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VIBRATION SIGNALS OF RECIPROCATING COMPRESSOR VALVES

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Abstract

This study deals with the problem of the diagnostics of self-acting valves installed in reciprocating machines. Piston compressors and their valves are characterized. Compressor diagnostic systems, those offered and used, are described, including their restrictions. The signal generated by closing the valve is estimated. The authors have examined the relationship between the vibration signal and the condition of plate delivery valve of an air compressor. Simulations were made of the spring and plate wear and valve assembly errors. It has been shown that there is a relation between the selected vibration measure, mean peak-to-peak value of vibration accelerations and the technical condition of the discharge valve and measurement point. The authors have concluded, that in case of plate valves, one should expect that values of vibration measures depend on the measurement point and will decrease depending on its technical condition. A rule of diagnostic inference was formulated.

Keywords: *technical diagnostic, reciprocating machines, self-acting valves, condition of discharge valve, vibration signal*

1. Introduction

Successful vibration-based diagnostics of rotor machines encourage manufacturers of diagnostic systems to develop and offer equipment and systems for piston (reciprocating) machines as well, including piston compressors. Various authors indicate possibilities of effective diagnostics of piston-crank mechanism components. However, certain problems occur when attempts are made to diagnose valves.

2. Compressors and their valves

There is a wide variety of design solutions in piston compressors. Also, various classifications exist. Depending on the piston-crank mechanism, there are compressors with and without crank mechanisms. Depending on the arrangement of cylinders (shape of the crankshaft) compressors can be of in-line, boxer, V-, star- and vane types. If we consider the shape of piston, there will be single-acting, double-acting, differential and stage-differential pistons. Due to piston guiding, compressors are with or without crossheads. Depending on the piston axis position, compressors may be horizontal or vertical.

The medium and required pressure lead to this classification [3]:

– non-critical use: low pressures, gases are not dangerous;
– semi-critical use: high pressures, gases are not dangerous;

– critical use: media are potentially dangerous.

Considering compressor dimensions, e.g. power N of compressor drive, compressors belong to these ranges: $N < 50$ kW, 50 kW$< N < 100$ kW, $N > 100$ kW [3].

The medium being compressed results in another classification: air compressors, hydrogen compressors etc.

For the control of compressor working process valves are used: self-acting and controlled (forced timing). Forced timing is used mainly for controlling the delivery rate of the compressor. The suction valve closing angle is controlled by an electronically controlled hydraulic cylinder. In most design solutions self-acting valves are used.

Self-acting valves are of various types [4]:

– plate valves; these valves, with a plate functioning as the sealing element made of metal or non-metallic materials; this type of valve is used in oil and gas, fertilizer and other industries;
– ring valve; instead of a valve plate, independent rings are used. The rings are opened and closed by springs adjusted to the working conditions. The materials for rings are carbon fiber-reinforced composites, characterized by very high tightness, impact resistance, resistance to plastic deformation and particles, etc. Their significant advantage is that cylinder liner sliding surface is not damaged in case of ring fracture. Rings can be covered with an anti-adhesive material. Ring valves are intended for most extreme working conditions: hydrogen or light gases (molecular weight < 8 kg/kmol) in refineries and oil and gas industry;
– mushroom valves; valves with a number of mushrooms in one assembly are used in compressors installed in natural gas transport systems where large quantities of gas are transferred at relatively low degree of compression, at low or medium flow rate, 180–600 rpm. These parameters allow to apply high lift (max. lift: 8 mm);
– concentric valve; used in single-acting compressors, concentric valves are the most effective solutions, comprising the whole piston surface. The suction and discharge valves are mounted in a joint body of cylinder head. Such solution is characterized by slight flow resistance and high durability. These valves are used in air and refrigerating compressors, sometimes in single-acting gas compressors;
– lamellar valve. Lamellar (tongue) valves are generally an integral part of cylinder head. The lamella of the suction and discharge valve is integrated on one joint plate of the valve seat.

3. Diagnostics of compressors

Depending on the scope of identification of machine condition, condition monitoring and diagnostics are distinguished. As various equipment may be used, condition monitoring and diagnostics can be executed as remote condition monitoring and diagnostics, in-site monitoring and diagnostics or distributed condition monitoring or diagnostics.

In practice [3], compressors are generally equipped with vibration transmitters sending signals to superior systems. TDC sensors are also installed in order to carry out measurements for various angles of the crankshaft by means of a portable measuring instrument. Vibration transmitters are usually mounted on the cylinder block. Major compressors are additionally equipped with vibration transmitters installed on the crosshead guide. Some compressors are fitted with indicator cocks, so that during their operation, if need arises, p–V diagrams can be drawn by portable measuring instruments. During diagnostic inference the measured signals as a function of crankshaft rotation angle are analyzed.

Those dealing with compressor diagnostics face a problem of identifying components of a vibration signal measured on the compressor cylinder. Normal vibration sources are impacts of the piston against cylinder liner and impacts of valves against the cylinder head, dependent on the condition and loads. The difficulty lies in the fact that both impacts occur at a similar rotation angle of the crankshaft, Fig. 1. Other sources of vibration are possible. These activate themselves only after a defect occurs. In [3] a case is described where an increase of acceleration of cylinder vibration was considered as a symptom of incorrect valve condition, while inspection revealed that the cause was piston seizure in the cylinder.

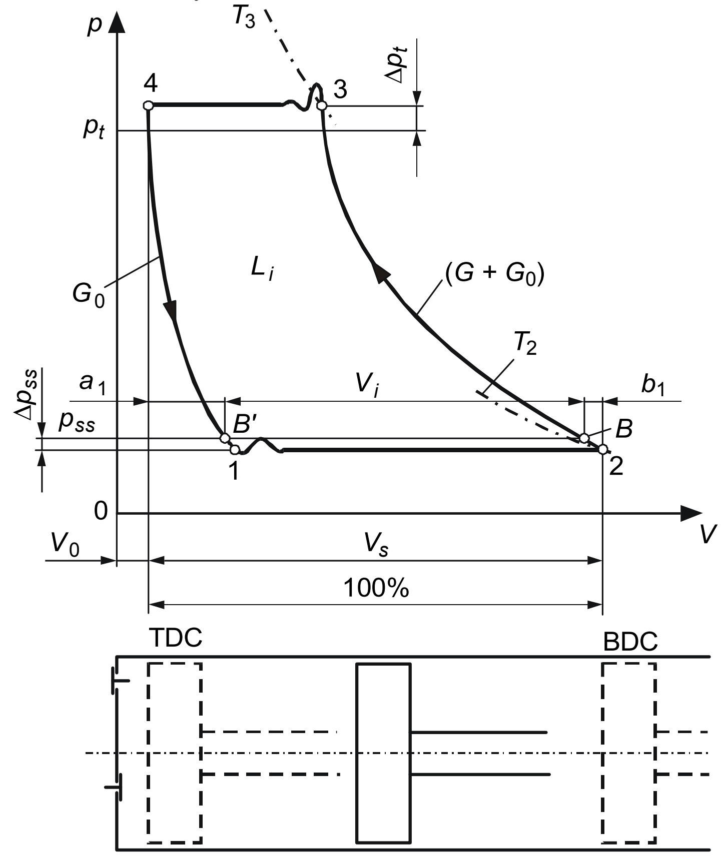

Fig. 1. P-v chart of a single-stage piston compressor: L_i – work of the circulation, TDC – top dead centre, BDC – bottom dead centre (own study on the basis of [2])

The problem of piston impact on the cylinder liner has been discussed in [1]. The impact of self-acting valves requires a separate analysis.

4. Examination of vibration signals from self-acting valves

The impact of valve plate against the valve seat depends on the pressure difference before and after the valve, valve spring tension, plate lift and "elasticity" of the plate and the seat. The mentioned quantities may change due to valve wear. As a result, the impact force may:

– increase, e.g. because of increased valve lift;
– decrease, e.g. due to decreased spring stiffness.

It should be expected that a diagnostic symptom of valves can be either an increase or decrease in the measure value of vibration generated by impacts in valves. The value of vibration measure also depends on the point of measurement. None of the authors on the subject indicates that the cylinder block is the best place for measuring vibration dependent on valve condition; points of measurement, however, are selected for their ease of mounting and conviction that the distance of vibration propagation of all possible sources is the shortest.

A vibration signal from a plate delivery valve was examined, Fig. 2.

The measurements aimed at examining the influence of changes in the technical condition of a discharge valve on the second cylinder of a three-cylinder single-stage air compressor on the vibration signals of the valve, valve plate and cylinder head, Fig. 3. Vibration accelerations were measured. Vibrations on the cylinder block were not measured as its construction made it impossible to measure vibrations by means of available transmitters.

Vibration accelerations as a function of time were measured in combination with a TDC signal from the second cylinder, Fig. 4.

One can conclude from Fig. 4 that:

– time signal of vibration has the highest value near the top dead centre, which means that that part of the signal is generated by the closing of delivery valve (see Fig. 1.),
– instantaneous values of that signal part are not constant, they change from cycle to cycle.

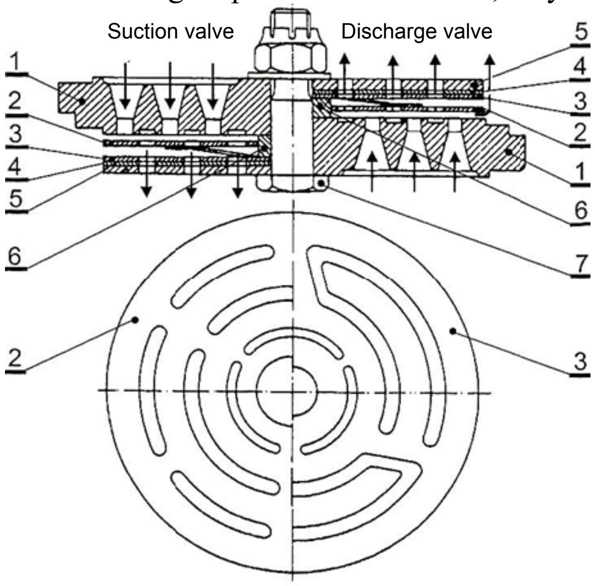

Fig. 2. Delivery and suction valves [2]: 1 – seat, 2 – plate, 3 – plate spring, 4 – damping plate, 5 – stop plate, 6 – guiding ring, 7 – bolt

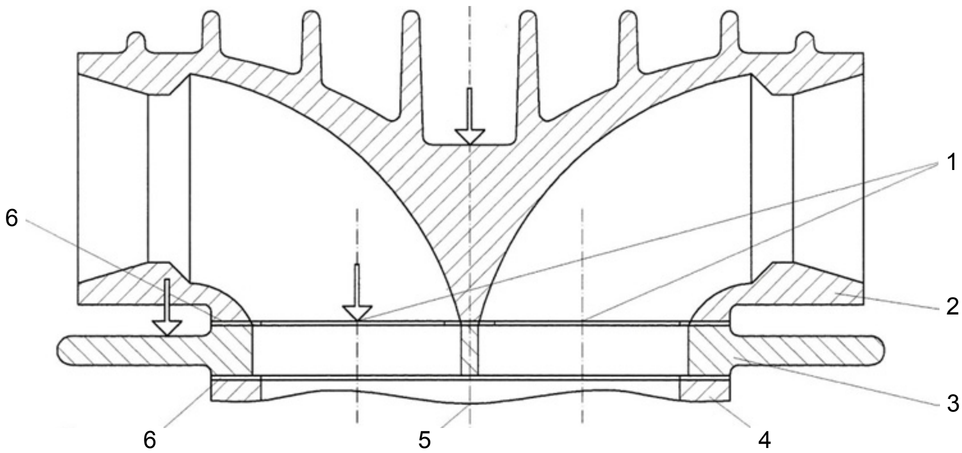

Fig. 3. A section across compressor components essential for vibration propagation (arrows show points where vibration transmitters were mounted): 1 – axes of valves, 2 – cylinder head, 3 – plate of the valve seat, 4 – cylinder block, 5 – axis of the piston movement, 6 – seal

Certain valve conditions were simulated in order to determine the influence of valve condition on the vibration signal. Then vibrations of the delivery valve bolt were measured. Tests were made without an external load of the compressor, i.e. with the cylinder cover removed and with low rotary speed. Simulated compressor conditions are given in Table 1. The peak-to-peak value of vibration was measured in a period much longer than one revolution. An example diagram is shown in Fig. 5.

The peak-to-peak values changing in time lead to a conclusion that after a time needed to stabilize the working conditions, the peak-to-peak value for simulated states reaches an average (constant value) and variable value Δ (difference between maximum and minimum values). Both values depend on the simulated condition of the valve. Table 2 contains simulated conditions and corresponding peak-to-peak values of vibration accelerations of the valve bolt.

Presented below in Table 3 are the peak-to-peak values of vibrations measured on the valve, valve plate and the cylinder head for one “normal” condition of the valve.

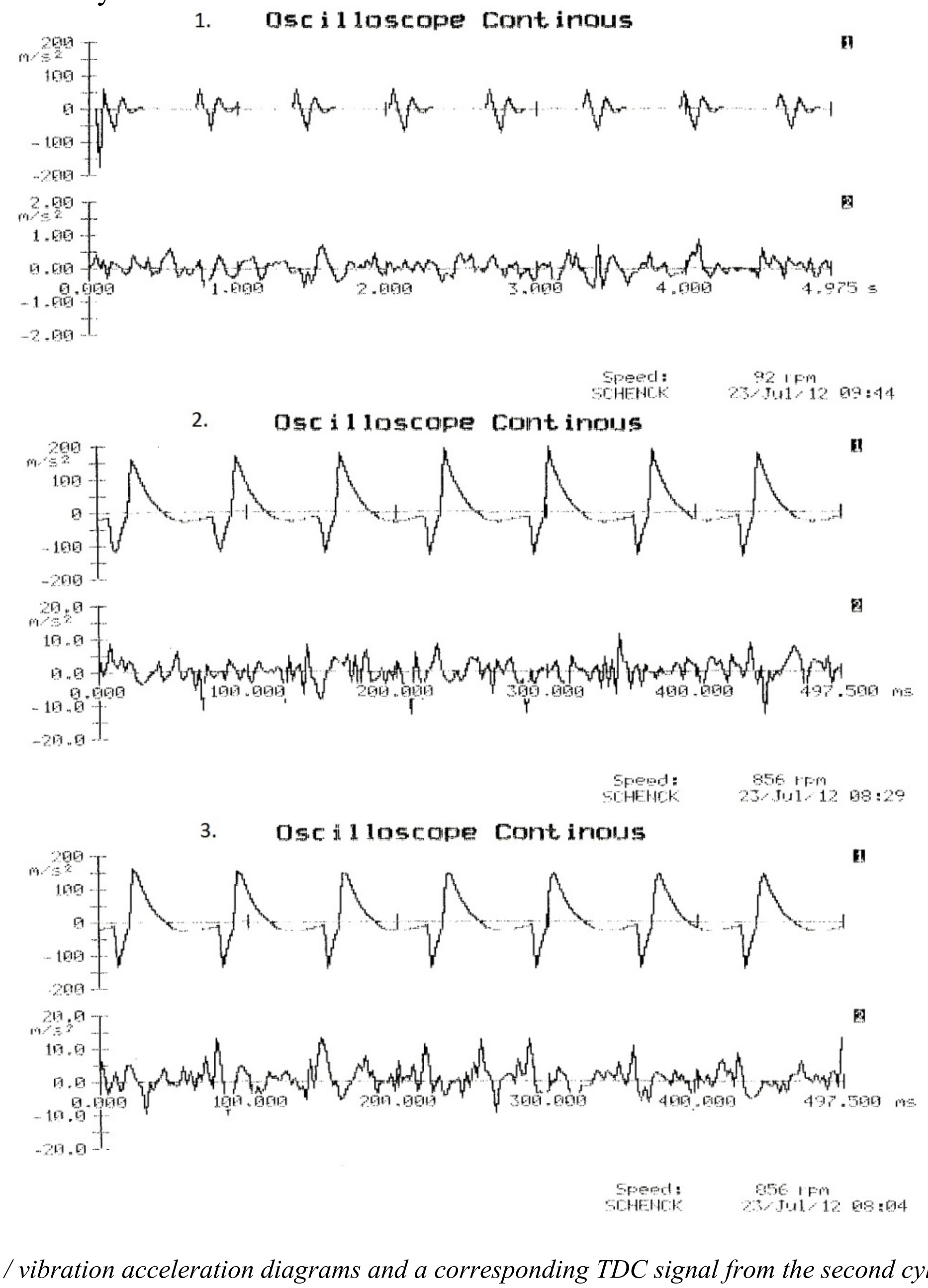

Fig. 4. Time / vibration acceleration diagrams and a corresponding TDC signal from the second cylinder, signal measured on: 1. valve bolt at 72 rpm and head removed, 2. valve plate at 856 rpm, 3. head at 856 rpm

Tab. 1. Measurements table

Item	Condition	Simulation
1	Normal	Valve is operational
2	Fracture A of valve plate	Radial “natural” fracture along 1/3 of the radius
3	Fracture B of valve plate	Lack of 1/3 external ring of valve plate
4	Fracture C of valve plate	Radial fracture along the whole radius length
5	High spring stiffness	3 springs
6	Low spring stiffness	1 spring
7	Suspension of valve plate	No valve plate
8	Spring wear	No spring
9	Backlash of two connected components	Loosened valve nut
10	Backlash in valve-plate connection	Loosened tightening ring

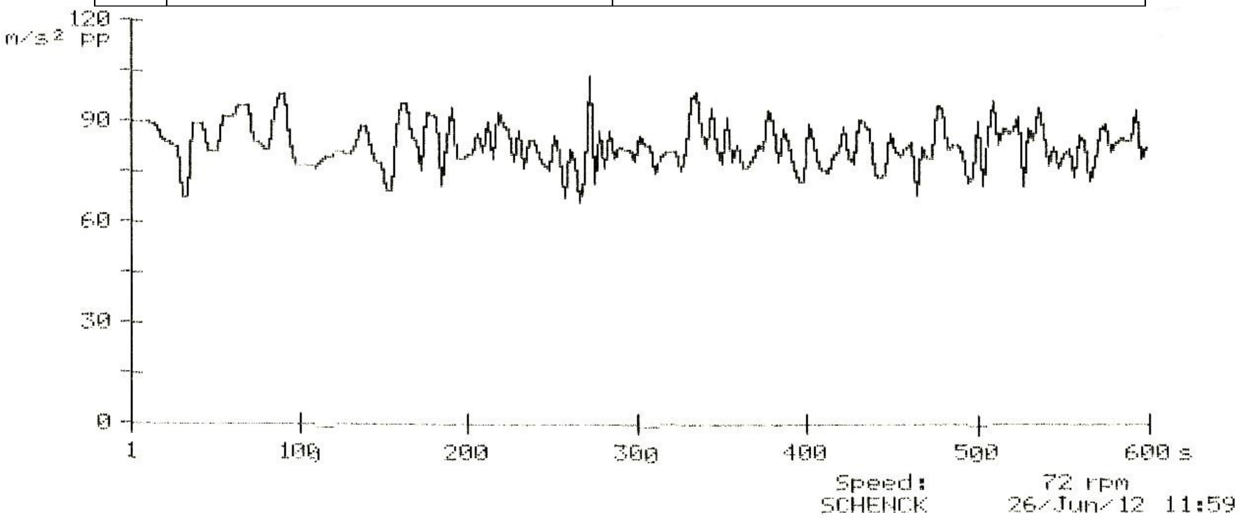

Fig. 5. Peak-to-peak values of vibration accelerations of an operational valve

Tab. 2. Measures of the time function of peak-to-peak values of vibration accelerations (valve bolt)

Condition	Constant value (mean) [m/s^2]	Δ [m/s^2]
Normal	79.2	38.7
Fracture A	61.6	37.5
Fracture B	4.15	(uncertain value)
Fracture C	55.6	34.4
High stiffness of spring	71.4	25.0
Low stiffness of spring	4.4	3.0
Suspension of valve plate	7.55	6.6
Spring wear	32.4	45.2
Backlash of two connected components	39.35	15.55
Backlash in valve-plate connection	69.975	25.7

Tab. 3. Peak-to-peak values of vibration accelerations generated by a valve working in the "normal" condition as a time function, for three measurement points

Measurement point:	Constant value (mean) [m/s^2]	Δ [m/s^2]
Valve bolt	79.2	38.7
Valve plate	11.4	10.0
Cylinder head	3.4	5.1

5. Conclusion

Simulated conditions of the valve tested consist in lower stiffness of the spring and lower tightness between the plate and t seat. Only a spring stiffness increase causes the valve lift to decrease. Lower spring stiffness leads to a drop in the mean peak-to-peak value. In the examined case − spring stiffness increase and related valve lift decrease − the decrease of mean peak-to-peak value turned out to be larger than the simultaneous increase caused by increased spring stiffness. Fractures in the valve plate cause a decrease in the mean peak-to-peak value. The causes may be as follows: decrease of the difference value of pressures on both sides of the valve and an increase of impact duration (due to lower plate stiffness). It seems that practically the wear of plate valves resulting in an essential increase of valve lift is unlikely. Therefore, it seems impossible that the vibration signal value will increase due to valve wear. In case of plate valves, one should expect that values of vibration measures will decrease depending on its technical condition. Additionally, these values depend on the measurement point: the signal weakens as the distance to a measuring point increases and if there are more indirect components in between.

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