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ABSTRACT

The measurement of the kinematics of reciprocating compressors, such as position, speed and acceleration, is an important part of a wide range of tests. Examples of these tests are the determination of the indicator diagram, which depends on the measurement of the piston position; the determination of the Coefficient of Performance (COP), which requires the measurement of angular speed; and friction loss experiments, which require the rotational speed and the deceleration of the compressor. Due to their importance for research and development of reciprocating compressors, this paper presents a review of the kinematic measuring methods for reciprocating compressors. Additionally, a test rig was designed to perform static and dynamic measurements of the piston and crankshaft kinematics. The applicability of a linear variable inductance transducer in a reciprocating compressor was evaluated by comparing its response to the measurements obtained from an encoder.

1. INTRODUCTION

Reciprocating compressors are subjected to numerous tests during different phases of their development. Many of these tests require information about the kinematics of the compressor, such as position, speed and acceleration. For instance, during energetic performance tests, described in ASHRAE 23 (ASHRAE, 2005) and AHRI Standard 540 (AHRI, 2015), the rotational speed of the compressor must not deviate more than ±1% from its reference value, because it has strong impact on the cooling capacity and power consumption of the compressor. Another test that requires the compressor kinematics to be measured is the one used to obtain the indicator diagram (also known as pressure-volume diagram, or pV diagram). This test is commonly used by compressor manufacturers to determine the work applied to the refrigerant and the thermodynamic inefficiencies. The determination of the volume of the compression chamber requires the measurement of the crankshaft position (Kim & Soedel, 1986), which is converted to piston position using the corresponding kinematic relationship, or a direct measurement of the piston position. Oh, Lee and Lee (1994) presented an experimental method to determine the power dissipated by friction from the deceleration of the compressor. The rotational speed and deceleration were measured with an accelerometer placed on the compressor shell. When the compressor is switched off, the compressor decelerates due to friction. Thus, by measuring the deceleration it is possible to determine the friction losses. Nagata et al. (2012) used a torque transducer to measure the friction torque, which was multiplied by the rotational speed to obtain the friction loss in the axial bearing.

Compressor kinematics can be either measured directly from the crankshaft or piston, or indirectly through externally measurable quantities. As described in Demay, et. al (2012), the compressor average rotational speed can be obtained from the supply current by identifying the frequency component that is equal to the motor slip using mathematical tools, such as interpolated FFT, Chirp-Z transform and Hilbert transform. Schantz, Remscrim, & Leeb (2010) presents a method that provides an estimate of the instantaneous rotational speed using electrical measurements. Similarly, Zhe, et. al (2018) proposed a measurement method based on supply current fluctuations due to the rotor roundness error. Another indirect method relies on the pressure pulsation inside the compression chamber. Time intervals between two maximum and minimum pressure points can be used to determine the average value of the compressor rotational speed (Demay, Flesch, Flesch, & Penz, 2012).

The main advantage of indirect measuring methods is that they do not interfere in the operation of the compressor. However, direct measuring methods are considered more reliable, are a one-step solution and are needed to validate the indirect measuring methods (Battley, 1995). Schantz, Remscrim, & Leeb (2010) compares an indirect method for estimating the instantaneous rotational speed with measurements obtained from an encoder.

For direct measurements, the choice of the transducer must be made carefully, because it must interfere minimally in the operation of the compressor. Among the options for kinematic measuring transducers are: eddy; hall; fiber optics; linear variable inductance transducer (LVIT); encoder, and magnetic pick-up sensor (Bela, 2003) (Pickerill, 2009). However, due to reduced space, constant exposition to oil and coolant impregnated environment, high temperature, operating frequency and electromagnetic interference, the transducers are normally encoders, to measure the angular position of the shaft, or a magnetic pick-up sensor, to measure the average compressor speed. Linear variable inductance transducers have gained prominence to measure the instantaneous piston position, due to its more robust and miniaturized construction.

Although there are plenty of articles describing direct and indirect forms to measure the compressor kinematics, they lack to describe why a specific type of transducer was selected for that kind of application. Furthermore, no study was found that compares the use of different methods to measure the compressor kinematics and presents clear guidelines for choosing the best method of instrumentation and measurement for a given specific application. This paper presents the results of the experimental evaluation of simultaneous measurements done with an LVIT and an encoder for measuring the kinematics of reciprocating compressors. The piston position, angular position and rotational speed measurements obtained from an LVIT were compared to the ones obtained from an encoder. Based on the experimental results, some guidelines are presented for defining the measurement method for different applications.

This paper is organized as follows. Section 2 presents the development of kinematic equations for the slider-crank mechanism of a reciprocating compressor, taking into account its eccentricity. Section 3 presents a review of LVIT and encoder measuring principles. Section 4 describes the test rig developed for comparing both measurement methods. Section 5 summarizes the conclusions.

2. COMPRESSOR KINEMATICS

This section presents the kinematic equations of a compressor based on the slider-crank mechanism. A schematic representation of such a compressor is presented in Fig.1.

The coordinate origin is located at the center of the drive shaft (axis of the electric motor) so that the *x*-axis is parallel to the axis of the cylinder, the *y*-axis is perpendicular to it, and the *z*-axis is parallel to the drive axis. It is assumed that the motion of the mechanism occurs only in the *xy*-plane. The connecting rod, with length l, transforms the rotational movement of the drive shaft, at a distance r, in a reciprocating motion of the piston. The drive shaft is located on the *y*-axis at a distance d from the cylinder axis. The compression cycle begins when the piston is at the Top Dead Center (TDC) at $\theta = 0^{\circ}$, and after a full rotation of the axis at a rotational frequency ω, the piston returns to the same point at $\theta = 360^{\circ}$.

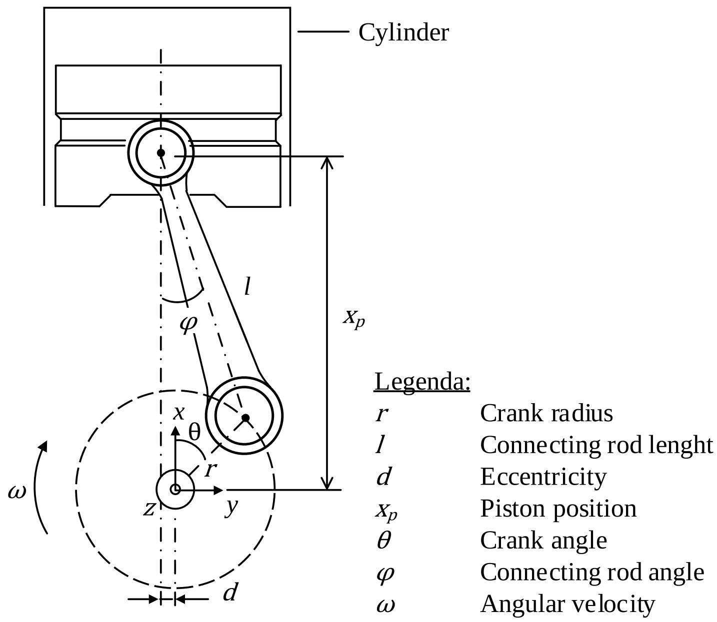

Figure 1: Slider-crank mechanism.

The position of the piston on the x-axis in relation to the origin of the coordinate system is represented by x_p. Using plane trigonometry, the piston position can be expressed as in Equation (1).

$$x_p = r\cos(\theta) + l\cos(\varphi). \tag{1}$$

The first derivative of Equation (1) with respect to time gives the following expression for the speed of the piston in the x-axis direction.

$$\dot{x}_p = -r\dot{\theta}\sin(\theta) - l\dot{\varphi}\cos(\varphi). \tag{2}$$

The second derivative of Equation (1) with respect to time gives Equation (3), which is the expression for the acceleration of the piston in the x-axis direction.

$$\ddot{x}_p = -r\dot{\theta}^2\cos(\theta) - r\ddot{\theta}\sin(\theta) - l\dot{\varphi}^2\cos(\varphi) - l\ddot{\varphi}\sin(\varphi). \tag{3}$$

Angle φ is measured from the x-axis towards the connecting rod and is assumed to be positive in the counter-clockwise direction. Using plane trigonometry, the angle φ can also be expressed as a function of the angle θ, as shown in Equation (4) (Norton, 2009).

$$\varphi = \arcsin\left(\frac{r}{l}\sin(\theta) + \frac{d}{l}\right). \tag{4}$$

The first and second derivatives of φ with respect to time result in Equations (5) and (6), which are the expressions for the calculation of the angular speed and the angular acceleration of the connecting rod.

$$\dot{\varphi} = \frac{r}{l}\dot{\theta}\frac{\cos(\theta)}{\cos(\varphi)}. \tag{5}$$

$$\ddot{\varphi} = \frac{r/l(\ddot{\theta}\cos(\theta) - \dot{\theta}^2\sin(\theta)) + \dot{\varphi}\sin(\varphi)}{\cos(\varphi)}. \tag{6}$$

The crankshaft angle depends on the piston position and on the connecting rod angle. Thus, expressing the crankshaft angle as a function of piston position is not a trivial task. Equation (7) relates all three variables by using Pythagorean theorem.

$$l^2 = (x_p - r\cos(\theta))^2 + (d + r\sin(\theta))^2. \quad (7)$$

Equation (8) is obtained by developing Equation (7).

$$l^2 - d^2 - x_p{}^2 - r^2 = 2r(d\sin(\theta) - x_p\cos(\theta)). \quad (8)$$

To express the crankshaft angle as a function of the piston position, sine and cosine terms in Equation (8) are grouped and the angle is isolated. The crankshaft angle can then be written as a function of the piston position, as in Equation (9).

$$\theta = \arcsin\left(\frac{l^2 - x_p{}^2 - d^2 - r^2}{2r\sqrt{d^2 + x_p{}^2}}\right) + \arctan\left(\frac{x_p}{d}\right) \quad (9)$$

3. DIRECT MEASURING METHODS

This section presents a brief overview of direct measuring methods traditionally used for the kinematics of reciprocating compressors: LVIT, magnetic puck-up, and rotary encoder.

LVIT consists of a core of paramagnetic and electric conductor material (commonly aluminum) that moves inside a coil. The displacement of the core causes an opposite magnetic field that changes coil inductance (Meydan & Healey, 1992) (Seco, Martín, Jiménez, & Calderón, 2005) (Alciatore & Histand, 2012). A conditioning module, which also drives the coil with a high frequency current source, measures the inductance variation and transforms it into an electrical signal. LVIT has high linearity (typical linearity error of ±0.15%), almost no hysteresis, good zero stability (an aluminum core has typically 140 ppm/K offset voltage drift) and can accurately measure dynamic displacement containing high frequency motion components. In addition, its dimensions are suitable for installation inside a hermetic compressor and it can be used exposed to oil (Alciatore & Histand, 2012). To allow the measurement inside a compressor, the core is fixed to the base of the piston so that both have the same motion. Since the LVIT signal represents the instantaneous position of the piston, the speed and acceleration of the piston can be determined by numerical differentiation.

The magnetic pick-up transducer consists of a permanent magnet fixed at a given position of the crank and a coil in a stationary part of the compressor, so that the magnet passes very close to it once per cycle (Pickerill, 2009). The magnet and coil positions are usually chosen such that the magnet passes close to the coil when the piston is at the midpoint of the suction process. The magnet causes a variation of magnetic flux in the coil as it travels close to it. This causes an induced voltage to be generated at the coil terminals. The disadvantage of the magnetic pick-up is that it measures a single angular position and, consequently, the rotational speed is an average value. On the other hand, the magnetic pick-up has a lower cost than its alternatives (Pickerill, 2009).

A rotary encoder measures the angular position of the axis by means of pulse counting (incremental encoder) or a unique code for each position (absolute encoder), allowing the measurement of the instantaneous rotational speed (Alciatore & Histand, 2012). When used inside a compressor, the encoder is normally connected to the oil pump by a flexible coupling, but it can also be connected to the counterweight at the point aligned with the axis shaft. However, the small volume inside the compressor is a limiting aspect for many encoders. Since optical encoders use light to identify the angular position, they need a clean environment to work properly. Thus, special care is needed when they are used in an aggressive environment, such as being impermeable to debris and oil (Alciatore & Histand, 2012). If an encoder does not have an adequate degree of protection, oil can cause severe damage to it. Magnetic encoders are an alternative to optical encoders and use magnetic field sensor technology to determine the angular position. Even though they are recommended for applications in dirty environments, their resolution and accuracy are still limited. In addition, their application is limited to areas with low external magnetic fields (Xu, Wang, Chin, Chang, & Sung, 2014).

When using an incremental encoder, the pulses from a reference position must be counted, such as TDC, and the number of pulses is multiplied by the angle spacing between them. To determine the rotational speed, it is necessary to measure the time interval between pulse transitions (Alciatore & Histand, 2012). The angular acceleration is obtained by numerically differentiating the rotational speed signal.

4. TEST RIG AND EXPERIMENTAL RESULTS

This section presents the design of the test rig used to evaluate an LVIT for measuring the instantaneous piston position, shaft angular position and rotational speed. For validation, the LVIT experimental results were compared to measurements obtained from an encoder.

4.1 Test Rig

A test rig was designed to measure simultaneously the position of the piston and the crankshaft angular position of a reciprocating compressor. The incremental optical encoder used in this study has two 600-pulse-per-revolution channels and a reference channel, which provides a pulse per revolution. The encoder was installed by connecting it to the oil pump using a flexible coupling. The LVIT used has an aluminum core and a cutoff frequency of 200 kHz. The coil was attached to the compressor block and one of the core tips was bonded to the base of the piston.

Dynamic and static tests were performed. The dynamic tests were performed at the rotational speed of 20 Hz. The discharge and suction lines were kept open to atmospheric pressure. The laboratory room temperature was controlled within (22±3) ^{o}C. The measurements of the two transducers were synchronized using the encoder reference channel as a trigger. Fig.2 shows the test rig, with its main components and the interconnection between them.

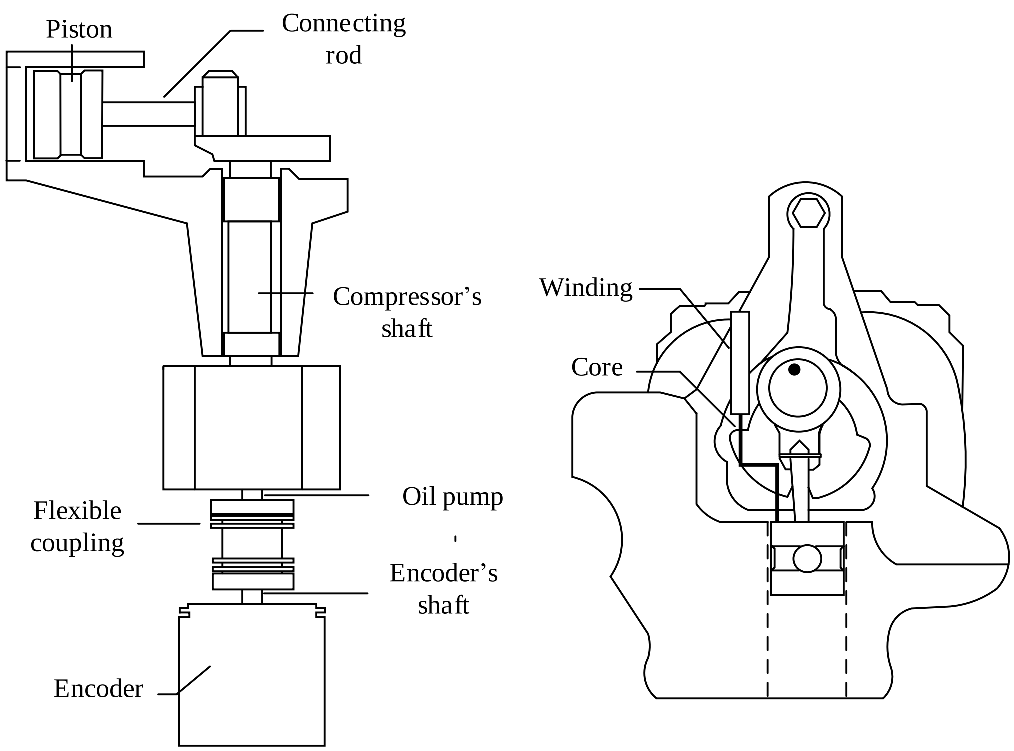

Figure 2: Mounting of the transducers.

4.2 Results

The dynamics results for the piston position are shown in Fig.3. Zero piston displacement is the Bottom Dead Center (BDC). The shaft angular position results are shown in Fig.4. The direct measurement of the encoder is compared with the value obtained from the LVIT transformed to crank angle using Equation (9).

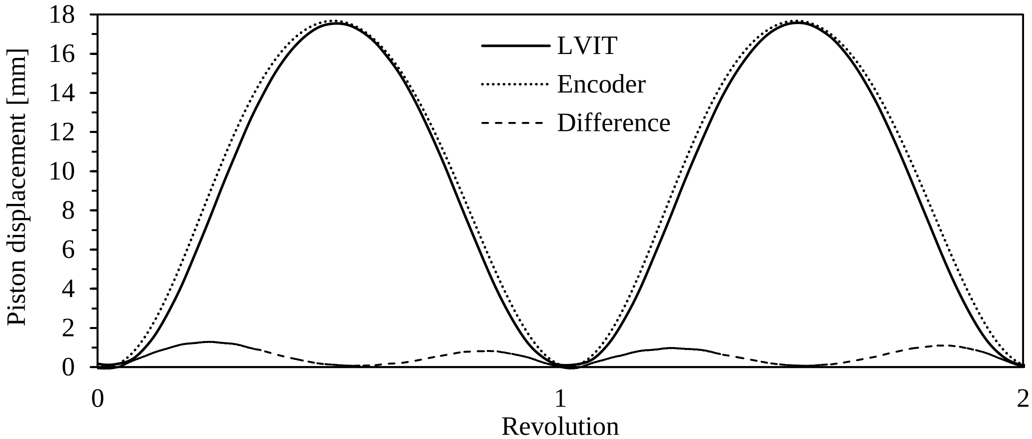

Figure 3: Piston displacement measurements.

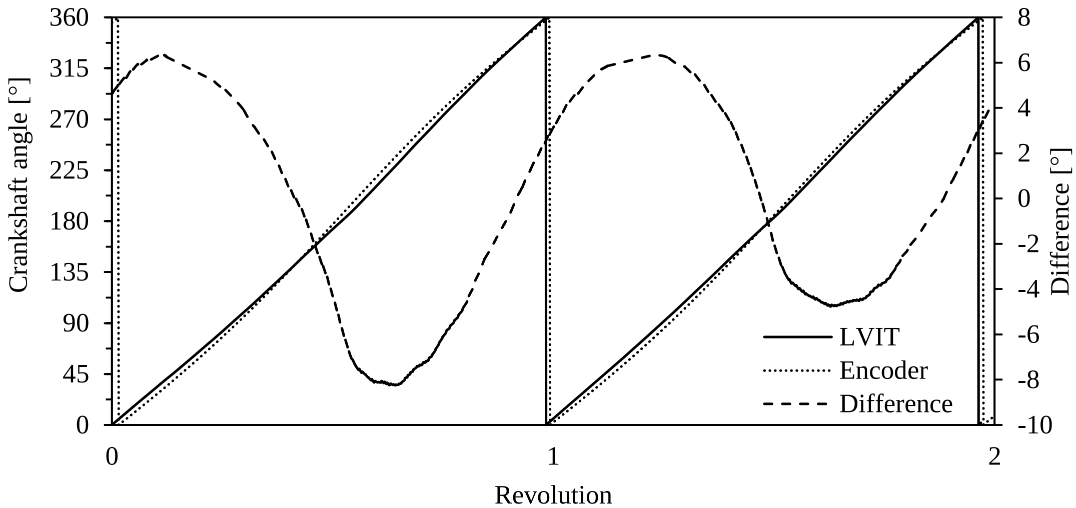

Figure 4: Crank angle measurements.

The difference accentuates as the piston moves away from BDC and TDC. Near TDC and BDC, the linear displacement of the piston is minimum, which reduces the difference. The maximum piston displacement difference is 1.3 mm. Since the stroke of the piston is 18 mm, this is a considerable difference for tests such as the one which determines the pV diagram. The maximum crank angular position difference is approximately 8°. One source of error is the accuracy of the mathematical models, which do not consider hysteresis and stiffness of the components. In addition, the input parameters also have a measurement uncertainty associated to them. Dynamic phenomena can also cause these differences. It is believed that the reciprocating movement of the piston affects the LVIT signal due to the finite stiffness of the core and lateral movements inside the coil. Contact forces can bend the core.

The rotation of the shaft does not change direction and its rotational speed is approximately constant. On the other hand, the piston speed continuously changes its direction. Furthermore, during its linear displacement, the piston can adhere to the cylinder walls near TDC and BDC, causing a sudden increase in the velocity of the piston when the static friction is overcome (Nachane, Hussain, & Iyer, 1998).

Additionally, a static test was performed to eliminate these dynamic phenomena that affect the measurements. The crank was rotated in steps of 18° from TDC to BDC and then rotated back from BDC to TDC in the same path. The results of this test are shown in Fig.5.

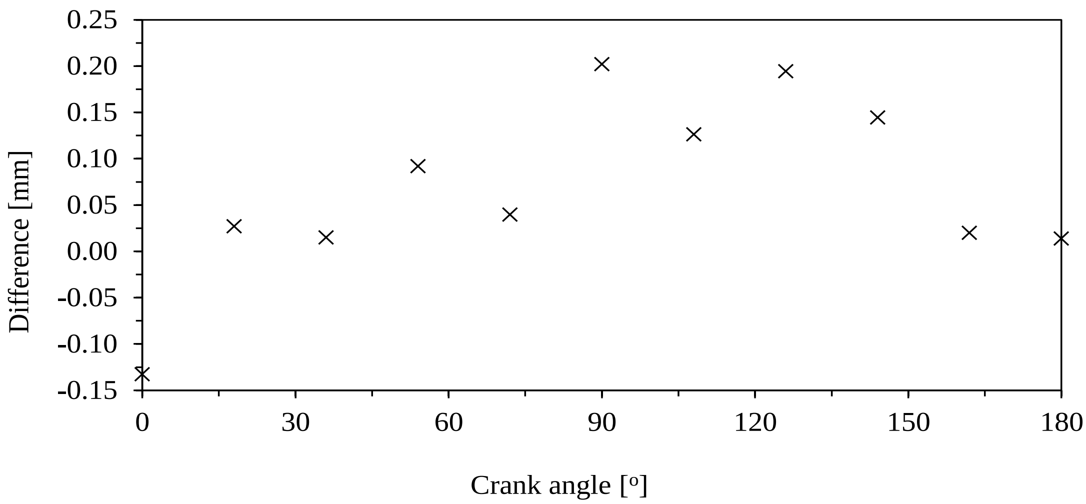

Figure 5: Crank angle static measurements.

The maximum difference between the encoder and the LVIT measurements is 0.20 mm, which is significantly lower than the differences obtained in the dynamic test. This shows that the dynamic phenomena make the piston motion more irregular when compared to the crankshaft motion. These phenomena are not transmitted to the crankshaft motion. In addition, the movements of the LVIT core are not present in this situation. Thus, the differences increase when the compressor is in operation. The rotational speed of the compressor obtained from both transducers is shown in Fig.6.

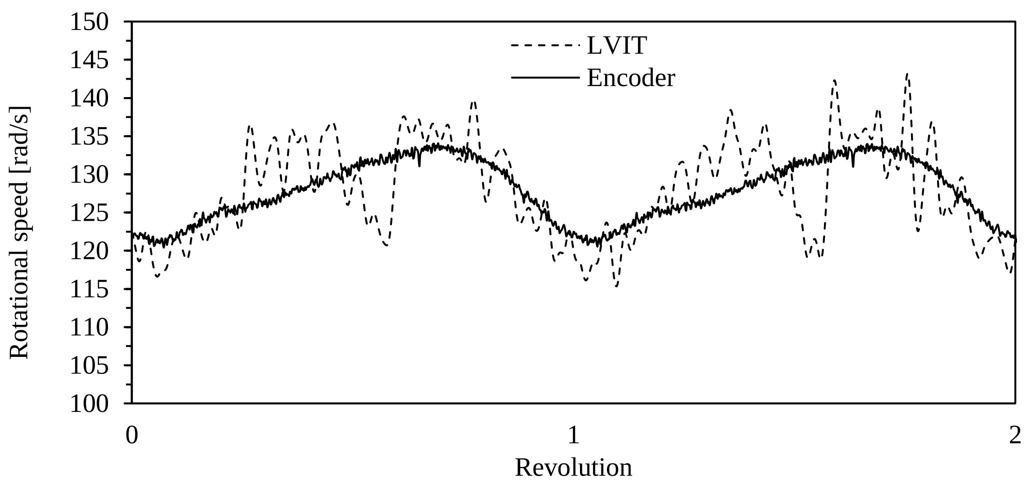

Figure 6: Crankshaft rotational speed.

The angular speed is not constant due to pressure, friction and inertia forces acting in the connecting rod that cause a torque on the shaft (ASHRAE, 2005). Although the instantaneous rotational speed has greater dispersion, the average values over one revolution are approximately constant, as shown in Table 1.

Table 1: Average angular speed.

	1st cycle [rad/s]	2nd cycle [rad/s]
LVIT	128.1	127.7
Encoder	128.0	128.0

Therefore, LVIT is an adequate alternative to encoders for tests that require the compressor rotational speed, because the difference between both transducers has the same order of magnitude than the measurement uncertainty, detailed in Section 4.3. It is also possible to obtain the instantaneous or average angular accelerations by differentiating the rotational speed signal. This information can be used to determine the compressor mechanical losses.

4.3 Measurement Uncertainty

Additional tests were performed to quantify the measurement uncertainty of the encoder. The encoder shaft was rotated at a very precise controlled speed to measure the time interval between successive transitions of the encoder output. Since the speed was constant, the time interval can be easily transformed into units of angle and compared with the theoretical angle, which depends on the encoder resolution. The result of this measurement uncertainty evaluation was $\pm 0.01°$ (95% of reliability).

For the LVIT used in this study, the measurement uncertainty was determined using the Monte Carlo Method (ISO, 2008). The input parameters were the specifications of the signal processing system and of the data acquisition module, as well as physical characteristics of the transducer, such as the misalignment between its core and coil. The result of this evaluation, for 95% of reliability, was ± 50 μm.

5. CONCLUSION

This paper presented a review of available kinematic measuring methods for reciprocating compressors. An experimental study was performed to evaluate the practical applicability of an LVIT for measuring the kinematics of reciprocating compressors. Both static and dynamic tests were performed. Results indicate that LVIT is suitable not only for measuring the instantaneous piston position, but also instantaneous shaft position and average rotational speed. Instantaneous rotational speed has a large measurement dispersion, which is amplified when the signal is differentiated. Although some difference was observed between the LVIT and encoder measurements, due to the adverse conditions inside the compressor, it is believed that LVIT will become the standard direct kinematic measuring method for refrigeration compressor tests. Special attention should be paid to the bending effect of LVIT core when dynamic measurements are performed.

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