Design of Capacity Regulation System for Reciprocating Compressor Based on Programmable Logic Controller

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Abstract: - The reciprocating compressor used in the industry consumes copious amounts of energy. Unnecessary energy consumption can be reduced by regulating the compressor capacity and by better satisfying its energy efficiency and environmental and production requirements. In this paper, a programmable logic controller (PLC)-based capacity-regulating system is successfully designed and put forward for the first time after an analysis of the time control technology of pressing-off the inlet valve plate during partial stroke. The actuator, which works on the principle of electromagnetism, is designed to press off the valve plate to control the opening and closing of the inlet valve. The control system, mainly composed of the PLC, controls the on-off time of the coil to implement the application and revocation of the open force acting on the valve plate. The control system also measures actual crankshaft speed to optimize the regulation result. The design process and experimental application of the system shows that the technology can realize continuously stepless capacity regulation for the compressor within the 0%–100% range. The entire system is cheap to manufacture, consumes low power during operation, and has high economic efficiency. The regulating system is safe, reliable, easy to operate, and can be applied in most working conditions.

Key-Words: - Programmable Logic Controller; Reciprocating compressor; Capacity regulation; Electromagnet; Actuator; Inlet valve;

1 Introduction

The reciprocating compressor widely applied in the industry is characterized by a wide range of discharge pressure, high efficiency, stability, and adaptability [1]. In normal working conditions, displacement of the compressor does not change due to its volume structure. However, in practical production, air consumption often changes with the requirement of process flow or because of the change of equipment properties. The reciprocating compressor is an example of equipment that consumes large amounts of energy. Consequently, appropriate methods should be developed to regulate the capacity of the compressor within a certain scope, meet the requirements of production process, avoid danger, and satisfy energy-saving and environmental requirements.

At present, various technologies [2, 3, 4, 5] have been adopted to regulate the capacity of reciprocating compressor. These technologies have their own advantages and shortcomings, and are applicable under different conditions. These methods fall into four categories, according to the positions upon which they act:

(1) Acting on the driving mechanism. The methods falling into this category include periodical shutdowns and speed regulation. These methods regulate the capacity of the compressor by shutting it down or adjusting its speed. Periodical shutdowns could realize zero energy loss and features low investment and easy design and operation. Nonetheless, periodical shutdowns can only realize intermittent control (single-machine) or step control (multi-machine). Relay contacts and other parts are subject to damage, and the reliability of the method drops accordingly. Moreover, periodical shutdowns are confined to low-power compressors owing to the restriction of motor capacity. Speed regulation is applicable to internal-combustion engine-driven or steam engine-driven compressors. A compressor provided with this regulating technology features low power consumption, ease of design, and high economical efficiency. Because of the restriction of the prime motor, such technology could fulfill continuous capacity regulation within 60%–100%.

(2) Acting on the pipeline. Control of air intake and control of connection between air intake and discharge fall into this category. Air intake control includes air intake stopping and intercepting air intake. The working principle is to intercept or partially close the air inlet to control the air volume entering into the compressor. Both regulating devices feature simple structure and low investment, but high power consumption. These devices are able to realize intermittent and continuous control. The bypass regulation widely applied in the petrochemical industry is the most typical example of the control of the connection between air intake and discharge. This method first compresses the air, and then returns it to the intake line to reduce the discharge. Power consumption is extremely high; however, it is capable of continuous regulation, and the regulating system is safe and reliable. These types of devices are only appropriate for compressor unloading and short-term or intermittent control.

(3) Acting on the valve. The methods falling into this category include completely pressing off the inlet valve and pressing off the inlet valve in partial stroke. The methods are based on principle of "returning the air into the intake line to conserve power" (i.e., discharging part of or all of the air in the cylinder to the intake line by pressing off the inlet valve before compressing, and then compressing the volume of air needed). The regulating system that involves completely pressing off the inlet valve is easy to design and operate. This system is appropriate for intermittent control and step control, and is applicable during unloading when the compressor starts. The regulating system that involves pressing off the inlet valve in partial stroke could be accomplished through pressing-off force and time control. The pressing-off force control system is relatively easy to design and operate, and is applicable in most working conditions. However, it requires a large investment and is only able to realize 55%–100% continuous regulation. The pressing-off time control is the most prevalent air regulating technology at present [6, 7, 8, 9]. Thus far, some developed products [10] have been launched into the market. Such a technology is able to realize 0%–100% continuous capacity regulation and is applicable in most working conditions, and its operation is not related to intake temperature, pressure, molecular weight of gas and other process variables. However, the R&D and popularization of the technologies are restricted because of their intricate structures and the large investments required. Moreover, collision between air valve and unloader must be considered to guarantee the reliability of the regulating system.

(4) Acting on the cylinder. The main method falling into this category is clearance volume regulation. This method reduces air displacement by adjusting the clearance volume of the cylinder to make the compressed air in the clearance swell and increase in volume during air intake process and to reduce the volume of the intake air accordingly. This method is reliable; however, it is difficult to design and operate, and requires a large investment.

Additionally, it is only appropriate for step control and applicable only in large compressors.

In this paper, a programmable logic controller (PLC)-based capacity-regulating system is designed for the first time. The entire system has a simple structure and is reliable, inexpensive to manufacture, and easy to operate.

2 PLC-based capacity-regulating system

The regulating system is mainly composed of an actuator and a control system. The whole system works based on the principle "returning the air into the intake line to conserve power" (Figure 1). If the inlet valve is forced by the actuator to remain open when the suction process reaches Point 1, then the compression process cannot go with the former curve from 1 to 2. The compression process reaches Point 3 from Point 1. A certain amount of gas is not compressed and returns to the suction line via the pressing-off inlet valve. When the piston moves to the Position 3 (corresponding to the required load), the actuator clears away the force that open inlet valve. The valve plate then returns to the valve seat and the inlet valve closes. The gas remaining in the cylinder is compressed and the compression process begins to go from 3 to 4. When the gas reaches the rated discharge pressure, it is discharged via the

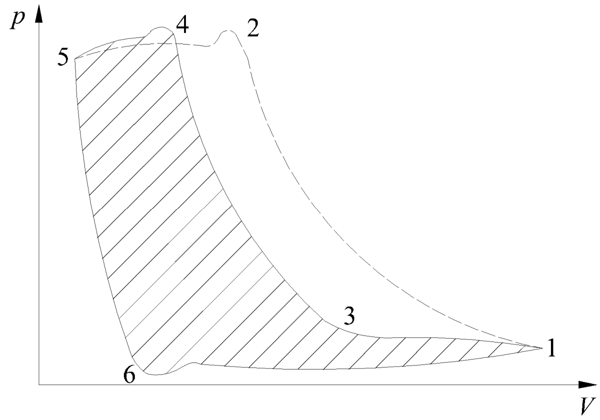

Fig.1Indicator diagram of the pressing-off inlet valve plate during partial stroke

delivery valve; decreasing volume flow. The advantage of this regulation is that the consumption of working input and the actual volume flow come into direct ratio. It is a simple and efficient capacity regulation method for the compressor. The energy needed in the compressor operation can be shown by the closed curve area formed by 1-3-4-5-6 in Figure 1.

2.1 Actuator

The structure of the actuator is shown in Figure 2. The actuator is composed of an armature, a limiter, a core, a coil, a connecting rod, an unloader, springs, and other parts. It works based on the principle of electromagnetism. The power for the unloader is provided by the electromagnet created by winding

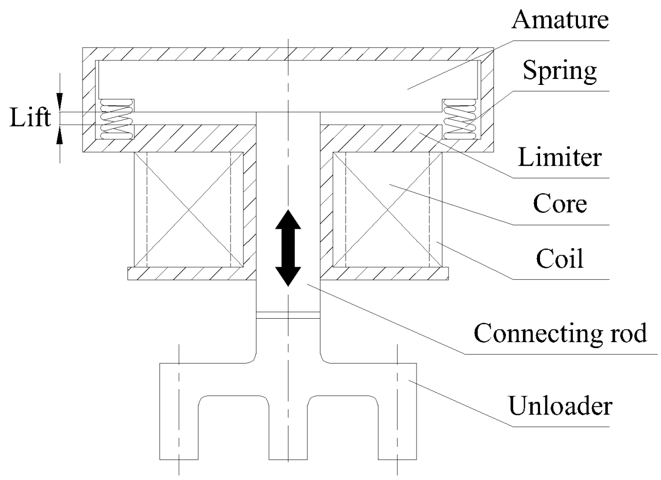

Fig.2Schematic diagram of electromagnet-based actuator the core with coils. To easily obtain or lose magnetization for the electromagnet when the coil is turned on or off, silicon steel is used for the core. The armature, also made of multi-layered silicon steel, is the main moving part in the actuator. The initial gap between the armature and the limiter is the maximum lift of the inlet valve. The connecting rod is made of non-magnetic, non-conductive, and lightweight material to realize easy movement without the influence of electromagnet, and to pass the movement of the armature to the unloader. When the piston enters the suction stroke, the coil is energized and the core is magnetized. Under the effect of the magnetic field, the armature overcomes the spring force and moves to the limiter. The unloader is driven by the connecting rod to press off the inlet valve plate to the maximum lift. As the piston enters the compression stroke, all or some of the gas in the cylinder flow back to the intake line via the opened inlet valve. When the open time of the inlet valve reaches the preset value, the coil is turned off and the magnetic field disappears. The armature is then returned to initial position by spring force. The force to press off the valve plate vanishes, the inlet valve closes, and the gas begins to compress. Therefore, only the needed amount of gas can be compressed to achieve capacity regulation by controlling the on-off time of the coil.

2.2 Control system

The control system of the actuator is composed of a microcomputer, a proximity switch, a PLC [11], and other components. The microcomputer controls the operating mode of the PLC as a supervisory computer and programs it. The proximity switch monitors the key-phase signal on the wheel of the reciprocating compressor and provides a digital input signal for the PLC. The low-cost proximity switch satisfies the requirements of the regulating system in response frequency and function. Therefore, the switch could be used as a substitute for the relatively expensive eddy-current transducer. The PLC adopts the circular scanning mode, and each circle is called one scanning cycle [12]. In one scanning cycle, the PLC performs all or any part of the following requirements: read the input, execute the control logic in the program, process the communication requests, execute CPU self-check, and write the output. The sequential scanning is easy because the programming has been simplified, guaranteeing the reliable operation of PLC [13]. An interrupt mode may occasionally suspend the program, which is scanned in case of emergencies. In the regulating system, the PLC controls the on-off time of the coil according to the input signal, to achieve the opening and closing of the inlet valve. The PLC measures the actual speed of the compressor crankshaft under different loads to provide reliable parameters for actuator movement.

The control system works according to the external input signals. First, the noise of the pulse signal input by the proximity switch is removed by the PLC digital input filter to lower the probability that the input state may unexpectedly change. The active duration of digital input signal should be longer than the stated delay time of the filter to be effectively read by the PLC. For instance, in the regulating system, the duration the key grove of the wheel is detected by the proximity switch probe should be longer than the stated filtering time. Some high-level or low-level pulses with short duration cannot be read by the CPU during periodical scanning. At this moment, the pulse capturing function is set to latch the changed state of the input end, and will not refresh it until the next scanning cycle. This way, a short pulse is captured and maintained until PLC reads the input point. To boost the sampling frequency, immediate refreshment is adopted for input signals.

The preset on-off time of the coil during a piston working cycle under different compressor loads is determined based on the nominal crankshaft speed of the load. In this paper, an on-delay timer of PLC is applied to turn off the coil and achieve delay in the closure of the inlet valve. When the piston enters the suction stroke, the proximity switch provides a digital pulse signal for the PLC. The PLC turns the coil on according to the rising edge of the signal and then the unloader presses off the valve plate. At the same time, the on-delay timer starts and the preset time is loaded. When the timer reaches the preset time, the PLC turns the coil off, so the open force disappears and the inlet valve closes. By changing the preset time, the open time of the inlet valve can be controlled to achieve continuous stepless capacity regulation for the compressor.

The crankshaft speed of the compressor is not unique under different loads, and there is a deviation between the nominal and actual speeds of the load. Therefore, the actual speed of the crankshaft should be measured to provide a reliable parameter for determination of the actuator's motion law. In this paper, frequency and frequency/cycle speed measurement methods [14, 15] are applied to measure the actual speed of the crankshaft.

The frequency method is realized using the timer interrupt and high-speed counter functions of the PLC. The speed pulse signal is counted by the high-speed counter in the preset sampling time. The speed of the crankshaft can be calculated by reading the current valve of the counter through the time-base interrupt. As shown in Figure 3, although the sampling time is a specified value, the starting and ending time of the sampling is not always the moment when the pulse jumps. The method will produce a maximum error of ±1 speed pulse. The higher frequency of the speed signal, the higher measurement accuracy will be obtained. Therefore, the method is suitable for measuring high speed, or the situation that more speed pulses generated in one rotation period of the measured object. When there is one pulse signal generated by the compressor wheel in the reciprocating cycle of the piston, the speed can be calculated by the following equation:

$$\omega = \frac{60 \times n}{T} \tag{1}$$

where ω is the crankshaft speed, T is the preset sampling time, and n is the number of signal pulses counted in the sampling time.

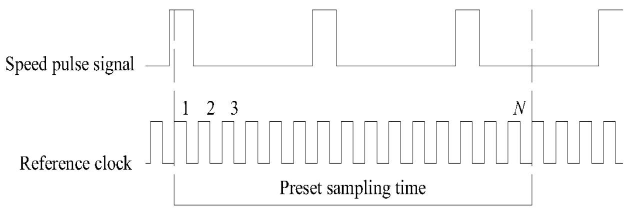

Fig.3Sequence diagram of frequency measurement based on time-base interrupt

The frequency/cycle method is realized using the high-speed counter interrupt and timer functions of the PLC. The speed pulse signal is counted by the high-speed counter, and the on-delay timer is started using the rising edge of the first pulse signal. When the preset pulses are counted, the value of the timer is read through the interrupt function. The counting of the default pulses should be less than the default time of the timer; otherwise, the timer will overflow. As shown in Figure 4, the method will produce a maximum error of +1 reference clock. Usually, the rotation period of the measured object is much longer than the reference clock of the timer, so the error can be ignored. The speed can be calculated by the following equation:

$$\omega = \frac{60 \times (N-1)}{t} \tag{2}$$

where N is the preset pulses, and t is the time needing to count the preset signal pulses.

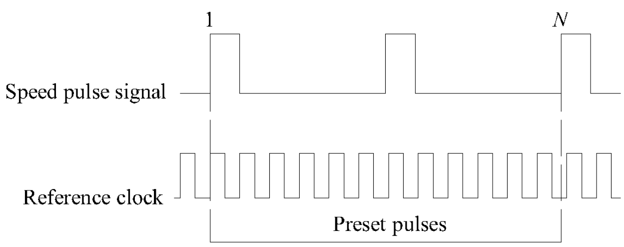

Fig.4Sequence diagram of frequency measurement based on speed counter interrupt

3 Theoretical analysis for the system

The relationship between the compressor loadand the revocation of open force, the value of the open force, and the motion law of the actuator should be derived theoretically to provide a basis for designing a capacity regulation system based on the PLC.

3.1 Compressor load

The compressor load η is equal to the ratio of the actual volume of the compressed air Q to the maximum volume to be compressed Q_{max}. To simplify the regulation system design, presumably, there are no air leaks or clearance volume loss [16], and the motion of the air in the cylinder is isothermal to the ideal gas flow. During the reflow process, the instantaneous sweep volume by the piston is equal to the gas volume that returns to the inlet line. The remaining volume of the cylinder after reflow is the actual gas displacement of the compressor. From Figure 5, the piston stroke is

$$\begin{aligned}\chi &= r + l - (r\cos\theta + l\cos\beta)\\ &\approx r\left(1-\cos\theta+\frac{\lambda}{2}\sin^2\theta\right),\left(0\le\theta\le 360°\right)\end{aligned} \tag{3}$$

Where θ is the compressor crank angle, r represents crank radius, and l is the center-to-center spacing of the two ends of connecting rod, $\lambda=r/l$.

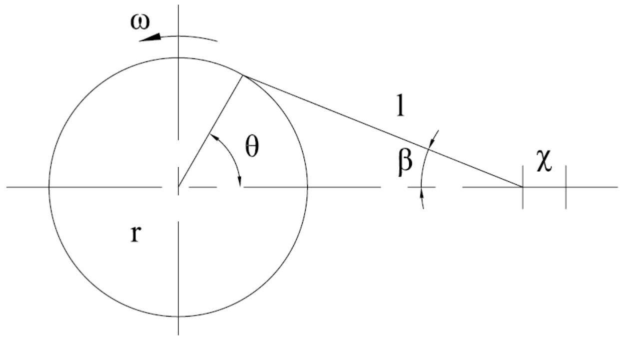

Fig.5Geometric relationship between piston stroke and crank angle

Taking the distal point of piston location as the initial reference of the regulation system, when the coil is turned off and the inlet valve is closed, compression starts when the piston is in compression stroke and the crank rotates at an angle $\theta'\left(180°\le\theta'\le 360°\right)$. The compressor load can be approximated as

$$\begin{aligned}\eta &= \frac{Q}{Q_{\max}} = \frac{\chi}{2r} = \frac{1-\cos\theta'+\frac{\lambda}{2}\sin^2\theta'}{2}\\ &= \frac{1-\cos\omega t+\frac{\lambda}{2}\sin^2\omega t}{2}\times 100\%,\left(T_c/2\le t\le T_c\right)\end{aligned} \tag{4}$$

From the foregoing equation, the desirable load for the compressor can be achieved by changing the open time of inlet valve t. T_c represents the reciprocating cycle of the piston. The error caused by the idealization of gas motion in the cylinder can be corrected through a PID (Proportion Integration Differentiation) [17] control algorithm according to the experimental data, so the displacement is adjusted to the actual condition.

3.2 Open force

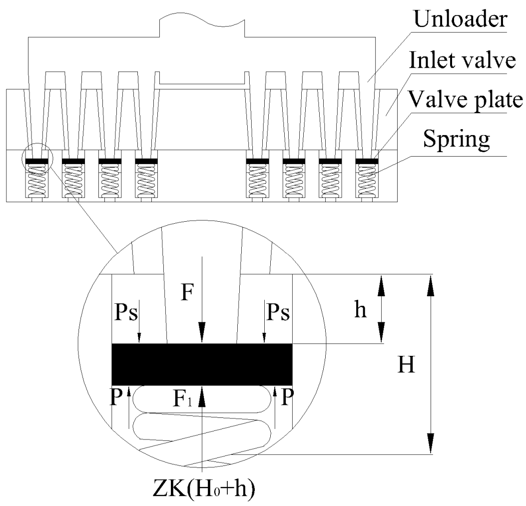

Fig.6Diagram of pressing-off inlet valve plate using unloader

As the piston enters the compression stroke, the gas reverse thrust F_1 acting on the valve plate increases gradually. From figure 6, F_1 can be calculated by the following equation:

$$F_1 = \left(\beta\alpha A_P\right)\Delta p_V \tag{5}$$

Where β is the thrust coefficient, α is the resistance coefficient, A_P is the flow area of valve seat outlet, and Δp_V is the differential pressure. And

$$\Delta p_V = \frac{k\pi^2}{8}\left(\sin\theta + \frac{\lambda}{2}\sin 2\theta\right)^2 M^2 p \quad (6)$$

Where k is the adiabatic index, M is the gap mach number, and p is the cylinder pressure. When (6) is combined with (5), the result is

$$F_1 = \frac{\beta\alpha k\pi^2}{8}\left(\sin\theta + \frac{\lambda}{2}\sin 2\theta\right)^2 M^2 p A_P \quad (7)$$

The external force F_2, which the unloader suffered is the gas trust and the spring force:

$$F_2 = \frac{\beta\alpha k\pi^2}{8}\left(\sin\theta + \frac{\lambda}{2}\sin 2\theta\right)^2 M^2 p A_P + ZK\left(H_0 + h\right) \quad (8)$$

Where ZK is the spring force, H_0 is the pre-decrement of the spring, and h is lift of the inlet valve. From the equation (7), the gas reverse thrust reaches the maximum, when the crank angle is equal to $270°$. Therefore, the maximum external force that unloader suffered during the reflow stroke is

$$F_{2\max} = \frac{\beta\alpha k\pi^2 M^2 p A_P}{8} + ZK\left(H_0 + h\right) \quad (9)$$

Under the control of the regulation system, the requirement of the regulation range can be satisfied only the pressing-off force of the unloader, which is the magnetic attraction on the armature by the electromagnet, is bigger than $F_{2\max}$. There is no need to change the pressing-off force with the compressor load, to simplify the system design.

3.3 Motion law of unloader

To realize capacity regulation by gas back flow, the unloader should press off the inlet valve plate before gas compression. If the unloader does not correctly press the valve plate, or if there is mutual impact between the two components, the back flow of the gas would be imbalanced and the working life of the components is shortened. Therefore, the reliability of the entire system is reduced. The ideal situation is that the unloader moves and correctly presses off the valve plate when the plate is in maximum lift before back swing.

The motion law of the inlet valve plate during suction stroke should first be studied to determine the turn-on time of the coil for driving the unloader. The set of equations for valve plate movement [18, 19] is resented below.

$$\begin{aligned}
&\frac{dh}{d\theta} = y \\
&\frac{dy}{d\theta} = \left[G_{12}(\varphi - 1)\frac{1+G_8 h^2}{\sqrt{1+G_7 h^2}} + ZK\left(H_0 + h\right)\right] \Big/ G_9 \\
&\frac{d\varphi}{d\theta} = -[G_1\varphi\sin\theta + G_2\varphi\sin 2\theta + \frac{G_3 h}{\sqrt{1+G_7 h^2}}\varphi^{G_{10}} \times \\
&\sqrt{\left|\varphi^{G_{11}} - 1\right|}] \Big/ \left(G_4 + G_5\cos\theta + G_6\sin^2\theta\right)
\end{aligned} \quad (10)$$

Where $G_1 = k\omega V_h/2$, $G_2 = k\omega V_h\lambda/4$, $G_3 = -2\alpha_v N_v k\pi\sum d_{cp}\sqrt{2Rk(T_s + 273)/(k-1)}$, $G_4 = \omega V_h(\varepsilon + 0.5)$, $G_5 = -\omega V_h/2$, $G_6 = \omega V_h\lambda/4$, $G_7 = \left(2\alpha_v\pi\sum d_{cp}/(\alpha_e A_e)\right)^2$, $G_8 = \left(2\alpha_v\pi\sum d_{cp}\right)^2/\left(A_e A_p\right)$, $G_9 = -M_v\omega^2$, $G_{10} = 1/k$, $G_{11} = (k-1)/k$, $G_{12} = A_p p_s$, V_h is the cylinder capacity, a_v is the flow coefficient of valve clearance, N_v is the number of inlet valve, $2\pi\sum d_{cp}$ is the circumference of the valve plate, R is the gas constant, T_s is the suction temperature, ε is the relative clearance volume, a_e is the flow coefficient of valve sheet passage, A_e is the area of valve seat passage, M_v is the equivalent motion mass of valve, and p_s is the suction pressure.

The variation law of valve plate displacement, velocity, and cylinder pressure with the crank angle can be obtained by computer simulation according to Equations (10) to determine the time for energizing the coil. The turning off time for the coil can be

Table 1 Parameters of the compressor

ω (Nominal)	875 r/min	λ	1/3.6	k	1.4	ZK	456 N/m	H_0	0.0080 m
h	0.0020 m	V_h	0.0040 m^3	a_v	0.69	N_v	1	$2\pi\sum d_{cp}$	0.512 m
R	287.04 $m^2/s^2 \cdot k$	T_s	6 ℃	ε	0.032	a_e	0.67	A_e	0.00182 m^2
A_P	0.0032 m^2	M_v	0.060 kg	p_s	0.98×10^5 N/m^2	Resilient coefficient	0.3	Step length	0.01°
Initial angle	38°	Close angle	142°						

calculated by Equation (4) to obtain a preset time for the PLC timer.

4 An application case

In this paper, a one-cylinder reciprocating compressor is adopted as the test device. The specific parameters of the compressor are shown in Table 1.

Taking the situation η=90% as an example, theelectromagnet force to press off the valve can be determined by the equation (9) using the parameters in table 1. No concrete description is given for specific calculation in this paper. The movement law of inlet valve with the change of crank angle can be obtained through Matlab [20] simulation using the Runge-Kutta method according to the parameters in Table 1 (Figures 7-9).

When the nominal speed of the crankshaft is selected as the initial speed under 90% load, the open time of the inlet valve can be calculated by Equation (4):

$$t_{open} = \frac{\arccos\left[\left(\sqrt{1-1.6\lambda+\lambda^2}-1\right)/\lambda\right]}{\omega} \text{ s}$$
$$= \frac{222.57}{5.25}\times10^{-3} = 42.4\times10^{-3}$$

From the Figures 7-9, when the crank angle is 50°≤ θ ≤140°, the valve plate is in the maximum lift, the speed of the plate is in low fluctuation and tends to 0, and the cylinder pressure is stable. When the crankshaft turns the angle θ, the time needed is $t=\theta/\omega$. Therefore, to avoid collision between components and ensure the valve plate is correctly pressed, the unloader should move and come in contact with the valve plate during the time period

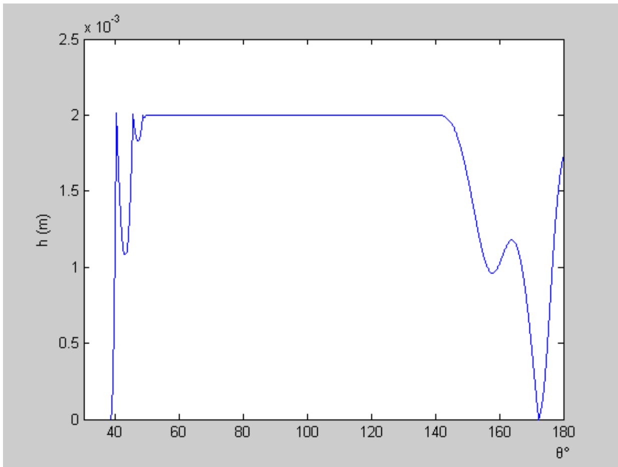

Fig.7Variation of valve plate displacement h with crank angle θ

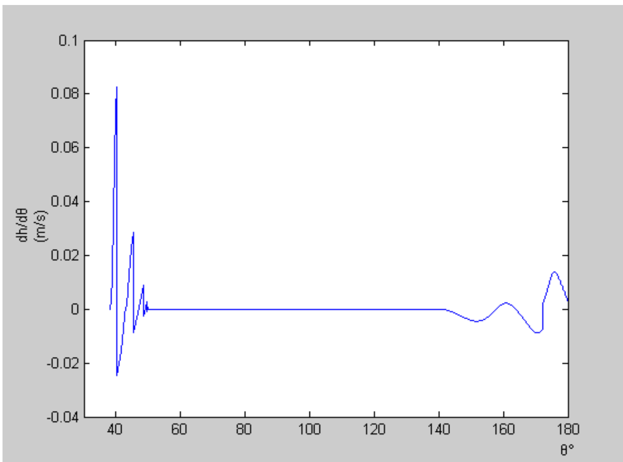

Fig.8 Variation of valve plate speed $dh/d\theta$ with crank angle θ

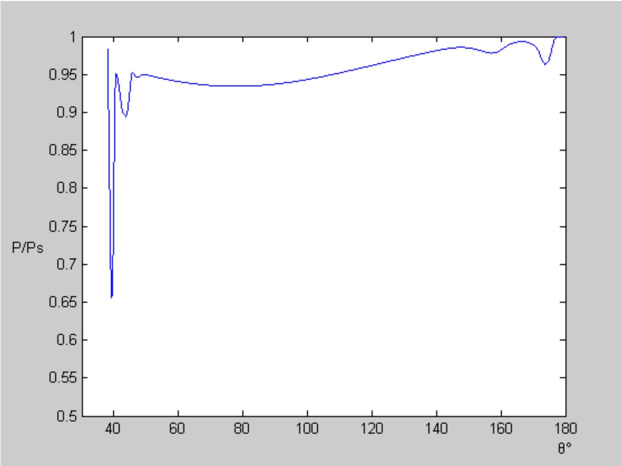

Fig.9 Variation of cylinder pressure p with crank angle θ

9.5×10^{-3}—26.7×10^{-3} s from the initial reference. In this case study, the time for coils to energize is 20.0×10^{-3} s. Therefore, the preset time Δt that loading in the timer requires for turning the coil off is

$$\Delta t=(42.4-20.0)\times10^{-3}=22.4\times10^{-3}\ \text{s}$$

As known from the nominal speed of the crankshaft, the rotating signal of the wheel to be measured is about 14.58Hz. The timer interrupt-based method could ensure the measurement accuracy, but the sampling time will be increased so the real-time is decreased. Therefore, the high-speed counter interrupt function of the PLC is applied to measure the needing time for counting preset pulses, to calculate actual speed of the crankshaft. The specific PLC programming for speed measurement is resented below.

Main (Calling sub routine, define timer)		
LD	SM0.1	// First scan flag
CALL	SBR_0	// Calling sub routine 0
R	M0.0, 1	// Reset M0.0
LDI	I0.3	//
O	M0.0	//
=	M0.0	//
TON	T32, 1500	// Defining the time of timer T32 as 1500ms
Sub routine SBR_0 (Define high-speed interrupt)		
LD	SM0.0	// Operation flag
MOVB	16#F8,SMB147	// Loading the control bit of HSC4
HDEF	4, 0	// Defining the mode of HSC4 as 0
MOVD	+0, SMD148	// Current value of HSC4
MOVD	+21, SMD152	// Preset value of HSC4
ATCH	INT_0, 29	// Connect interrupt event 29 to INT_0
ENI		// Global interrupt enable
HSC	4	// Start HSC4
Interrupt sub routine INT_0 (Speed measurement, reset timer and high-speed counter)		
LD	SM0.0	// Operation flag
MOVW	T32, VW0	// Read the current value of timer
ITD	VW0, VD2	// Convert to double integer
DTR	VD2, VD6	// Convert to real
/R	1200.0, VD6	// Convert to speed
ROUND	VD6, VD10	// Rounding
R	M0.0	// Reset M0.0
MOVD	+0, SMD148	// Reset HSC4

Once the regulating system is operational, the time for the PLC high-speed counter to count 21 high-level pulses input by the proximity switch under 90% load is 1364 ms. Using Equation (2), the calculated reciprocating cycle of piston is T_c=68.2 ms and the crankshaft speed is ω=880 r/min. The initial speed will be ceaselessly refreshed according to the actual speed and modified preset time of the timer to reduce the error between actual and required displacements.

5 Conclusion

From the design process and application, the PLC-based capacity regulation system could achieve continuous stepless capacity regulation from 0% to 100% by controlling the on-off time of the coil to satisfy production requirements. Unnecessary energy losses are thereby decreased. The regulating system features good economical efficiency because the processing fee for the actuator is low and the control system adopts relatively low-cost elements, such as proximity switches and PLC. The actuator that has a simple structure can instantly and reliably press off the valve plate. The application and revocation of the open force can be easily controlled by the PLC timer function, and the actual speed of the crankshaft can be accurately measured by the PLC high-speed counter interrupt and timer functions. The PLC is well-known for reliability, thus, the safety of the overall system is boosted.

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