

## Methods for large reciprocating compressor capacity control: A review based on pulse signal concept

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Large reciprocating compressors are important equipment used in a wide range of process industries. Most of these compressors have huge power requirements and their capacity often needs to be regulated. Advanced technologies in compressor capacity control are effective approaches for saving large amounts of energy in process industries. This paper reviews the basic theories and the state of the art of the capacity control technologies. In particular, a compressor working procedure is first presented using an analogy to the pulse signal concept. Compressor capacity control methods are classified into pulse frequency modulation and pulse amplitude modulation from a perspective of pulse signal processing. The mechanisms and feasibility of some important methods, including dead volume variable control, partial-stroke and full-stroke suction valve opening, are reviewed. Based on the pulse signal concept, a duty cycle regulation method for capacity control is introduced, and the performance and implementation of the new method are compared with those of the existing suction valve opening methods. The duty cycle regulation method has integrated advantages over the other methods in terms of regulating precision, pressure stability, energy saving and reliability. All the suction valve opening methods can cause gas reflux, resulting in the so-called breathing effect. The breathing effect has negative effects on regulation performance and compressor security, which needs further investigation in the future.

**reciprocating compressor, capacity control, suction valve opening, duty cycle regulation**

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Large reciprocating compressors, characterized by high energy efficiency throughout a wide range of output pressure and variable conditions, are important equipment in various process industries. For example, about 10–15 compressors are needed in a  $1 \times 10^6$  t/a petroleum refining unit, and about 20–30 compressors in a  $10 \times 10^6$  t/a ethylene processing unit. Most of these compressors have input power of more than 2000 kW; some even more than 3000 kW. Thus, higher power efficiency in compressor operation is essential for energy saving and emission reduction in the economy development. A process compressor should supply a rated capacity under the specified conditions, but its capacity

must overcome the peak capacity, which may be occasionally encountered in the process; therefore, only 60%–85% of the designed capacity is needed during most of its running time. On the other hand, actual capacity is subjected to changes in raw material properties, production techniques and product components or structure. Thus, proper methods of capacity control are necessary. Because the power of a compressor is in positive relation with its capacity, capacity control with high efficiency and accuracy is an important technology and an effective approach to significant energy savings in process industries. The actuator for capacity control of a process compressor needs to be highly reliable, ensuring continuous gas supply with stable output pressure. In short, the advanced capacity control technologies should integrate the following characteristics: high reliability, high

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stability, high accuracy and high energy efficiency.

### 1 Basic theories of compressor capacity control

#### 1.1 Thermodynamic principle of capacity control

Reciprocating compressors encloses gas in a cylinder and then reduces the cylinder volume with a movable piston to raise the gas pressure. A working cycle of the compressor commonly consists of four successive thermodynamic processes: suction (d-f-a), compression (a-b), discharge (b-c) and expansion (c-d), as shown by the dashed lines in Figure 1. The occurrence of the expansion process is due to the residual high-pressure gas in the dead volume  $V_0$  at the end of the discharge process (i.e. Point c).

Since the suction and discharge processes are discontinuous, the term “capacity” refers to an average value over a period. It is defined as the gas flux entering the first-stage cylinders during a unit time under the specified inlet and outlet conditions, usually expressed in mass flow units, or in volume flow units under the standard conditions (specified as 101325 Pa and 0°C). If the gas can be treated as an ideal gas, the capacity of a compressor is calculated [1] by

$$Q_m = \frac{V_{s1} p_{s1}}{R_g T_{s1}} \cdot \frac{n}{60} = \lambda_{v1} \lambda_{p1} \lambda_{T1} \lambda_{l1} \frac{n V_{h1} p_{s1}}{60 R_g T_{s1}}, \quad (1)$$

where the subscript 1 denotes the first-stage,  $n$  is the rotating speed, and  $V_h, p_s, T_s,$  and  $R_g$  are the total swept volume, suction pressure, suction temperature and gas constant. According to the documented results [1], the pressure coefficient,  $\lambda_{p1}$ , varies over a range of 0.95–0.98, the temperature coefficient,  $\lambda_{T1}$ , over 0.93–0.97, and the pressure coefficient,  $\lambda_{l1}$ , over 0.95–0.97. The volume coefficient,  $\lambda_{v1}$ , is determined by

$$\lambda_{v1} = 1 - \alpha_1 (\varepsilon_1^{\frac{1}{m}} - 1), \quad (2)$$

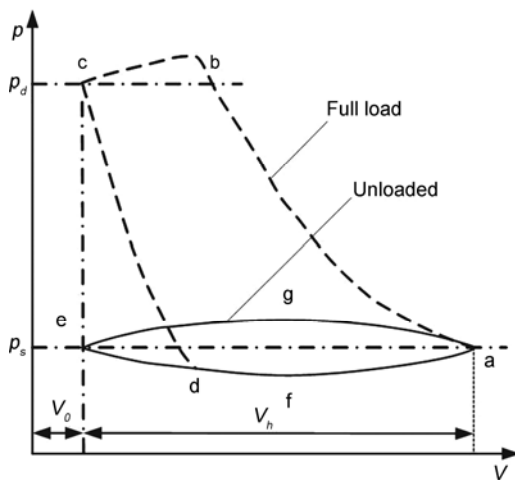


Figure 1 Actual working process.

where  $\alpha$  is the dead volume coefficient,  $\alpha=V_0/V_h$ ;  $\varepsilon$  is the stage compression ratio, and  $m$  is the expansion indicator.

As far as the thermodynamics are concerned, the compressor capacity can be regulated by adjusting the value of the parameters in eq. (1). Besides practical and economical aspects, the power of the compressor under regulation is also essential for the selection of parameters. The power of a compressor is the sum of indicated power for all stages plus the mechanical loss. The indicated power  $P_i$  of stage  $j$  is calculated by

$$P_{ij} = \frac{n}{60} (1 - \delta_{sj}) p_{sj} \lambda_{vj} V_{hj} \frac{m_j}{m_j - 1} \{ \varepsilon_j (1 + \delta_{sj} + \delta_{dj}) \}^{\frac{m_j - 1}{m_j}}, \quad (3)$$

where  $\delta_s$  and  $\delta_d$  are the relative losses of suction and discharge pressures. The mechanical loss can be involved in the power calculations using a mechanical efficiency, so it is not discussed individually.

The specific power of a compressor is the power needed for compressing a unit mass flow of gas. This factor is usually used to evaluate the performance of air compressors. Since capacity control will usually change the power consumption, advanced control methods for process compressors are expected that the actual power will reduce proportionally with the flow rate turndown, or in other words, the specific power keeps constant. The specific power is introduced in this paper as an economic factor to assess the different control methods.

#### 1.2 Analogy analysis between compressor capacity control and the pulse signal concept

Because the suction and discharge processes are discontinuous, the flow throughput is actually a periodic pulses output, thus the capacity control is analogous to pulse signal processing.

Rectangular pulse signals are often encountered in electric and electronic technologies. For a rectangular pulse with amplitude  $A$  and sustaining time  $\tau$ , we define its energy  $w$  as  $w=A\tau$ . During a period of time  $T$ , if an inertial object receives a sequence of  $N$  pulses with an interval  $s$  between two adjacent pulses, it is the equivalent of undertaking a continuous constant signal of amplitude  $\bar{A}$ , which energy  $W$  equals to the summing energy of the pulses sequence; thus,

$$W = \int_0^T \bar{A} dt = \bar{A} T \simeq \sum_{j=1}^N w_j = \sum_{j=1}^N A_j \tau_j. \quad (4)$$

Both  $A$  and  $\bar{A}$  have the meaning of “power”, and the time  $T$  is confined to

$$T = \sum_{j=1}^N (\tau_j + s_j). \quad (5)$$

Eqs. (4) and (5) provide different ways to adjust  $\bar{A}$ :

one simple method is the pulse amplitude modulation (PAM), in which the amplitude  $A$  of all the pulses is uniformly stretched. Another method is the pulse width modulation (PMW), in which  $\tau$  or  $s$  is variable while their sum remains constant. In these two methods, the energy of each pulse is adjustable while the number of pulses in a unit time is invariant. A third approach is the pulse frequency modulation (PFM), in which the energy of every pulse is invariant, but the number of pulses in a period is adjustable. All these methods are used extensively in electronic technologies.

As mentioned above, the reciprocating compressor works periodically with one revolution as a working cycle. Suppose that the thermodynamic conditions at the inlet and outlet are stable, the last stage of the compressor will discharge a gas micelle with a certain mass in every working cycle. The micelle can be regarded as a discharge mass pulse, which is logically analogous to the electric pulse: the sustained time of the discharge stroke corresponds to the pulse width; the discharge mass of a working cycle, to the pulse amplitude; and the interval between two discharge strokes, to the pulse interval.

Based on the above mass pulse concept, the output of a compressor is a sequence of mass pulses. Such a sequence is imposed on a surge tank and changes the pressure in the tank. This pressure change continues to influence the output flow rate from the tank into the downstream pipelines and equipment. Within a certain range of pressure, the relationship between the mass flow rates into and out of the surge tank,  $Q_{mo}$  and  $Q_{mi}$ , can be described as a transfer function:

$$G(s) = \frac{Q_{mo}(s)}{Q_{mi}(s)} = \frac{1}{T_c s + 1} e^{-\tau s}, \quad (6)$$

where  $T_c$  is the temporal constant of the tank transient process, which is a product of the tank volume and the pipeline resistance of flow; and  $\tau$  is the time lag, which can be neglected when the tank is close enough to the compressor. According to eq. (6), the surge tank is a first-order inertial object, acting as a low-pass filter to the compressor: the surge tank receives the mass pulse sequence from the compressor, filters the high-frequency fluctuation and then outputs a constant signal—the average mass flow.

Based on the pulse signal concept, eq. (1) can be rewritten as

$$Q_m = \frac{1}{60 R_g T_{s1}} \cdot n \cdot (\lambda_{v1} \lambda_{p1} \lambda_{T1} \lambda_{t1} V_{h1} p_{s1}) = k \cdot n \cdot dm, \quad (7)$$

where  $k$  is an invariant coefficient, and  $n$  can be considered as the frequency of the output gas pulses. We combine the units of  $k$  into  $dm$  and regard  $k$  as a dimensionless number, then  $dm$  will have the unit of mass, representing the mass of gas entering the cylinders in every working cycle.

Eq. (7) indicates two approaches for compressor capacity control: either changing  $dm$ , the amplitude of the gas pulses;

or changing  $n$ , the frequency of the gas pulses sequence. According to the pulse signal concept, they fall into the PAM or PFM categories, respectively.

## 2 State of the art of compressor capacity control technologies

A variety of methods is available for the capacity control of compressors used in various fields [2,3]. Traditionally, based on the positions of the regulating devices, the methods are classified into four groups:

- (1) Speed control, such as on/off control and gradient speed-variable control;
- (2) Pipeline regulation, such as inlet throttle, inlet shut-off, outlet free-bypass and outlet throttle-bypass;
- (3) Dead volume control, which requires modifying the cylinder structure and connecting the cylinder to an extra dead volume pocket; and
- (4) Suction valve opening, in which the suction valves keep open and the gas flows back to the suction chamber.

These methods can also be re-classified into PAMs and PFMs, as listed in Table 1.

The outlet throttle-bypass method is excluded in Table 1, because it is not strictly a capacity control method. With this method, the excessively compressed gas is bypassed from the compressor outlet, throttled into low-pressure gas, and finally induced back into the inlet. From the perspective of the pulse signal concept, the compressor serves as a constant flow source with two resistant branches connected parallelly at the compressor outlet. How the flow is distributed has little effect on the compressor power; thus, despite the need for partial flow, the compressor power still remains at full-level, causing a great deal of wasted energy. However, because of its simplicity in implementation and continuity of gas supply, it is still used today.

Each of the methods listed in Table 1 has merits for its applications on small or mini compressors in pneumatic air, refrigeration and other fields. However, their applicability on large multi-stage process compressors still needs investigation. To ensure both practicability and reliability, the API 618 standard [4] accepts the outlet throttle-bypass, the variable dead volume control, the suction valve opening and the combinations of these methods as the capacity control of multi-stage process compressors.

**Table 1** Classification of the capacity control

PAMs	PFMs
Inlet throttle	Outlet free-bypass
	Inlet shut-off
Variable dead volume control	On/off control
	Speed-variable control
Partial-stroke suction valve opening	Full-stroke suction valve opening

### 3 Analysis of PAM methods

The PAM methods are to change the value of  $dm$  in eq. (7). Increasing the suction temperature can reduce the temperature coefficient  $\lambda_T$ , but this is unsuitable for capacity control, for it will cause a rise in power and discharge temperature at the same time. The inlet throttle method can reduce the pressure coefficient  $\lambda_p$ , but this will change the pressure ratio between different stages, increase the piston rod force and the discharge temperature, and eventually cause damage to a large compressor. Therefore, the inlet throttle method can only be applied to small compressors. The practical methods are the variable dead volume control and the partial-stroke suction valve opening. The former reduces the volume coefficient  $\lambda_v$  by additional dead volume, while the latter reduces the leakage coefficient  $\lambda_l$  via gas reflux.

In the case of variable dead volume control, when the dead volume enlarges from  $V_0$  to  $V'_0$ , the end of the expansion process will shift from  $d$  to  $d'$ , shown in Figure 2. According to eq. (2), if the other parameters remain invariant, the capacity will be reduced linearly with the rise of the dead volume coefficient  $\alpha$ . Generally,  $\alpha < 0.3$ , there is little space for regulation. However, when an additional dead volume pocket is installed,  $\alpha > 1$  is possible. For an instance of  $m=1.3$  and  $\varepsilon=2$ , when  $\alpha > 1.42$ , the capacity will theoretically fall to zero.

A comparison of eq. (1) with eq. (3) indicates that  $\alpha$  has the same influence on the capacity and the power; thus, the specific power remains constant. The variable dead volume control has effective energy savings when compared with the outlet throttle-bypass method. The Southwest Institute of America (San Antonio, Texas) carried out an investigation into the variable dead volume control and regarded this method as one of the most useful technologies [5]. However, this method has a low resolution to the regulation, and it may change the force of the piston rod when the capacity is massively reduced.

In the other method, partial-stroke suction valve opening,

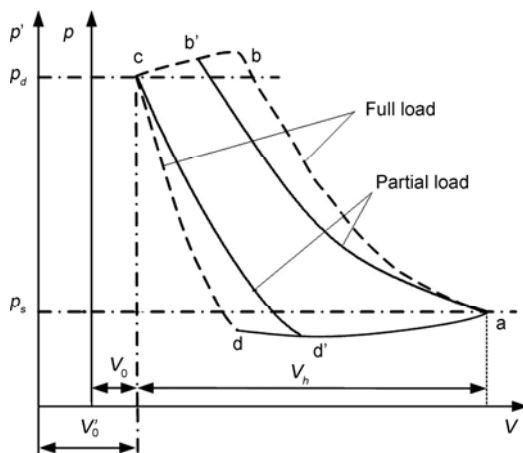


Figure 2 Variable dead volume control.

the suction valve keeps open via a servo apparatus, from the beginning of compression and for a specified time. Part of the gas flows back to the suction chamber during this period. When the specified time is due, the suction valve is released and the residual gas in the cylinder is enclosed, compressed and discharged, as shown in Figure 3. In this method, only the residual gas is compressed and the specific power remains constant. This method can steplessly regulate the capacity and has marked energy savings when compared with the outlet throttle-bypass method. Based on this method, the HydroCOM System (from Hoerbiger, Austria) [6, 7], the ISC System (from Dress-Land, USA) [8, 9] and the DHU System (Zhejiang University, Hangzhou, and Zhejiang Petrochemical Co., Zhenjiang, both in Zhejiang Province, China) [10–18] have been developed and applied in industry. However, since the actions of valve opening and release must be accomplished in every back stroke, the servo apparatus must achieve extremely high dynamic performance and reliability. Take a compressor of 600 r/min as an example. The servo apparatus should have a response time of less than 10 ms, and it should be competent for 288 million actions in 8000 hours in one year. With the large compressors developing toward higher speeds [5], the general acceptance of the method will be a more critical challenge.

### 4 Analysis of the PFM methods

PFM methods control the capacity by changing the value of  $n$  in eq. (7). Among the methods listed in Table 1, only the speed-variable control can achieve continuous regulation. The compressor speed is varied usually by an inverter-fed motor. Although the bulk-capacity inverter is weak for speed variant range and cost-efficiency so far, it is one of the active research topics in science and technology. Other methods listed in Table 1 are for discontinuous regulation and may, to some extent, cause fluctuations in the supply pressure. The on/off control can not be applied to the capac-

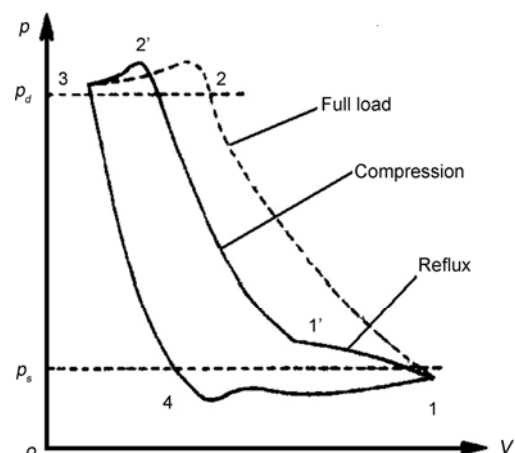


Figure 3 Partial stroke suction valve opening.

ity control of large compressors; the outlet free-bypass method can be used for temporary unloading, but not for long-term regulation; the inlet shut-off method can greatly change the pressure ratio, which is harmful to the compressor. Only the full-stroke suction valve opening method is an acceptable PFM method for large compressors.

#### 4.1 Full-stroke suction valve opening

In this method, the suction valve keeps open throughout the whole stroke, and all the gas in the cylinder flows back to the suction chamber. Though the compressor output will be cut off for a short time, the gas supply continues out of the surge tank. The indicating diagram consists of a suction arc e-f-a and a reflux arc a-g-e, as shown by solid lines in Figure 1. Little energy is consumed during this time. When the suction valve is released depends on the pressure in the surge tank, thus this method is known as pressure-feedback on/off control, and the servo apparatus is called "valve unloader" [19,20]. From the perspective of the pulse signal concept, the mass pulse sequence is applied or withdrawn as a whole in this method. It may cause pressure fluctuations in the pipelines and can only achieve step regulation. In China, Wuhan University of Technology (Wuhan, Hubei Province) [21,22] and Tianhua Institute of Chemical Machinery and Automation (Lanzhou, Gansu Province) [23] are dedicated to the improvement of the valve unloader in different ways.

#### 4.2 Duty cycle regulation method

Duty cycle regulation (DCR) is a new method introduced by us [24] based on the full-stroke suction valve opening method. Instead of pressure-feedback on/off control, duty cycle regulation on the pulse sequence is used as the control strategy in the DCR method. The duty cycle of a pulse sequence,  $R$ , is defined as the ratio of the sum of the sustaining times of all individual pulses to the duration time  $T$  of the pulse sequence, expressed as follows:

$$R = \frac{1}{T} \sum_{i=1}^N \tau_i. \quad (8)$$

For a sequence of uniform-width pulses, eq. (8) can be simplified as  $R = N\tau/T$ , and the continuous constant of the sequence  $\bar{A} = RA$ . That is,  $\bar{A}$  can be altered simply by varying the duty cycle  $R$ . Thus this method is named as DCR. Since the gas output of a compressor can be regarded as a sequence of mass pulses, the DCR method can be applied to the capacity control of the compressor. The control procedure of the DCR method can be as following:

- 1) Take  $M$  working cycles as one control unit;
- 2) Use a servo apparatus to open or release the suction valves for whole strokes, and the compressor is thus loaded or unloaded alternately, and the working cycles are accordingly divided into loaded cycles and unloaded cycles;

3) Define the compressor duty cycle  $R_c$  as the proportion of loaded working cycles in the whole control unit; if the number of loaded working cycles is  $N$ , then  $R_c = N/M$ ;

4) If the full capacity is  $Q$ , the output capacity under DCR is  $\bar{Q} = RQ$ .

The above is the basic theory of the DCR method for compressor capacity control. In this method, the parameters of each pulse in eq. (5) are invariant while the number  $N$  of pulses in time  $T$  is changed by adjusting the time interval  $s$  between pulses, so the DCR method falls into the PFM category. An example of the DCR procedure is shown in Figure 4. To make 70% capacity, take 10 working cycles as one control unit; so  $M=10$ . The suction valve is pressed open at the third, sixth and ninth working cycles by a servo apparatus and is released at the other cycles. The output mass is then zero for the 3 unloaded cycles, while the output remains full scale for the other 7 cycles; so  $N=7$ , and  $R_c=70\%$ . Therefore, the average capacity for the 10 cycles is 70% (see the left chart in Figure 4). By contrast, 100% capacity is presented in the right chart of Figure 4. The contrast between the two charts in Figure 4 also indicates that the specific power in the DCR method is a constant.

In the DCR method, there are 3 adjustable parameters: the duty cycle  $R_c$ , the control unit length  $M$ , and the mode of DCR. The parameter  $R_c$  corresponds to the percentage of the actual output over the full capacity; while the latter two parameters affect the performances of the capacity control.

The control unit length  $M$  should be in a proper range. Firstly, if  $M$  is too long, the compressor may be obliged to run at full load for too long a time, and the pressure in the surge tank will exceed the upper limit; or on the contrary, the compressor has to run at zero load for too long a period, causing the pressure below the lower limit. With such a DCR operation, although the compressor seems to be under an active control, in fact it results in the same consequences as pressure-feedback on/off control. On the other hand, if  $M$  is too short, the precision rate of DCR is limited. If  $M=10$ , there are only 10 choices for  $R_c$ , and the DCR will degrade to step control with an accuracy of 10%. In brief, the control unit length  $M$  should be long enough to ensure the regulating accuracy; at the same time, it should also be short enough to avoid discharge fluctuation.

The DCR mode refers to the arrangement of the positions of the unloaded pulses in the sequence. Different arrange-

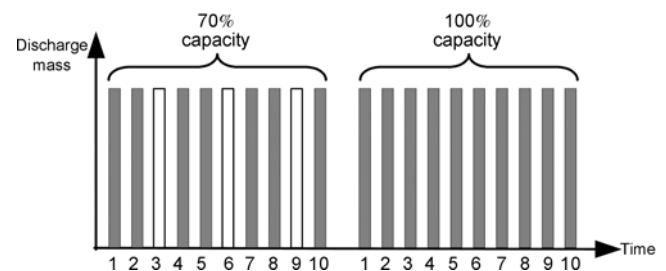


Figure 4 Regulation procedure of the DCR method.

ments have different restraint effect on discharge pressure fluctuation. The unloaded cycles with a uniform interval between each other may be a good arrangement for most cases. Because of the breathing effect (refer to Section 5.3), the DCR mode also affects the actual capacity.

## 5 Comparison of DCR with other suction valve opening methods

### 5.1 Regulation performances

The DCR method, valve unloader and HydroCOM system are all suction valve opening methods. The performances of the three methods are compared in this section based on the concept of the pulse signal processing.

The regulating procedure of the HydroCOM system is shown in Figure 5. To produce 70% capacity, the HydroCOM system has to complete an action in each working cycle, and the compressor outputs a gas pulse in each cycle. However, the amplitude of each pulse is reduced to 70%, thus the total capacity is 70% (see the left chart in Figure 5). By contrast, 100% capacity is presented in the right chart in Figure 5. A comparison of Figure 4 with Figure 5 indicates the DCR method is similar to the HydroCOM system in precision and energy-saving benefits. In the DCR method, however, much fewer actions are needed for the servo apparatus, and the control procedure is easier to be carried out. Thus, the DCR method should have a higher reliability and longer expected life span than the HydroCOM system.

On the other hand, the DCR method is the same as the valve unloader in that the suction valves open for whole strokes during the unloaded period, but it is different from the valve unloader in the timing of the alternation between the loaded and unloaded status. The status alternations in the valve unloader occur when the pressure in the surge tank reaches the upper or lower limit, at  $t_1$  or  $t_2$ , as shown by the dashed arrows in Figure 6. Thus, the output pressure fluctuation is necessary for the valve unloader to work. In comparison, the status alternations in the DCR method occur as a pre-scheduled time series. As shown by the dashed arrows in Figure 7, the status alternation can restrain the output pressure fluctuations before they grow noticeably strong. The post-cooler and surge tank will further damp the low-amplitude pulses, so the output pressure is more stable

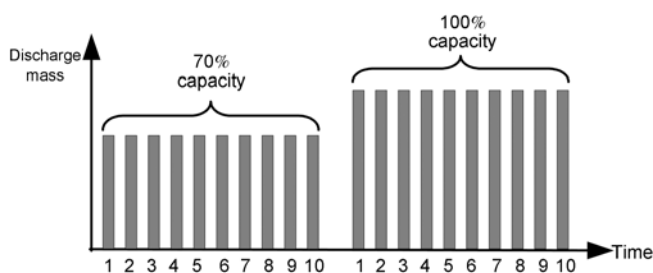


Figure 5 Regulation procedure of the HydroCOM system.

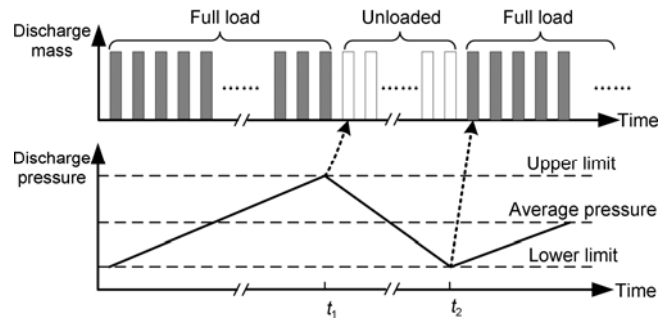


Figure 6 Status alternations in the valve unloader.

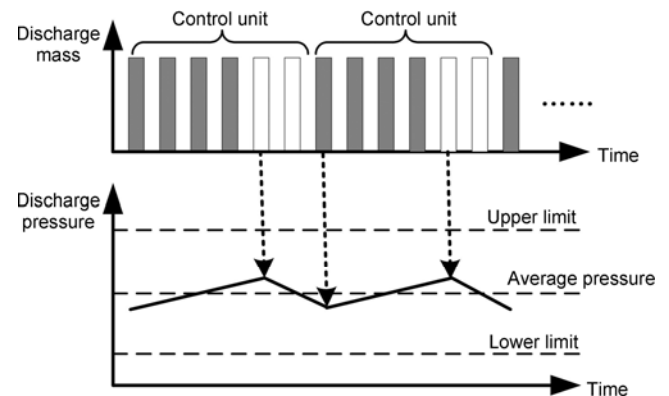


Figure 7 Status alternations in the DCR method.

in the DCR method, and its regulation precision is higher than that of the valve unloader. What's more, the specific power keeps constant in the DCR method theoretically.

From the above analysis, we can reach the conclusion that the DCR method has the integrated advantage of fewer actions, higher reliability, longer expected life span, and well-controlled fluctuation of the output pressure.

### 5.2 Implementation of the regulations

All the three methods handle the mass pulse of the compressor, using a servo apparatus including a pressing prong, which is installed into the suction chamber to press or release the valve. The pressing prongs in the three methods are driven by different power sources: the valve unloader uses pneumatic air, and the HydroCOM system uses an electric-hydraulic servo driver. The DCR method needs no particular design of pressing prong or power source, for its innovation is in the control logic and procedure, that is, the timing for pressing the prong and the time span for each action of pressing. A general hydraulic servo driver could also work in the DCR method. Because fast dynamic response is no longer necessary for the DCR method, even the pneumatic devices of the valve unloader can be used for the cylinders of low-pressure stages. The capacity control can be realized as long as this servo apparatus follows the control logic and procedure of DCR mode.

Another issue that needs attention is the impact on the compressor, which occurs at the alternations between full-load and zero-load status. The impact intensity is much weaker in the DCR method than that in the valve unloader method, and it is slightly stronger than that in the HydroCOM system. Because of their working principles, the latter two methods have no means of reducing the impact, while the DCR method may weaken it through proper configuration of the DCR mode in the different stages and the different cylinders in the same stage. To take full advantage of the DCR method, detailed research on the operational characteristics of the multi-stage compressor under variable working conditions needs to be carried out to determine the proper DCR mode and control strategy.

All the above analyses and conclusions need verification of experiments on a real machine.

### 5.3 Influence of the breathing effect

Whether we consider the DCR method, the valve unloader method or the HydroCOM system, all the regulation results in the gas suction and reflux repeated in the complex flow channel of the cylinders, valves and suction chamber. This phenomenon is the so-called breathing effect. The nature of the breathing effect is an engineering thermodynamic issue of unsteady flow combined with complex heat transfer. Because of the breathing effect, the suction temperature rises firstly, then the discharge temperature and the power of the compressor increases, and finally the lubrication inside the cylinder gets worse. These consequences will become even more serious with greatly decreased capacity. On the other hand, the amplitude of each pulse will drop from the expected value and the obtained capacity will be less than required. Li [25] calculated the breathing effects of these methods on a 3L-10/8 air compressor using the lumped parameter method and the qualitative analysis indicates that the breathing effect influences are:

Valve unloader >> HydroCOM > DCR method.

Although the breathing effect cannot be ignored from the safety and capacity control perspectives, little research has been done on the occurrence mechanisms or the influencing factors underlying the breathing effect. Experimental study and further theoretical analysis are necessary for successful implementation of the DCR method.

## 6 Conclusions

Large reciprocating compressors are important equipment used in a wide range of process industries. Most of these compressors have huge power input while their capacity often needs to be regulated in practice. Advanced compressor capacity control technologies are effective approaches for saving energy for process industries. In this paper, the basic theories and state of the art of compressor capacity

control technologies are reviewed. In particular, an analogy between the compressor working procedure and the pulse signal concept is introduced to help the analysis of the principles of these control technologies. The capacity control methods are classified into pulse frequency modulation and pulse amplitude modulation from a perspective of pulse signal processing. Some important methods, such as the dead volume variable control, the partial-stroke suction valve opening, the whole-stroke suction valve opening, and the duty cycle regulation method, are investigated to reveal their mechanisms and feasibility for large compressors.

Based on the pulse signal concept, a comparative study on the three suction valve opening methods, in terms of system reliability, gas supply stability, capacity control precision and implementation, is carried out in this paper. The duty cycle regulation method has integrated advantages over the others, with fewer actions of the servo apparatus, higher reliability, longer expected life span and well controlled fluctuation of the output pressure.

All the methods of suction valve opening can cause gas suction and reflux, resulting in the so-called breathing effect. The breathing effect in the DCR method is weaker than in the other two. The breathing effect has a negative impact on the performance and security of capacity control and needs further investigation.

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