulating the discharge valves during part of the compression stroke, b) design example (Borsig).

Figure 6. Compressor with bypass ports in the cylinder wall (York).
Figure 7. a) Electro-magnetic valve with external coil: 1 electro-magnetic coil, 2 permanent magnet, 3 diamagnetic coil core, 4 diamagnetic valve core, 5 ring plate  
b) p,v-diagram for fully opened discharge valve

Figure 8. Theoretical indicator-diagram for an over-re-expansion control system  
a) with constant pressure ratio  
b) with decreased pressure ratio

Figure 9. a) Indicator-diagram for a constant sized re-expansion chamber, b) theoretical indicator-diagram for connecting an over-re-expansion volume during part of the compression stroke

Figure 10. Self-activated over-re-expansion control with respect to the impact pressure

Figure 11. p,v-diagram under variation of the clearance volume of the cylinder  
Legend:  
\( \varepsilon_0 \) clearance volume ratio  
\( V_0 \) clearance volume  
\( V_{\text{min}} \) geometrical swept volume  
\( V_{\text{max}} \) maximum cylinder volume  
\( P_S \) suction pressure  
\( P_D \) discharge pressure

Figure 12. Blocked suction system /5/