APPLICATION OF VIBRATION DIAGNOSTICS IN EVALUATING INDUSTRIAL SAFETY OF COMPRESSORS

R. V. Zhukov, V. M. Davydov, V. V. Grib, M. V. Korolkov

OOO NPP Mechanic, Moscow, Russia

Moscow Automobile and Road Construction Institute (State Technical University), Moscow, Russia

The operation of hazardous production facilities requires special attention to safety issues. Under current conditions, expert evaluation of equipment is one of the ways of ensuring its safe operation. In the process of operation, wear of components and tribo-couplings, aging and corrosion of materials and accumulation of fatigue damages lead to a situation where parameters of technical condition of a machine change from original values set by a designer to the threshold values, which determine machine operability.

Dynamics of any mechanism is the source of vibration of the entire machine. Therefore, change of dynamics produces effect, first of all, on parameters of the vibration processes. Computer modeling of mechanical system behavior provides significant assistance in interpretation of vibro-acoustic signals and also in detection of defects during machine diagnostics.

A common technique used for solving a problem of changes occurring in a complex mechanical system envisages discretization of such system in time and space with the follow-on analysis of a number of sequential states of the system at preset short enough time intervals δt (operating intervals). It is assumed that the rate of change in the state of a system during this time interval depends on the system properties, operating conditions and state of the system in one of the terminal points of this time interval and remains unchanged during it.

The mechanism motion law is described by Lagrange equations, numerical solution of which is found through sequential analysis of a state of a system at short finite time intervals, which are significantly shorter than one cycle, or after small changes of a generalized coordinate. The motion law changes with the operating time and depends on extent of wear of kinematic pairs. Gaps in kinematic pairs, which size grows with the wear of tribo-couplings, lead to the increase of degrees of freedom, which is taken into account by the application of additional equations of motion of components in these gaps − the so called constraint equations. The solution of such problems is addressed in [1]. At the same time, this makes it possible to determine the impact load experienced by kinematic pairs, i.e. to assess loads tribo-couplings and components are exposed to.

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{q}_j} - \frac{\partial T}{\partial q_j} = Q_j + Q_j^R + Q_j^{mp},$$

$$(j=1,2,...,r) \qquad (1)$$

$$Q_j^R = \sum_{a=1}^{s} \lambda_a \frac{\partial f_a}{\partial q_j},$$

where q_j is generalized coordinates; $\dot{q}_j$ is generalized rates; Q_j is generalized active forces; Q_j^R is generalized reactive forces; Q_j^{mp} is generalized friction forces; T is kinetic energy of the system; λ_a is Lagrange multiplier featuring constraint reaction a; r is the number of generalized coordinates; s is the number of constraints imposed on the system.

Generalized reaction is determined by the constraint equations:

$$f_a(t,q_j,\dot{q}_j) \leq 0, \quad (a=1,2,3,...s) \qquad (2)$$

The system solution (1) was through the application of numerical methods along with the use of constraint equations (2). This allowed us to determine, during a mechanism motion cycle, all generalized coordinates featuring positions of mechanism links and components in the gap field with consideration for their possible displacement, their time derivatives and reactions of constraints with friction forces of respective assemblies. The nominal radial size of gaps in bearings and piston block are used as initial conditions.

Modeling of movable components displacement and impact processes allowed us to discriminate diagnostic characteristics in a vibro-acoustic signal for assessing state of corresponding assemblies. Some examples of compressors and pumps, being examined by OOO NPP Mechanic as part of expert evaluation efforts, contained in this report show how the results of vibration diagnostics helped to take appropriate measures to ensure their safe operation.

Examination of compressors used in the food industry shows that pitting of babbit in main and rod bearing liners occurs often enough in low speed machines. AGK-56 and AGK-47 two-stage horizontal ammonia compressors and German-made NA-20 one-stage horizontal hydrogen compressors installed at various facilities were considered to be typical machines of this type.

As an example, the report contains the results of examining the NA-20 hydrogen compressors installed at Moscow Fat Production Plant with the following main characteristics: rotational crankshaft speed of 290 rpm, electric motor power of 44 kW, output of 10.2 m^3/min, discharge pressure of 1 kgf/cm^2. Diagnostics of these machines continued for 2 years.

Prior to measuring vibration characteristics of the compressors, their dynamics under established operating modes and with consideration for the actual size of gaps in friction assemblies was modeled. Modeling showed that the ideal friction surface of bearings does not cause any impacts therein.

Diagnostic of the compressors revealed the following: typical diagnostic indicators of damage of rod-bearing liner were observed on the main bearing covers (caps). Visual examination performed after disassembling this unit confirmed that "diagnosis" (Fig. 1). After the first of the compressors was repaired, the vibration parameters have decreased significantly. The report shows time realizations and spectra before and after repair of the liner (Fig. 2). The spectral characteristics indicated presence of several harmonics multiple to frequency of about 285 Hz. According to the examination results this frequency features the state of a rod bearing.

After 5,146 hours of operation of the repaired compressor, a decrease in the vibration level was observed in time realizations. Sub-harmonics almost disappeared in the envelope spectrum. At 285 Hz, the level of vibration decreased 1.8-fold. The 1st harmonic vibration level also decreased 1.7-fold. All this indicated that the components were aligned. No damage of the liner and seating-in signs on the shaft were detected during the next scheduled repair of the assemblies.

Operating conditions, specifically a discharge pressure of the compressor stage, also produce a considerable effect on vibration characteristics.
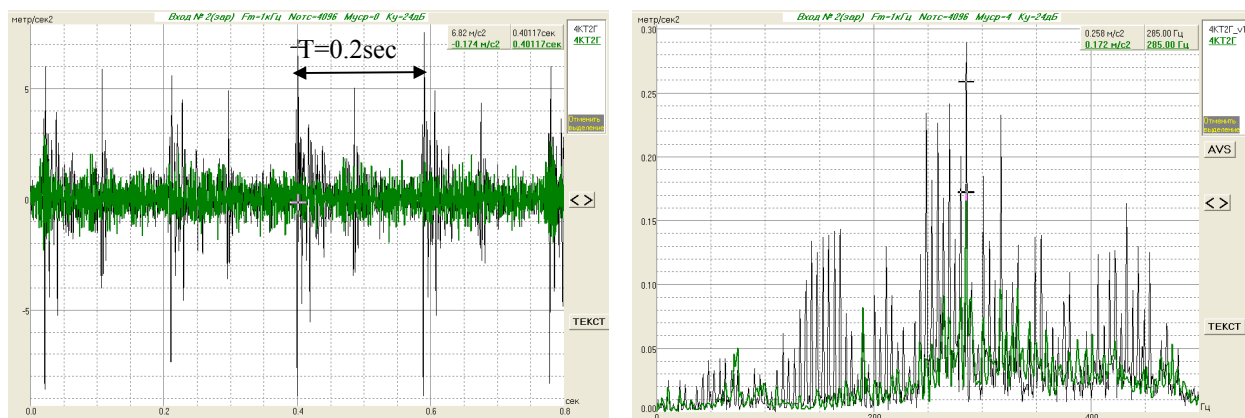

Fig.1 Defect of Half-Liner of Rod Bearing of NA-20 Compressor, Position No.35-II

Fig. 2 Time Realizations and Vibration Acceleration Spectra of NA-20 Compressor, Position No.35-II (measured when the compressor ran idle):

—————— before repair of the bearing; —————— after repair of the bearing

Below are some results of examination of a TV2S3K-400/630 nitrogen compressor installed at Azot Chemical Plant in the large-scale separation of residual gases resulting from ammonia synthesis. The compressor's specifications are as follows: output of 110 m³/min, final discharge pressure of 10 MPa, motor power of 800 kW, rotational shaft speed of 293 rpm. In the course of the machine overhaul, gaps were examined in all bearings, crosshead slide assembly and cylinder-piston block. In the course of examination of the compressor during the entire repair cycle (3 years) being recorded were vibration spectra, which indicated various levels of wear of the assemblies. The report shows spectral characteristics of vibration for the assemblies under consideration (Fig. 3).
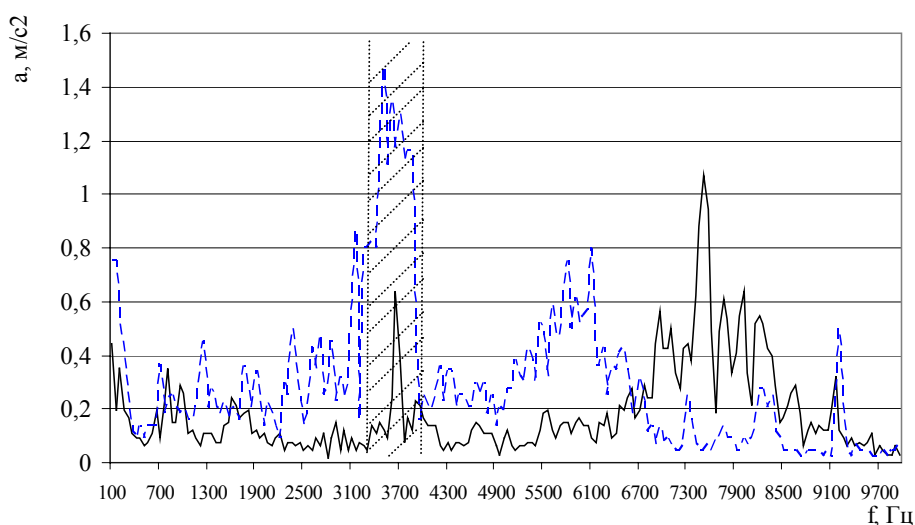

Fig.3 Spectra of Vibration Acceleration of Cylinder-Piston Block of TV2S3K–400/630 Compressor

—————— – 0.6 mm gap in cylinder-piston block;
- - - - - - – 2 mm gap in cylinder-piston block

The results of evaluation of industrial safety of screw and centrifugal compressors also include some interesting examples when the results of vibration diagnostics were compared with actual defects detected on the rotor contact surfaces during visual examination after machines were disassembled.

It should be noted that the existing regulatory documents do not fully reflect specificities of piston compressors related to vibration rating. Due to shuttling movement of components and associated cyclical piston forces and low rotation frequencies, which are often lower than 600 rpm,

the threshold values of vibration parameters should be higher than those set for rotor machines. The experts of OOO NPP Mechanic used the results of vibration diagnostics, the results of compressor fault detection and the recommendations of ISO 10816-6:1995 to develop norms of root-mean-square values for vibration velocity and vibration displacement. These norms are included in the Piston Compressor Diagnostics Methodology approved by the Federal Service for Environmental, Industrial and Nuclear Oversight (Rostechnadzor).

Specialists of the expert team of OOO NPP Mechanic collect data for the purpose of the follow-on rating of general level of vibration of piston compressors. These efforts included examination of about 100 refrigerating piston compressors of various types. Expert examination and assessment of the said machines allowed us to accumulate a database containing information on vibration characteristics and technical conditions of the machines.

Statistical processing of the recorded levels of vibration was in compliance with the requirements set forth in [2]. Permissible values of vibration intensity for various state zones were calculated with the application of the following formula:

$$U = \overline{A} + k \cdot \sigma, \tag{3}$$

where $\overline{A}$ is the arithmetic mean value of tested parameter; σ is the root-mean-square deviation of tested parameter; k is the figure indicating probability of parameter value being below limit value (for various state zones value of k ranges from 0 to 2).

- The examined compressors were grouped in terms of drive power, number of cylinders and rotational shaft speed.

Pearson criterion was used for selecting a distribution law, which the "sampled" vibration levels adhered to. Calculations showed that for all three classes of compressors value of actual criterion for RMS value of vibration velocity and vibration displacement is lower than the critical value ($\chi^2_{actual} < \chi^2_{critical}$). This fact indicates that the used "sampled" vibration levels adhere to the normal distribution law.

The obtained permissible limit values of vibration are shown in Table 1.

Table 1

Ve, mm/s	Se, µm	Evaluated Zones*		
		Class 1	Class 2	Class 3
1.1	18	A		
1.8	28	A		
2.8	45		A	
4.5	71	B		A
7.1	113	C	B	
11.2	140	D	C	B
18	220		D	C
28	283			D

* A is vibration of a newly commissioned machine; B is machines with vibration within this zone can operate for a long period of time; C is machines with vibration within this zone can operate for a limited period of time; D is vibration within this zone is intensive enough to cause damage to a machine

Comparison of the obtained results with the norms specified in the instruction [3] showed that "D" zone vibration levels of Class 1 compressors are within the limits specified in [3]. Apparently, it is explained by the effect of relatively low dynamic forces, which are determined, *inter alia*, by

inertial loads produced by motion mechanism on compressor crankcase block. As for Class 2 and 3 compressors, values of "D" zone vibration levels are 4 dB higher than those prescribed by the requirements set forth in [3]. In practice, the state of such compressors is often in the "C" zone, which indicates presence of significant defects and need for their shutdown for repair in near future. However, operational practice shows that they can operate properly for a long period of time.

Special attention should be paid to expert evaluation of compressor plants, in which a pressure tank is used as a support frame. Typically, such design is a screw type (less frequently, a piston type) refrigerating ammonia compressors and also in mobile piston air compressors. The tank experiences high cyclical loads induced by vibration of the compressor and the motor. High inertial forces cause cyclical deformation of the tank in sections where supports are welded, which facilitates occurrence of cracks in these places and leads to weakening of the motor-compressor system stiffness. This, in its turn, leads to quick misalignment of shaft axes and premature failure of driver antifriction bearings. Experience of examination of compressors mounted on tanks shows that in the course of technical diagnostics one should use a program (methodology) that takes into account all aspects and specificities of the system operation. Additional inspection of the shell section to which compressor and motor support feet are welded is required. It is also necessary to systematically monitor the vibration state of a compressor.

The report shows the instances where cracks occurred in places where a compressor unit frame was welded to a tank. As an example, Fig. 4 shows a system incorporating 2P-110 compressor. There also were some cases when fatigue cracks occurred in the similarly designed systems incorporating screw ammonia compressors.

Fig. 4 Compressor Unit with Crack on Tank

Typically, companies pay little attention to maintenance and examination of technical condition of mobile compressor units. They are designed such that receiver (condenser, oil separator) is also used as a support frame of the compressor and the motor. The above mentioned problem led to the tragic accident at the UKP-1/10 mobile air compressor unit. The unit consisted of a two-stage compressor with an 8 kW motor mounted on 250l air receiver (tank) with a pressure of 8 kgf/cm^2.

Due to a combination of operating parameters of the receiver, it was subject to the requirements of the PB 03-576-03 Safety Regulations [4]. The company, which operated the

compressor unit, did not perform regular examination of the tank, had no safety relief valve operability inspection log, and, as it was found out, no examination/inspection of the valves was performed. Pressure switch was set to off since it was faulty. The design of the air outlet envisaged installation of a valve. At the time the compressor was turn on, this valve was closed, pressure and temperature exceeded the permissible limits, the pressure switch failed, the air-relief valves did not open, all of which resulted in the explosion which destroyed the receiver.

OOO NPP Mechanic experts took part in investigation of the accident root-causes. In the process of examination, the following was found out. The receiver body was destroyed mainly along the welded joints, and the receiver shell plate was torn and rolled outside (Fig. 5). The fact that all welded joints were destroyed precisely along the thermal effect zones indicates high enough quality of welding work performed on the tank. Macro-examination of the fracture surfaces did not reveal any major defects in the joints and confirmed that they had the fine-grained structure. Absence of plastically deformed metal on fracture edges definitely indicated a brittle nature of fracture. The drive end of the compressor crankshaft was cut at the fillet connection point. Macro-examination of the cut section of the crankshaft showed that it had the structure typical for a brittle facture. Large quantity of oil crust was detected in the valve chamber from the I stage side as well as on the inner surface of the discharge pipe of the I stage. The paint and lacquer coating of all pipes was burned, which indicated that they had been exposed to high temperature. The back-flow valve installed on the discharge pipe of the II stage was destroyed. The pressure-relief valves of the I and II stages were torn off the sockolets.

Fig. 5 Compressor Unit after Destruction

The practical operation of air compressors [5] shows that there is a hazard of ignition and explosion of oil crust accumulated in a cylinder and pipes. It may happen if a discharge temperature exceeds the values specified in the "technical passport'. In this particular case, it could occur when the compressor unit was operating with almost closed valve at the receiver air outlet. Thermodynamic calculations of the compressor showed that the discharge temperature of +190°C, at which self-ignition of oil crust occurs, was reached in the I stage cylinder when the final pressure was 14 kgf/cm^2. And the motor was powerful enough to enable the compressor to produce such pressure.

Calculations of strength of the receiver's shell (at 14 kgf/cm^2), along welded joints of which fracture occurred—with allowance for inner pressure, the minimum wall thickness and degradation of mechanical properties of metal heated up to +200°C—showed that strength of the tank was

ensured. At 14 kgf/cm^2, the receiver could not be destroyed without being affected by other factors. The tank completely lost its strength at a pressure of not less than 26 kgf/cm^2.

Therefore, destruction of the air receiver resulted from ignition of oil crust in the cylinder of the first stage of the compressor and the follow-on explosion of oil mist in the receiver.

Thus, in the case of these systems, it is necessary to evaluate industrial safety of both the compressor and tank as a whole.

References:

1. V. V. Grib, R. V. Zhukov, I. M. Petrova, I. A. Buyanivsky. Diagnostic Models of Changes in Technical Condition of Mechanical Systems / Under the general editorship of V. V. Grib / MADI (STU) – Moscow, 2007 – p. 300

2. RD 50-638-87 Guideline Document. Methodological Guidelines. Vibration. Procedure for Development and Regulations for Rating of Vibration of Machines and Equipment – Moscow: Standards Publishing House, 1987 – p. 16

3. RD 09-244-98 [RDI 09-513(244)-02] Guideline Document. Instruction for Technical Diagnostic of Tanks, Pipes and Compressors of Industrial Refrigerating Ammonia Plants. Moscow: Industrial Safety Research Center, 2004 – p. 100

4. PB 03-576-03 Safety Regulations. Regulations for Design and Safe Operation of Pressure Vessels. Moscow: Research Center for Industrial Safety under Gostechnadzor of Russia 2003 – p. 192

5. A. A. Tikhomirov. Ensuring Safe Operation of Equipment of Compressor Plants Incorporating Piston Air Compressors // Industrial Labor Safety. 2003 No .6 pp. 19-24