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# A new non-destructive method for fault diagnosis of reciprocating compressor by measuring the piston rod strain

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**Abstract**. As an established method to analyse the thermodynamic process inside the cylinder, the p-V diagram can identify typical faults such as leakages through the sealing rings and self-acting valves, loosening and fluttering of the vales, etc. However, the installation of the pressure sensor within the cylinder for recording the p-V diagram may lead to degradation of the cylinder integrity, and hereby potential leakage and lowered strength of the cylinder. This paper presents a new non-destructive method for fault diagnosis of the reciprocating compressor by measuring the piston rod strain, from which the p-V diagram was transformed. An algorithm for reconstruction of the p-V diagram was proposed based on the key feature points on the piston rod load curve that reflect the opening and closing events of the compressor valves. The reconstruction method was validated by comparing the calculated p-V diagrams with the experimental results. The reconstructed p-V diagrams were further used for fault diagnosis of the reciprocating compressors, with which the acoustic emission method was integrated. The results show that this method can easily monitor the working conditions and identify whether and where leakage, fluttering and delayed closing of the valves occur, which indicates that this method can be utilized as a powerful tool of non-destructive condition monitoring and fault diagnosis of the reciprocating compressors.

**Key words:** reciprocating compressor; fault diagnosis; non-destructive testing; strain

## 1. Introduction

Reciprocating compressor is widely used in in refining, chemical, petrochemical plants and gas transmission pipelines, etc. mainly owing to its availability of high-pressure ratio [1]. However, the reciprocating compressor has a complicated structure with many vulnerable components such as self-acting valves, sealing rings, piston rod and so on, which usually lead to unscheduled shutdown of the compressor and result in the safety accidents and economic losses. As reciprocating compressor is the core component of many critical systems in process industries, it is very important to ensure the safe and continuous operation of reciprocating compressor. Until now, various physical state parameters have been used to monitor the operating status of the compressor, such as vibration [2,3], temperature [4], pressure [5] and acoustic emission [6,7].

Pressure in the cylinder are usually converted into a pressure-volume (p-V) diagram for analysis, which describes the relationship of internal pressure with the volume of the compressor cylinder [8]. Matsumura et al. [9] employed the p-V indicator diagram as a method to diagnose valve malfunction. Up till now, typical approach for recording the pressure signal in the cylinder is installing a pressure sensor within the cylinder or alternatively through the compressor valves[10]. Liu et al. [11] proposed a method for recording the pressure signals by punching holes in the bolts to transmit the pressure in the cylinder to the pressure sensor mounted on the bolts through the pressure guiding tube, which significantly promoted the advancement of fault diagnose technology of p-V diagram. Similar methods have been used for recording the p-V diagram in the twin-screw compressor. Peng et al. [12] reported another method in which only one sensor was embedded into the groove of the female rotor on the discharge side to measure the whole working process.

However, this measurement method inevitably does harm to the cylinder integrity to install the pressure sensor. Besides, it also does adverse effects on the strength of the cylinder and brings potential threat. Worse still, pressure extraction hole increases the cylinder clearance volume. Such intrusive measurement method severely limits the popularisation and application of the p-V diagram. Hence, the demand exists for a tool with which to better and safer assess dynamically varying pressure in cylinder and for condition monitoring and fault diagnosis.

In this paper, a new method of obtaining the p-V indicator diagram for fault diagnosis is proposed. The piston rod load was measured and used to reconstruct the p-V diagram, which was further validated by comparing it with that obtained from the pressure measurement. Then, the method was applied for fault diagnosis in a reciprocating compressor, in which two typical operating conditions i.e. suction valve leakage, discharge valve leakage was mocked up.

## 2. Methodology

Figure 1 shows the schematic diagram of the working cycle and force analysis of the reciprocating compressor in single and double working mode. The position of piston is determined by the crank angle. The top dead centre (TDC) is the origin of coordinates of the piston displacement. The working volume of cylinder can be obtained from the following equations:

$$x = l + r - (lcos\beta + rcos\theta) \tag{1}$$

where $\theta$ is crank angle of crank, $x$ is the piston displacement, $l$ is the length of the connecting rod, $r$ is the crank radius, $cos\beta = 1 - \sqrt{1 - \lambda^2 sin^2\theta})$ can be obtained from Figure 1 and the λ is the ratio of the crank radius to the length of the connecting rod, thus:

$$x = r\left[(1 - cos\theta) + \frac{1}{\lambda}(1 - \sqrt{1 - \lambda^2 sin^2\theta})\right] \tag{2}$$

The working volume of cylinder is calculated as:

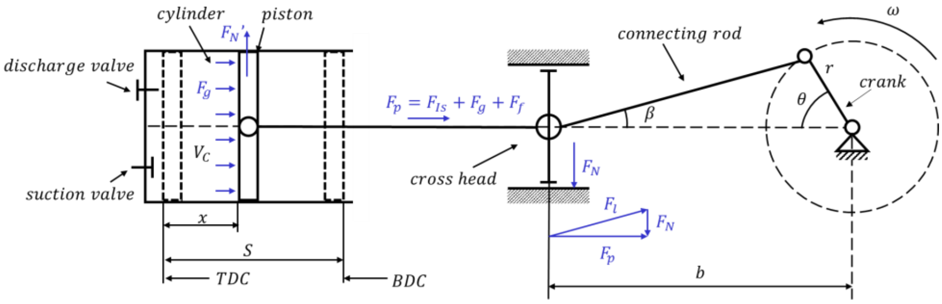
$$V_C = V_0 + \frac{\pi D^2}{4}x \tag{3}$$

where $V_0$ is the clearance volume, which is fixed when the compressor is designed, and $D$ is the cylinder diameter.

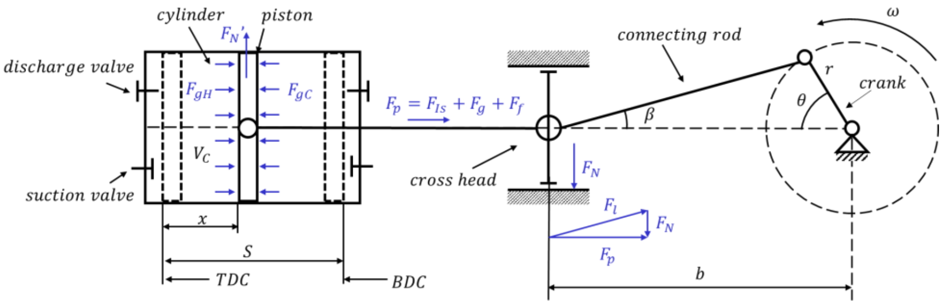
For the piston rod load, it is mainly comprised of gas force, reciprocating inertial force and friction. Thus, resultant of the gas force $F_g$ can be obtained:

$$F_g = F_p - F_{Is} - F_f \tag{4}$$

where $F_{Is}$ is the reciprocating inertial force, $F_f$ is friction, and $F_p$ is the piston rod load; In this study, the piston rod load $F_p$ was computed based on the strain measured by a strain gauge pasted to the piston rod.




(a) Single-acting compressor




(b) Double-acting compressor

**Figure 1.**The schematic diagram of the working cycle and force analysis of a double-acting reciprocating compressor

For the single-acting compressor, $F_g$ is the gas force at the working end. Thence, the pressure change at the working end can be calculated directly from the gas force. For the double-acting compressor, is the resultant force of the gas forces on the crank-end and the head-end. How to extract the pressure values on both sides from the gas force curve belongs to a blind source separation (BSS) problem. We promote an algorithm for reconstruction of the p-V diagram based on the key feature points on the piston rod load curve that reflect the opening and closing events of the compressor valves, with which the controlled iterative optimization is integrated.

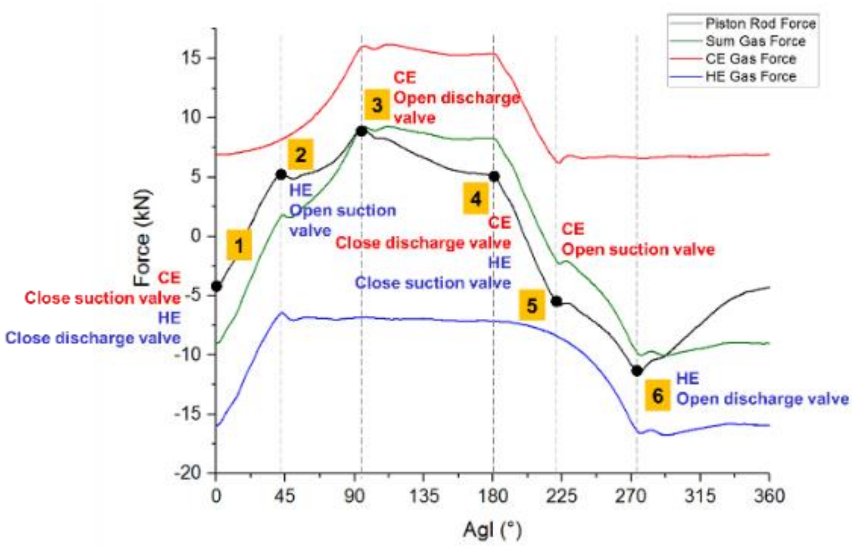



**Figure 2.** Gas force, piston rod load and the feature events

Eight valve events (opening and closing times of suction and discharge valves on HE and CE) can be indicated on the gas force curve and piston rod curve as six feature points (Figure 2). The six feature points are in accordance with the gas force balance equation:

$$p_{sCE}(i) \times S_{CE} - p_{dHE}(i) \times S_{HE} = F_g(i) \tag{5}$$

where $HE$ is head end, $CE$ is the crank end, $i$ is number of the feature point($i$=1,2,3,4,5,6), $p_s$ is suction pressure, $p_d$ is discharge pressure. In the process of compression and expansion, the variation rate of the pressure satisfies the established equation, and in the process of suction and discharge, the variation rate of the pressure is proportional to that of the piston displacement [13].

According to the established method for the theoretical calculation of p-V diagram. The p-V diagram can be deduced preliminary based on the parameters solved from the Eq. (5). Subsequently, the calculated pressure results are iteratively adjusted to make the corresponding calculated gas force based on the reconstructed gas force. Thereby, the optimised p-V diagram is achieved.

## 3. Experimental investigation

In order to verify the non-destructive method being used for obtaining p-V diagram or performance prediction, an experimental study on recording the piston rod strain, the pressure in the cylinder and the TDC (top dead centre) signal is carried out. The experimental facilities consist of a reciprocating air compressor, a strain gauge, two pressure sensors, a remote optical sensor.

The acquisition system is built with National Instruments (NI) hardware and software. The C series modules were plugged into a cDAQ chassis to ensure that all the signals of different channels were acquired synchronously. The sampling frequency for strain is 4kHz and 10 seconds' data was saved for analysis in each condition. Figure 3 shows the experiment facilities and the air compressor used for the experiential investigation, which is modified from a two-stage double-acting water-cooled and reciprocating air compressor for industrial application.

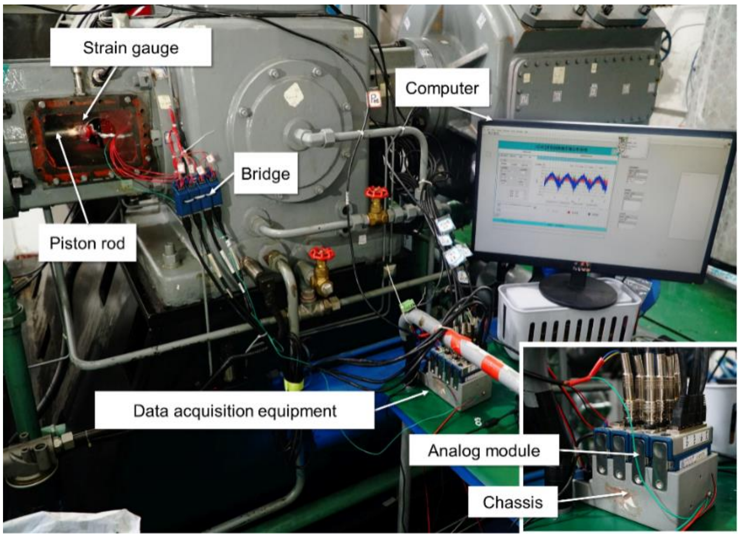



**Figure 3.** Fault diagnosis testbed

The quarter bridge arrangement with the gauge in Figure 4 can measure the external of the force-initiated minute elongation or shrinkage (strain) on the side surface of the piston rod. The strain gauge is preferably located along the axial direction on the side of the piston rod, depending on the tension and compression load direction of the piston rod. The voltage signal produced by the bridge is proportional to the strain and can be converted to tensile and compressive stresses (Figure 4). The piston rod load can be computed.

The method for recording indicator diagrams by measuring pressure in this paper involves the use

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of two pressure sensors installed respectively into the discharge valve studs of crank-end and head-end, as shown in Figure 5, to allow the actual pressure in both two sides of the cylinder to be recorded instantaneously.

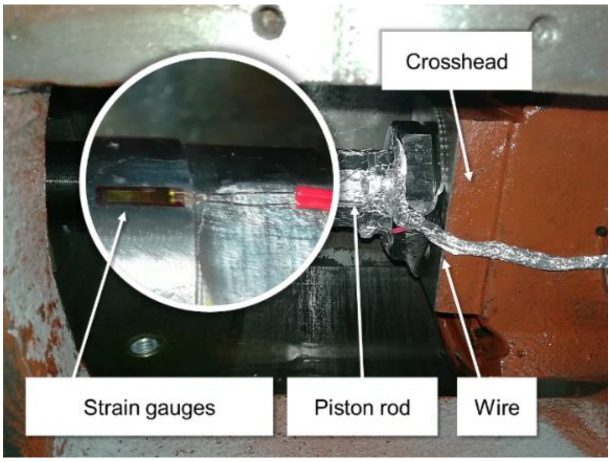



**Figure 4.** Quarter-bridge arrangement with the gauge

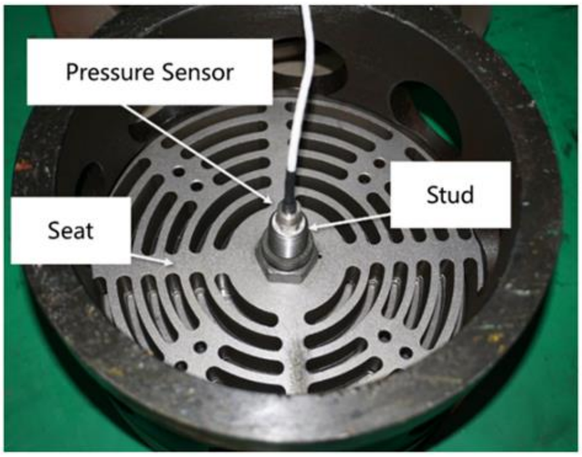
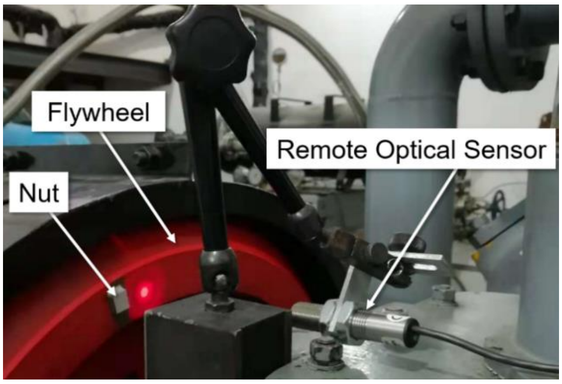
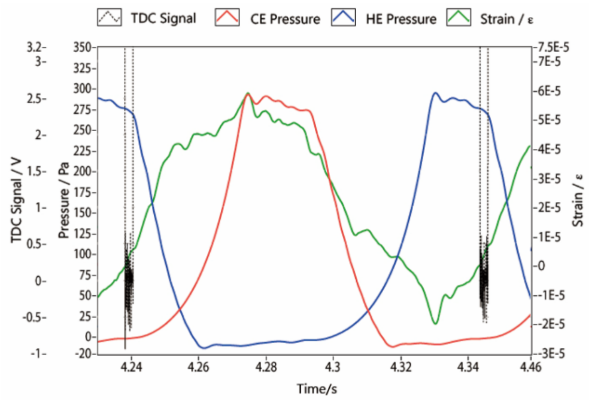



**Figure 5.** Arrangement of pressure sensor

For the piston location was correlated with the rotational location of the crank and flywheel, an remote optical sensor was attached to the steel frame, and the special nut is facing the remote optical sensor was installed on the flywheel when the piston reached the TDC, as shown in Figure 6(a). Thus, a pulse signal was generated when the piston was at the TDC during the working cycle, which is the TDC signal. When the piston is at TDC, the crank angle is 0°. Figure 6(b) shows the pressure signal, strain signal and TDC signal for two complete working cycle. The pressure signal and strain signal are mapped from the time domain onto the crank-angle domain according to the TDC signal.




(a) Arrangement position of remote optical sensor



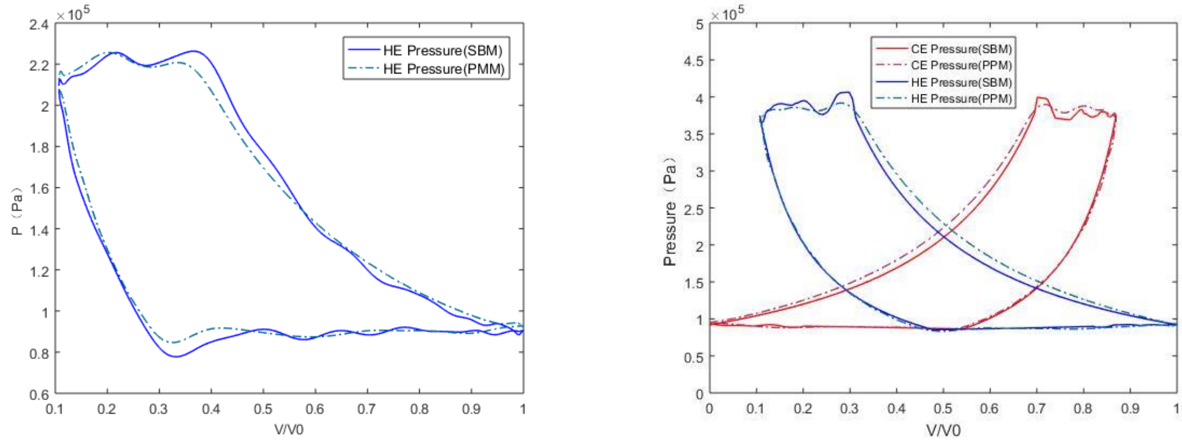

(b) The tested TDC signal

**Figure 6.** Installation of remote optical sensor and the tested p-t diagram

## 4. Results and Discussion

### *4.1. Validation of the reconstructed method for obtaining p-V diagram*

To validate the reconstructed method for obtaining p-V diagram, the reconstructed p-V diagrams and the measured p-V diagrams in the same working condition for single-acting and double-acting working modes are shown together in Fig.7. The p-V is reconstructed by the method introduced in section 2 based on the strain collected by the strain gauge in Figure 7 (a). The measured p-V diagram is recorded by the pressure sensor as shown in Figure 7 (b).




(a) Single-acting mode (b) Double-acting mode

**Figure 7.** Comparison of reconstructed p-V diagrams with measured p-V diagrams

The reconstructed results are nearly in agreement with the measured results, especially in the re-expansion progress in both single-acting and double-acting mode. This indicates that the proposed model in this study is valid and can be used to predict the change of pressure in the cylinder. However, some errors can still be found in the curves. From both the reconstructed and measured results, it can be found that the pressure pulsation in the discharge stroke is much more than that of suction stroke. Comparing the p-V diagrams of two method, the pressures reconstructed by piston rod load is a bit higher than that measured by the pressure sensor in discharge stroke, while the reconstructed pressure is slightly lower when the suction valve open. It may be attributed to that, in the experiment, the piston ring and the filler which complete sealing fluid in the friction process, and its working mechanism is complicated, and its working characteristics are unstable, and it is difficult to determine its regularity. The refraction algorithm does approximate processing to the friction force and does not consider the influence of oil film thickness between piston rod and oil scraper ring, piston movement speed, heat transfer and other factors on the friction force. Therefore, there is a small amount of error in the calculation of gas force. However, as can be seen from the Figure 7, the calculation results can still accurately reflect the trend of pressure pulsation. As to double-acting working mode, the p-V diagram is reconstructed based on the feature points on the feature points and established calculation formula, where both the compression and expression indexes are considered as constant. Nonetheless, because of the complex heat transfer during compression and expression stroke, the indexes change with the direction and intensity of heat transfer, which induces some error between reconstructed and measured methods. Measurement results of pressure in the cylinder and piston rod load.

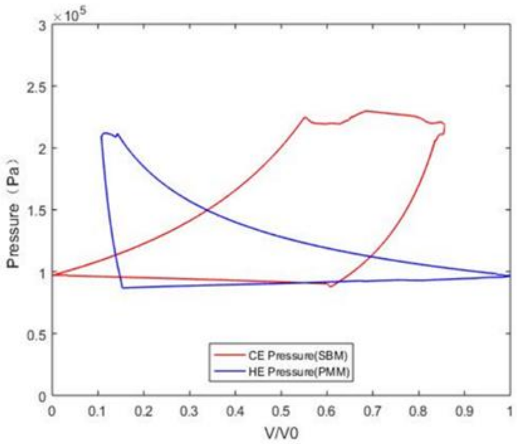
### *4.2. Method application in fault diagnosis*

#### *4.2.1 Diagnosis of suction valve leakage.*

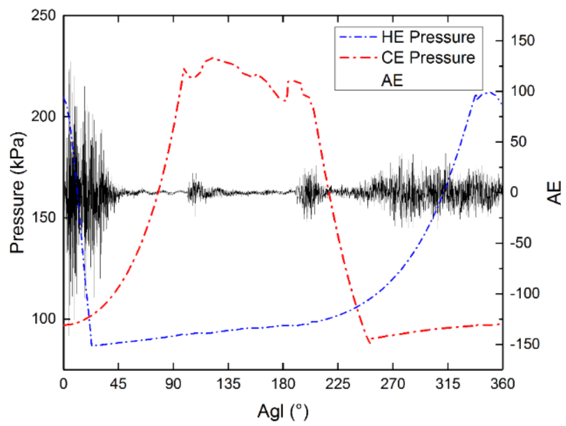
In order to simulate the valve leakage fault, for the suction valve of HE, the feeler gauge of different thickness was fixed between the valve seat and the valve plate. In this way, the valve plate will not close completely with valve seat and gas will escape from the valve. Figure 7(c) shows the actual fault simulation.

Figure 7(a) illustrates the reconstructed p-V diagram with the suction pressure of 0.097MPa and the discharge pressure of 0.22MPa when the suction valve leaks on head-end of the cylinder. Compared to the normal curve of crank-end, the p-V curve on the head-end produces severely distorted. It can be explained as follow. During compression, since gas leaks out through the suction valve, the piston must move further to reach the discharge valve open pressure. Hence, the compression slope occurs more slowly than normal. During the discharge stroke, gas exits through both the suction and discharges, which is reflected in the shorter discharge progress in the p-V diagram. In the process of expansion, the continuing gas leakage through suction valve leads to a rapid decline on the expansion curve, which makes the pressure in the cylinder decreases to the suction valve open pressure in advance and the suction process to be significantly lengthened.

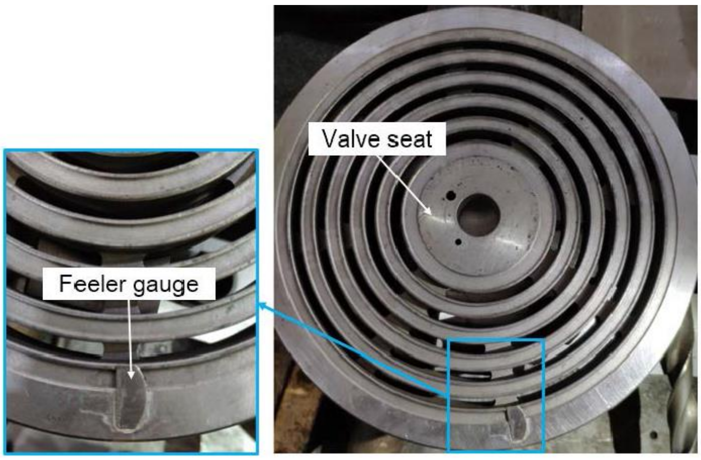
Acoustic emission (AE) technology [14] is used to detect discharge valve leakage, because the leakage leads to an increase in the amplitude of the continuous acoustic emission signal. It can be seen in the Figure 8(b) that the amplitude of the continuous AE signal increases at first and decreases during the time between the suction valve close event (248°) and open event (23°), owing to the suction valve leakage faults.




(a) Reconstructed p-V diagram




(b) AE signal



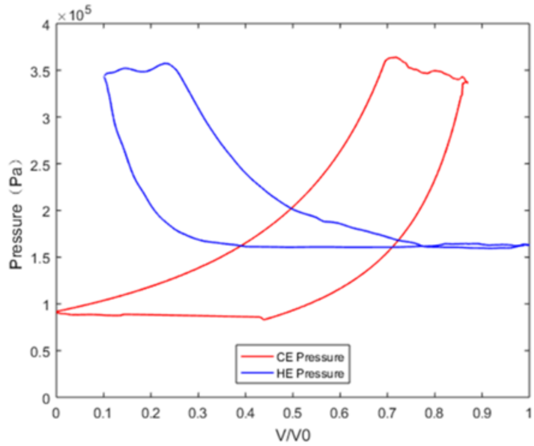

(b)Actual leakage fault simulation on the suction valve

**Figure 8.** Reconstructed p-V diagram of head-end suction valve leak (HE)

*4.2.2 Discharge valve leakage faults*.

Figure 9 (a) presents the reconstructed p-V diagram with the suction pressure of 0.095MPa and the discharge pressure of 0.335MPa when the discharge valve is experiencing leakage on head-end of the cylinder. During the expansion portion in the head-side of the cylinder, the leakage of gas through the discharge valve into the cylinder slows down the rate of pressure drop. However, the pressure in head-end of the cylinder does not decline to the suction valve open pressure, which is because the leakage of the discharge valve is so severe that the suction valve cannot be opened. When the rate of pressure reduction caused by the increase in cylinder volume is agreed with that of pressure increase caused by leakage of the discharge valve, the pressure in the cylinder remains unchanged until the compression process begins. In the compression stroke, the gas continuous leaking into the cylinder. When the pressure in the cylinder rises to discharge pressure the valve opens.

In accordance with above analysis, we inspected the tested reciprocating compressor, and found a piece of rag was sandwiched between the seat and plate of the head-side discharge valve, which obstructed the closing of the valve and resulted in leakage, as Figure 9 (b) shows.




(a) Reconstructed p-V diagram

(b) Inspection result

Figure 9. Reconstructed p-V diagram of discharge valve leakage failure in HE

## 5. Conclusion

An experimental investigation has proved that the method for obtaining a p-V diagram of the reciprocating compressor by measuring the piston rod strain. The experiment has validated the algorithm of reconstruction of the p-V diagram based on the key feature points. Consequently, several conclusions have been made.

- The reconstructed p-V diagrams demonstrate excellent agreement with that recorded by pressure sensors. And the reconstructed p-V diagrams can be used to diagnosis the valve fault.
- The key feature points that reflect the events of the compressor valves can be recognised clearly on the piston rod load curve, and those points are extracted as features to reconstruct the p-V diagram.

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- The severity of the leakage can be determined by the shape of pressure curve. The longer suction process and shorter discharge process can diagnose the suction valve leakage, and the obvious increase of suction pressure in the p-V diagram indicates the serious leakage of discharge valve.

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