MEASUREMENT AND EVALUATION OF MECHANICAL VIBRATION OF RECIPROCATING MACHINES

Piotr Jan BIELAWSKI

Maritime University of Szczecin, Diagnostics and Machine Repairs Department
70-205 Szczecin, ul. Podgórna 51/53, Poland, e-mail: p.bielawski@am.szczecin.pl

Summary

Mechanical vibration is used in condition monitoring and diagnostics of machines. Measurement and evaluation of vibration depends on machine type and machine part. ISO standards describe evaluation of machine vibration by measurements on non-rotating parts and on rotating shafts of non-reciprocating machines. For condition monitoring and diagnostic purposes measurements and evaluation of rotating shafts vibration of reciprocating machines have to be carried out. This paper describes: types of rotating shaft vibration of reciprocating machine; longitudinal vibration and torsional vibration of crankshaft as diagnostic signal; measurement of displacement axial (longitudinal) vibration of crankshaft free end; measurement of acceleration axial vibration of crankshaft free end; measurement of acceleration torsional vibration (angular acceleration) of crankshaft free end; evaluation of torsional and axial vibration of crankshaft free end. Courses of angular accelerations, axial displacement and axial acceleration of crankshaft free end will be presented and discussed.

Key words: reciprocating machines, rotating shafts vibration, longitudinal vibration, torsional vibration.

POMIARY I OCENA DRGAŃ MECHANICZNYCH MASZYN TŁOKOWYCH

Streszczenie

Drgania mechaniczne elementów są wykorzystywane w dozorowaniu i diagnozowaniu maszyn. Pomiary i ocena drgań zależą od typu maszyny i części maszyny. Normy ISO opisują ocenę drgań mierzonych na niewirujących elementach maszyn i drgań mierzonych na wirujących wałach maszyn nie będących maszynami tłokowymi. Celem rozwoju dozorowania i diagnozowania maszyn tłokowych dokonano pomiarów i oceny drgań wirującego wału korbowego maszyny tłokowej. W artykule przedstawiono: rodzaje drgań wirujących wałów maszyn tłokowych; drgania wzdłużne i skrętne wału korbowego jako sygnały diagnostyczne; pomiary drogi drgań wzdłużnych wolnego końca wału korbowego; pomiary przyspieszenia drgań wzdłużnych wolnego końca wału; pomiary przyspieszenia drgań skrętnych (przyspieszenia kątowego) wolnego końca wału; ocenę drgań skrętnych i wzdłużnych wolnego końca wału. Zaprezentowano i przeanalizowano przebiegi drgań wzdłużnych i skrętnych wolnego końca wału.

Słowa kluczowe: maszyny tłokowe, drgania wirującego wału, drgania wzdłużne, drgania skrętne.

INTRODUCTION

Compared to rotor machines, reciprocating machines are of more complex construction, as they have gear changing the reciprocating motion into the rotating motion of the crankshaft. A fault of one element of the piston-crank unit makes the whole machine faulty. Such faults often result in machine failure.

Machine failures and component faults are prevented by condition monitoring and diagnosing of machines, where vibration signals are important signals appropriately utilized. The methodology of condition monitoring and diagnostics is subject to standardization – international ISO standards have been and are being developed, and on their basis national ISO-compliant standards are established.

This article analyzes published international standards in view of their use in condition monitoring and diagnostics of reciprocating machines. Possibilities of developing methods for diagnosing these machines are indicated.

1. CONDITION MONITORING AND DIAGNOSING OF MACHINES IN THE LIGHT OF ISO STANDARDS

In [22] Kolerus overviews the up-to-date standards relating to vibration diagnostics of machines. The relevant standards are divided into those related to condition monitoring during machine operation and standards oriented towards diagnosing. It follows from the grouped standards that monitoring-directed standards are those object-oriented.

There are separate standards for reciprocating and non-reciprocating machines as objects of diagnostics. For the two types of machines standards are specified for measurements of mechanical vibration on non-rotating parts and on rotating shafts. Due to the reference point, measured vibration is divided into vibration measured without a fixed reference point (absolute vibration) and vibration measured in relation to a specific reference point (relative vibration).

The standard ISO 10816 Part 1–7 refers to the evaluation of vibration measured on reciprocating and non-reciprocating machines. The evaluation of vibration measured on non-rotating parts depends on the machine type, use, dimensions and operating conditions [2, 3, 4, 5, 6, 7, 8, 19, 20]. Evaluation refers to absolute vibration values measured in places specified by the relevant standard. The RMS value of vibration is measured in this case.

Standards ISO 7919 Part 1–5 refer to the evaluation of vibration measured on rotating parts of non-reciprocating machines [11, 12, 13, 16, 18]. Shaft lateral vibration is evaluated by measurements in defined points: at bearings or in their direct vicinity. The standard determines the method of measurement of absolute and relative vibration.

The standard ISO 22266-1 refers to the evaluation of vibration measured on rotating parts of some turbines operating in specified conditions. Torsional vibration is evaluated [12].

2. VIBRATION OF ROTORS

Rotating elements are termed "rotors" [11, 21]. Rotors are modeled by replacing masses and elasticity of the rotor with masses in the form of disks with equivalent masses connected by massless springs or massless shafts with equivalent rigidity. Such models have vibration nodes and arrows. The number of nodes is related to the frequency of rotor free vibration. In vibration nodes the material stresses reach maximum values and vibration amplitudes of a given form equal zero. In the arrows vibration amplitudes reach maximum values and the stress equals zero. Rotors can have lateral, torsional and longitudinal vibration. Stresses caused by vibration add up with working stresses and may lead to material damage of the rotor shaft. In many machines rotor vibration has to be monitored.

The standard ISO 7919 describes measurement of lateral vibration. Longitudinal vibration can be measured, similarly to lateral vibration, by non-contact eddy current sensors. In the case of non-reciprocating machines longitudinal vibration is negligibly small. Torsional vibration of rotors in operation observed outside vibration nodes causes changes in rotor angular velocity in a given plane of observation. Measures of torsional vibration can be as follows: stresses at the nodes, deformation of a shaft section, changes in angular velocity of the shaft. The results depend on the position of the measurement plane / section on the rotor shaft axis.

"In general torsional vibration is more difficult to measure than lateral vibration. Torsional response – both strains and motions – can be measured at intermediate points in a system. Strain gauges are available in a variety of sizes and sensitivities and can be placed almost anywhere on a shaft. They can be calibrated to indicate instantaneous torque by using static torque loads on drive shafts. If calibration is not possible, stresses and torques can be calculated from strength of materials theory. Strain gauges are usually mounted at 45° angles so that shaft bending does not influence torque measurements. The signal must be processed by a bridge-amplifier unit that can be arranged to compensate for temperature. Because strain gauge signals are difficult to take from a rotating shaft, such techniques are not common diagnostic tools. Slip rings can be used to obtain a vibration signal from a shaft. Wireless telemetry is also available. A small transmitter mounted on the rotating shaft at a convenient location broadcasts a signal to a nearby receiver. Commercial torque transducers are available for torsional measurement. However, they must be inserted in the drive line and thus may change the dynamic characteristics of the system. If the natural frequency of the system is changed, the vibration response will not accurately reflect the properties of the system" [3].

"The general method of determining the rotational speed is to use some form of tachometer or shaft encoder. Common ways to measure torsional vibration angular velocity oscillations are by means of toothed wheels or gears and magnetic pickups (a fixed sensor). "The signal generated by the teeth of the wheel passing the fixed sensor has a frequency equal to the number of teeth multiplied by shaft speed. If the shaft is undergoing torsional vibration, the carrier frequency will exhibit frequency modulation (change in frequency) because the time required for each tooth to pass the fixed pickup varies. This is a very rugged and reliable measurement approach and is suitable for long-term monitoring of turbomachinery when required. Other approaches involve optical methods using grids or stripes on the shaft as the target. Sometimes the stripes or grid patterns are etched on a tape that is stuck to the shaft. In such cases care must be exercised to ensure that there is no large optical discontinuity where the ends of the tape butt together. Optical methods involving lasers and the Doppler principle are sometimes used as well" [3].

3. VIBRATION OF RECIPROCATING MACHINE ROTORS

Rotors of most reciprocating machines consist of pistons, connecting rods and a crankshaft. Crankshafts are capable of producing significantly large lateral, torsional and longitudinal vibration.

Evaluation of that vibration has not been standardized. The difficulties lie in a multitude of types and models of reciprocating machines and inter-relations between various types of vibration. Some authors indicate coupled torsional and longitudinal vibration of the crankshaft.

Rotors of reciprocating machines are modeled similarly to those of rotating machines. Contrary to rotor machines, mass moments of inertia of the model disks are not constant, but depend on the rotation angle of the crankshaft. Angular velocity as a function of rotation angle is not constant – in other words the machine does not run uniformly. A major difficulty is that the journal axes during vibration do not remain parallel to the axes of the supporting sleeves.

The place and method for measurements of rotor vibration in reciprocating machines depend on the purpose of the tests. It follows from [2] that the crankshaft free end is a convenient location of measurement, and that lateral and longitudinal vibration is useful for diagnostic purposes.

At the crankshaft free end the most significant forms of all types of vibration achieve maximum values. While a crankshaft rotates, the centre of the free end journal does not overlap with the rotating axis. The free end journal centre moves along a roughly elliptical trajectory. The free end normally does not stand out of the bearing supporting the last main journal, but it can be prolonged and brought outside the machine.

Measurements of axial vibration of the crankshaft free end, like lateral vibration [13], can be performed as contact (absolute vibration) or non-contact (relative vibration) measurements. In both cases the sensor axis should be co-axial with the shaft journal axis, the front plane of the journal cannot have shape errors and must be perpendicular to the axis. In non-contact measurements, e.g. by eddy current sensors aligned with the shaft journal, the effect of non-perpendicularity of the shaft end front can be negligibly small. Deflection of a crankshaft, typical of operating reciprocating machines, causes the position of journal axis to change. For this reason sensors of longitudinal vibration of reciprocating machines have to set themselves co-axially in relation to the shaft or compensate for the existing misalignment. The first scenario is realistic for contact sensors, that touch the journal front through a slide plate. Then the problem to be solved is to properly press the sensor to the journal and provide for appropriate conditions for the plate to slide on the shaft front end. The effect of this method may be additional casual vibration. The other scenario is real for non-contact measurements by using a pair of sensors set so that shape and position errors will produce signals of the same values and opposite signs in each sensor.

Practically, the only accessible place on a crankshaft suitable for measurements of torsional vibration (without disturbing the machine integrity) is its free end. Torsional vibration measurements in this place started as long ago as 1912 when the Sulzer company introduced a torsiograph, later named Geiger torsiograph after Dr Josef Geiger. A detailed description of torsiograph can be found in J. Geiger's: *Der Torsiograph, ein neues Instrument zur Untersuchung von Wellen. ZVDI 60 (1916) 40 und 42* [22]. The Geiger torsiograph works according to the principle of seismic mass (like contact transducers of absolute vibration velocity and accelerations). The characteristic feature of this type of torsiograph is that inside a drum directly mounted on the free end or a drum driven by a belt transmission there is a mass with high moment of inertia elastically coupled with the drum. The drum generates vibration similar to that of the free end, while the mass, due to the large mass moment of inertia, rotates inside the drum at a constant angular velocity. The relative motion between the drum and mass is measured: the angle of torsional vibration. Torsiographs mounted directly on the shaft free end require an expanded system of signal transmission to peripheral devices.

The transducer that could perform non-contact measurements of torsional vibration accelerations seems to be the one working on the Ferrari principle [1]. In this case, however, the problem lies in undesired relative motions occurring in the machine between the measuring disk mounted on the shaft and the transducer mounted on the machine body. These motions generate measurement errors and can even lead to transducer damage, due to a little width of measuring gap.

4. TESTING OF ROTOR VIBRATION OF RECIPROCATING MACHINES

Tests were made to examine how axial run-out of the measuring disk mounted on a dual mass rotor influences transducer angular accelerations. The transducer and the run-out sensor were mounted on the rotor machine body. Tests were carried out for a constant rotating speed, far from the resonance speed of rotor vibration. The values of measuring disk deviations and the corresponding angular accelerations are shown in Fig. 1.

Torsional and longitudinal vibration of a single-crank, three cylinder air compressor were evaluated at its small load. The compressor only discharged air through a pipeline fitted with an oil separator, cooler, filter and silencer. Torsional vibration was tested by means of a non-contact angular acceleration transducer, while longitudinal vibration by means of a non-contact distance sensor and a contact linear acceleration transducer.

The angular acceleration transducer was mounted on the shaft in relation to the compressor body in such a way that the transducer could follow all motions of the shaft except rotation. The measuring disk run-out against the transducer was measured as well as the run-out of the free end against the compressor body. The axial run-out of the measuring disk (close to the disk edge) relative to the transducer was 0.12 mm. The radial run-out of the shaft free end relative to the compressor body was: vertically – 0.92 mm, horizontally – 0.38 mm.

immovable in relation to the compressor body. A synchronically averaged course of longitudinal vibration distance is shown in Fig. 3.

A piezoelectric linear vibration acceleration transducer was aligned with the shaft free end axis

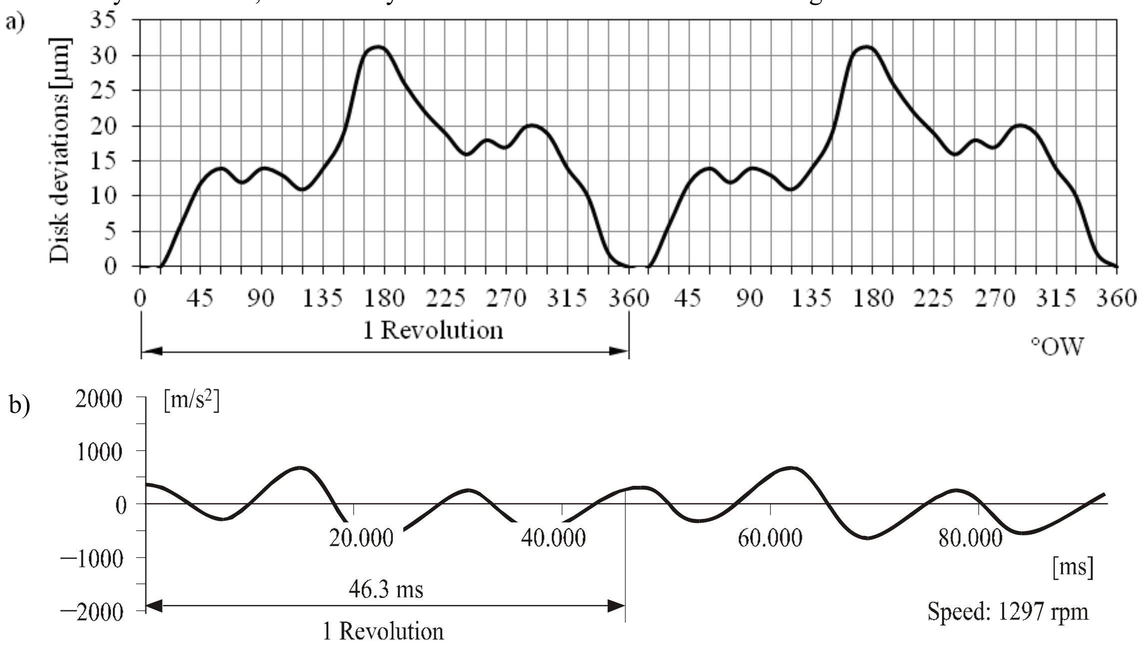

Fig. 1. Values of measuring disk deviations (a) and the corresponding angular accelerations (b)

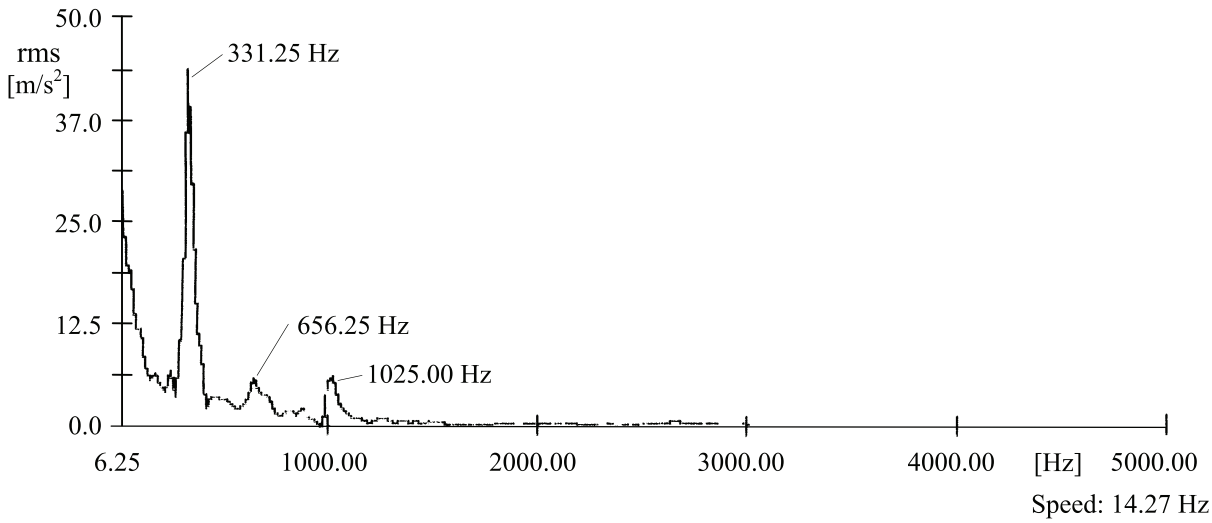

Fig. 2. Spectra of torsional vibration: 0–5000 Hz and 0–100 Hz

It can be assumed that there were two causes of shaft run-out: free end journal shape deviations and shaft deformations (deflection) caused by the weight of the compressor flywheel and the tension force of compressor drive vee-belts. In the torsional vibration measurements the data were synchronically averaged in the time and frequency domains. Spectra of torsional vibration are shown in Fig. 2. The timing of vibration and the course of position marker for cylinder No. 2 with the piston in the top dead centre are given in Fig. 3.

An eddy current sensor of longitudinal vibration distance was placed on the free end journal axis, and pressed through a carbon plate to the shaft front. The transducer handle, flexibly pressing the transducer to the shaft front, was rigidly attached to the compressor body. A synchronically averaged course of longitudinal vibration accelerations is shown in Fig. 3.

A significant influence of vibration from other sources on the measured vibration, torsional vibration in particular, was observed during the tests. Besides, it was noticed that the way the acceleration transducer is pressed to the shaft front is also important.

5. SUMMARY

There are no standards for the evaluation of rotor vibration of reciprocating machines. The establishment of such standards may significantly assist in condition monitoring and diagnostics of reciprocating machines.

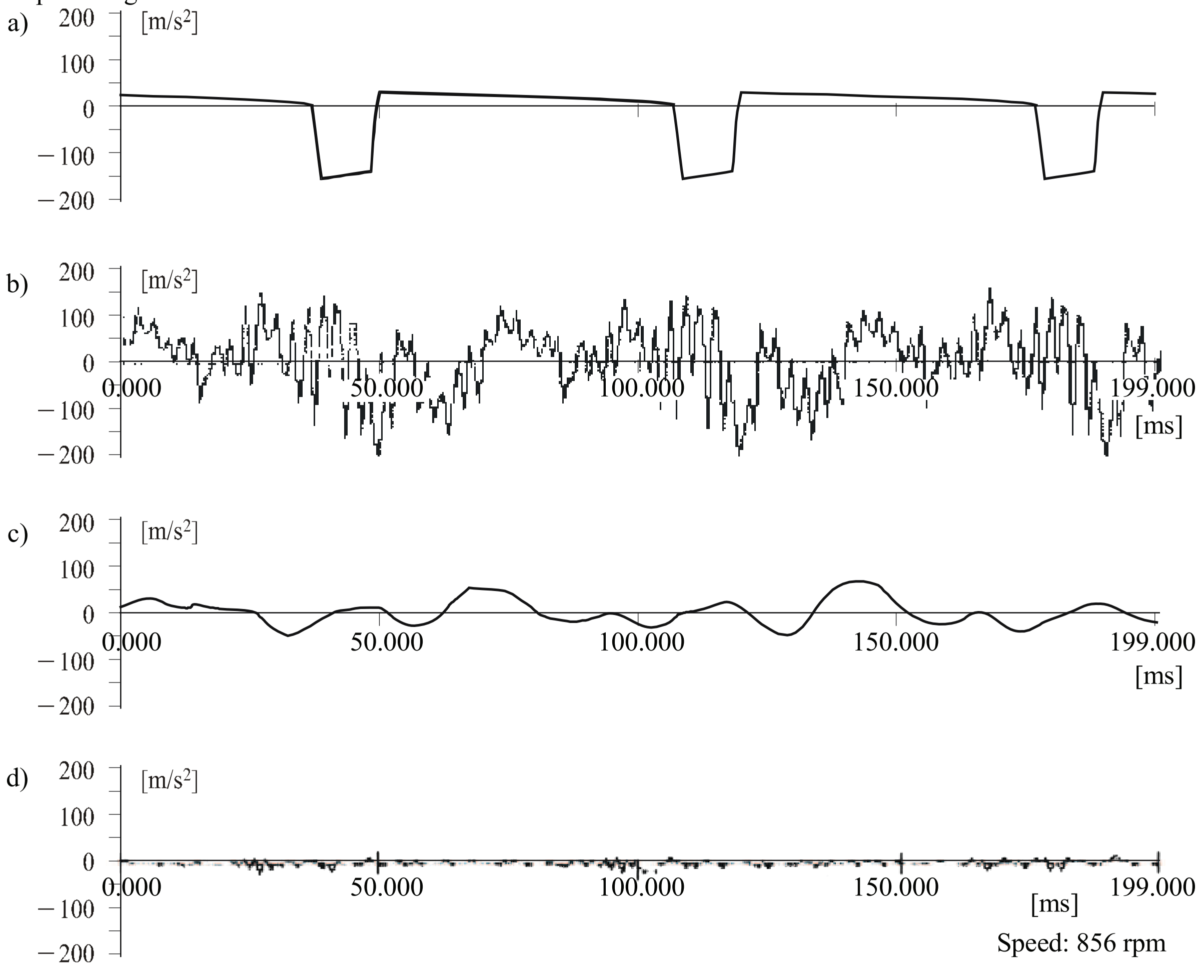

Fig. 3. The timing of vibration: a) the course of position marker for cylinder No. 2 with the piston the top dead centre, b) a synchronically averaged course of torsional vibration accelerations, c) a synchronically averaged course of longitudinal vibration distance, d) a synchronically averaged course of longitudinal vibration accelerations

Acceleration transducers working on the Ferrari principle are also sensitive to the motion transverse to the principal rotating motion of the measuring disk. Transverse motion may be caused by lateral and longitudinal vibration and by deviations of shape and position of the measuring disk and rotor shaft. One can see from Fig. 1 that the impact of measuring disk axial run-out on angular accelerations is unique and measurable.

It is possible to measure torsional vibration accelerations of reciprocating machines using a non-contact acceleration transducer. The torsional vibration spectrum from Fig. 2 is a spectrum typical of reciprocating machines. It consists of two parts:

– high frequency range dependent on the frequency value of rotor free vibration and the number of torsional vibration forms,
– low frequency range connected with non-uniform run (varying rpm) for this type of machines.

The timings of torsional vibration are correlated with those of longitudinal vibration, Fig. 3.

There are no difficulties to perform non-contact measurement of longitudinal vibration distance of the shaft free end. The course of this distance, presented in Fig. 3, satisfies the expectations, although it contains components resulting from the shaft front axial run-out and shaft lateral vibration. The measurement method has to be properly modified.

Contact measurement of longitudinal vibration accelerations of reciprocating machine rotors is difficult due to the shaft free end run-out and rotor lateral vibration. An additional problem to be solved is accurate transmission of vibration from the shaft onto the transducer via a slide plate. It follows from Fig. 3 that despite a high level of disturbances, the acceleration signal appears at places typical of these machines and after an improvement of the measuring method it can be useful in condition monitoring and diagnostics.

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Piotr Jan BIELAWSKI, a professor in the Condition Monitoring and Maintenance of Machinery Department of Maritime University of Szczecin. His researches are focused on Maintenance of Ship Machines: Identification of Elements, Assembly of Machines, Repairs of Machines and Machine Elements, Diagnostic of Machines particularly Diagnostic of Reciprocating Machines.