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ENGINE CRANKSHAFT SPEED MEASUREMENT ERROR COMPENSATION

ABSTRACT: Engine speed signal contains valuable and rich content information about the engine working process. Measurement of the engine speed is usually carried out by sensor wheels mounted on the crankshaft. Measurement errors are mostly influenced by geometric tolerances of the engine speed sensor wheel as well as by its radial run-out. In order to avoid the occurrence of this systemic deviation, in the lab and highest accuracy engine speed measurements are usually conducted by means of incremental encoders, radially and axially fixed to the crankshaft with the anti-twist safeguard toward the crankcase. This paper shows how this type of engine speed measurement, which is considered to be the most accurate, is prone to systematic errors as well. The influence of the kinematic parameters of the encoder - crankcase linkage, on the error occurrence is discussed as well as suggested methods for systematic deviation parameters identification and engine crankshaft angular speed error compensation.

KEYWORDS: engine speed measurement, measurement error, crank angle optical encoder, IC engine

INTRODUCTION

Engine crankshaft speed is a parameter which is often unavoidable to be used in numerous internal combustion engine applications. Depending on the application, this parameter can be used as an averaged or discrete instantaneous value. In applications which involve a monitoring of engine speed, this parameter is usually time averaged, but in more demanding applications the measurements of engine speed are often averaged on per cycle or per revolution period. When used as an input parameter for the control of internal combustion engines, as a crucial and indispensable parameter, it is measured in angle based resolution, thus becoming a signal which gathers information on instantaneous engine crankshaft speed.

Demands for a high quality and accuracy of the measured engine speed become stricter in more sophisticated research applications both in terms of signal amplitude and resolution. This is particularly true in research which is related to the identification of the dynamic parameters of crankshaft or, more general, complete crankshaft mechanism. It is well known fact that the combined action of gas and inertial forces, within an internal combustion engine, generate highly varying torque, which, coupled with the torsionally elastic behaviour of the engine crankshaft, produce rapidly changing instantaneous engine speed. This complex picture of instantaneous engine speed contains valuable information about engine processes and it is therefore widely used in a vast variety of engine applications - from engine control to engine diagnostics.

During the past decades a lot of effort is put in research of methods which can provide the needed accuracy in the measurement of engine crankshaft angular speed. The problem in defining the most appropriate method which leads toward the highest possible measurement accuracy is in the necessity to solve multiple and rather diverse issues. Moreover, as usual in the engineering practice, researchers are additionally and always faced with the

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common demands about keeping the costs of measurement chains as low as possible. Although mentioned problems are common for all types of engine speed measurement devices, it is a general belief that the usage of more sophisticated and expensive devices, like optical encoders, can solve almost all of these problems. This research shows that above mentioned assumption can be misleading and that serious measurement errors can be made even by using optical encoders – despite their high geometrical tolerances and design quality.

SOURCES OF ANGULAR SPEED MEASUREMENT ERRORS

Angular speed measurement basics

The key component of an angular speed measurement chain is a teethed or slotted wheel mounted on a crankshaft. Teethes or slots are used as a markers for generating of digital signals which are further used for triggering and stopping of the time period measurements between successive signals. Number of teethes Z defines the angular increment φ_0 of the crankshaft which can be detected and is firmly related to the capabilities of measuring instantaneous angular speed.

$$\varphi_0 = \frac{2\pi}{Z} \tag{1}$$

Measurement of the time period T_m needed for the angular movement of the crankshaft between two succeeding angular increments gives averaged angular speed on that (i -th) interval:

$$\omega_m(i) = \frac{2\pi}{T_m(i) \cdot Z} \quad i = 0 \dots Z - 1 \tag{2}$$

The measurement of the time period T_m is usually done by counting the number of impulses N of the digital pulse train with reference frequency f_0 . The goal of this procedure is to quantise time period T_m into product $N \cdot T_0$ (where $T_0 = 1/f_0$). This quantisation gives the discrete, and thus approximate, averaged angular speed on a measured interval - ω_q . The relative error of this approximation can be evaluated as :

$$\epsilon_\omega = \frac{\left| \frac{\varphi_0}{N \cdot T_0} - \frac{\varphi_0}{T_m} \right|}{\frac{\varphi_0}{T_m}} = \left| \frac{T_m}{N \cdot T_0} - 1 \right| \tag{3}$$

The absolute value of the quantisation error has to be smaller than quantisation time interval:

$$|T_m - N \cdot T_0| \leq T_0 \tag{4}$$

Therefore an estimation of the relative quantisation error can be expressed as:

$$\epsilon_\omega \leq \frac{1}{N} = \frac{1}{N \cdot T_0 \cdot f_0} = \frac{\omega_0}{\varphi_0 \cdot f_0} \tag{5}$$

The quantization error depends on the angular speed itself and, at higher engine speeds, it can be improved by increasing the reference signal frequency f_0 or by enlarging the angular interval between increments φ_0 . The latter measure has its drawbacks since the enlargement of the measuring interval decline measurement resolution. Smaller number of teethes i.e. larger angular increments gives not only low resolution in instantaneous speed measurements, but also in crankshaft angular position tracking which is important in engine control applications. These problems can be solved by using additional layer in sensor signal processing chain by producing multiple micro-ticks during one angular interval [2].

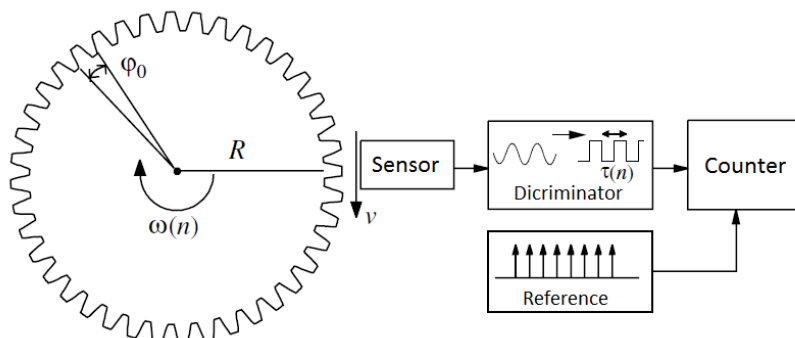


Figure 1 Measurement chain for period based measurement of angular speed [1]

Mechanically introduced measurement errors

Measurement errors, which can be exceptionally larger than those introduced by quantisation process have an origin in geometrical irregularities of measurement wheels or in the undesirable kinematics of the sensor elements. When the speed measurement is based on the usage of toothed wheel, error which is often introduced is a consequence of the radial run-out of the wheel.

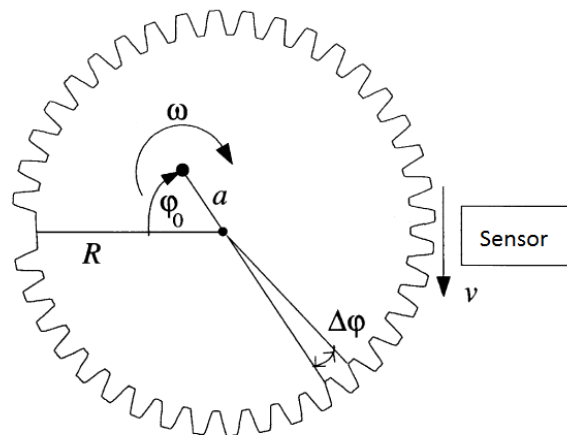


Figure 2 Angular speed measurement errors due to measurement wheel run-out [1]

The sensor, which is static, detects relative speed as:

$$v(i) = \omega \cdot (R - a \cdot \sin(i \cdot \Delta\varphi - \varphi_0)) \quad (6)$$

i.e. the measured angular speed:

$$\omega_m(i) = \frac{v(i)}{R} = \omega \cdot \left(1 - \frac{a}{R} \cdot \sin(i \cdot \Delta\varphi - \varphi_0)\right) \quad (7)$$

where parameter a is the wheel eccentricity (run-out value) and R is a wheel base radius.

When known, parameters a and R can be directly used for the calculation of the measured angular speed correction. In practice, this parameter should be identified and this can be achieved by measuring the angular speed of the wheel by providing constant shaft speed. This is, in most cases, rather unpractical approach on the engine and therefore other methods are developed and used. Excellent results can be achieved by driving the engine on regimes where the influence of the gas torque on the engine speed can be minimised as much as possible e.g. during short period of engine fuel cut-off and crankshaft deceleration affected by external engine load [3]. When measurements are made the unknown kinematic parameters can be identified by means of least squares method [1].

Besides measurement wheel run-out problem, additional errors can be introduced through unequally spaced increments on a wheel. These errors are almost unavoidable due to mechanical tolerances of toothed wheels in the production process. This mechanically introduced errors can be compensated through additional signal processing and previous deviation parameter identification. There are different approaches and parameter identification methods. Simple and straightforward compensation parameters estimation is suggested by Kiencke in [1], while Fehrenbach et. al. proposed method which is based on the energy balance equation of the crankshaft mechanism and is well described in [4], [5] and [6]. A similar approach is also used by Rämisch in [7].

When using toothed wheel, mounted on the crankshaft, with statically mounted Hall or variable reluctance sensor, for the angular speed measurement, a compensation of the error due to above mentioned mechanical problems is almost unavoidable. Integral and precise design of a typical industrial type optical incremental encoder and its usage should eliminate this problem with its nature. Nevertheless, errors in measuring crankshaft speed can be still encountered due to several reasons.

Optical incremental encoders are typically mounted on the engine crankshaft in two manners – as directly or indirectly attached. In one of them the encoder is statically mounted in front of the crankshaft and indirectly connected to by means of a torsionally stiff coupling. Problems encountered in this configuration are usually related to the fact that the coupling, which stiffness is not infinite, dynamically modifies and distorts the crankshaft motion [8].

Optical encoder, used in this research, was radially and axially fixed to the crankshaft (direct connection), while the encoder's stator motion was disabled by means of the anti-twist safeguard toward the crankcase. This configuration is often believed to be the superior one in terms of measurement precision and accuracy and is widely used in research applications on engine test benches. Although, this configuration is very sensitive to a mounting faults and this paper's motivation is mostly to emphasize how much this type of faults can introduce huge errors while measuring instantaneous engine crankshaft speed. The proposed method for a correction of the measured engine speed and encoder mounting errors compensation is developed and applied during a research on a 1.4 litres 4-cylinder engine (produced by DMB and very similar to FIAT 1372 ccm SOHC Engine).

CRANK ANGLE ENCODER – MOUNTING ERRORS COMPENSATION

Figure (3) shows an example of the measured angular speed on one of particularly interesting working points. The chosen working point is in the area of middle range engine speed at relatively high load – area in which gas torque dominates and prevails over inertial torque components. Within that engine working area, it is common to expect a picture with four similar engine speed peaks within a complete engine cycle. As evident, the recorded picture looks quite different and this can originate either from the engine itself, if the cylinders differ in energy contribution, or from faulty engine speed measurement. Each of four main oscillation shapes corresponds to a single cylinder activity and there should be noticed slight differences between them as a natural consequence of interference of crankshaft's dynamical behaviour. It can be noted that engine speed fluctuation on cylinders no. 1 and 4 (firing order 1-3-4-2) are much higher than those on cylinder 2 and 3. Since no other parameters measured on the engine did not indicate any problems in difference in cylinder energy contribution the suspicion led to measuring deviation which is periodic over a single crankshaft revolution. Moreover, the spotted problem looked very similar to an error produced by a run-out of the measuring toothed wheel.

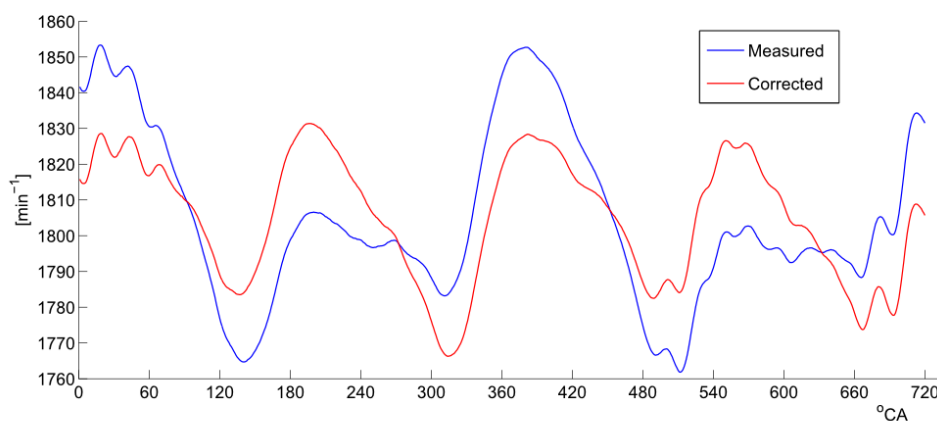


Figure 3 Measured and corrected angular speed – an example (DMB 1.4, $n=1800 \text{ min}^{-1}$, $p_e=6 \text{ bar}$)

Figure (4) shows the crank angle encoder, used for measurement, mounted on the free end of the crankshaft. The connection between the encoder and the crankshaft is realised in the usual manner on almost exactly the same way as on the Kistler type engine crank angle encoders. The encoder's stator motion is disabled by fixing a secure arm to the adjustable linkage, which is placed between the arm and engine crankcase. The fixing linkage consists of two spherical joints and adjustable length screw rod, thus enabling the wide possibilities in fixing the encoder's secure arm to the most suitable point on the crankcase.

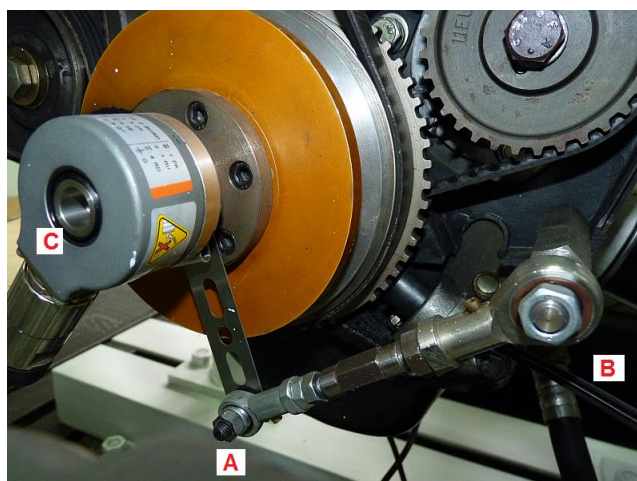


Figure 4 Crank angle encoder mounted on the crankshaft's free end

Due to the tight tolerances used in encoder assembly, the typical run-out of the optical disk is a problem which cannot be expected from this type of device. But, as will be shown, slight radial misalignment between the encoder and crankshaft axis can lead to a circular motion of the complete encoder assembly. This motion, constrained by secure arm and fixing linkage kinematic chain causes a relative motion between the optical disk and encoder's stator which consequences as false crankshaft position reading what further influences angular speed measurement.

Direct mounted single body encoder kinematics

The kinematics of the crank angle encoder which is exposed to the radial run-out motion is explained on figure (5) on which "C" marks the crankshaft axis, "B" marks the attachment point (spherical joint) of the encoder linkage to the crankcase and "A" marks the position of the other spherical joint between the encoder's arm and linkage rod. In order to make analysis easier, the schematic is built around coordinate system xCy . Some arbitrary initial position of the encoder can be chosen but, it is convenient to choose the position in which axis center "C", encoder center "E1", and linkage endpoint "A" are collinear. The chosen initial position of the encoder body is highlighted with circle "k_o" which is intentionally positioned eccentrically from crankshaft axis i.e. the center of circle "k_o" is positioned in the eccentricity circle "k_e". The fixed distance between crankshaft axis and anchoring point on the crankcase "BC" is marked with letter "d".

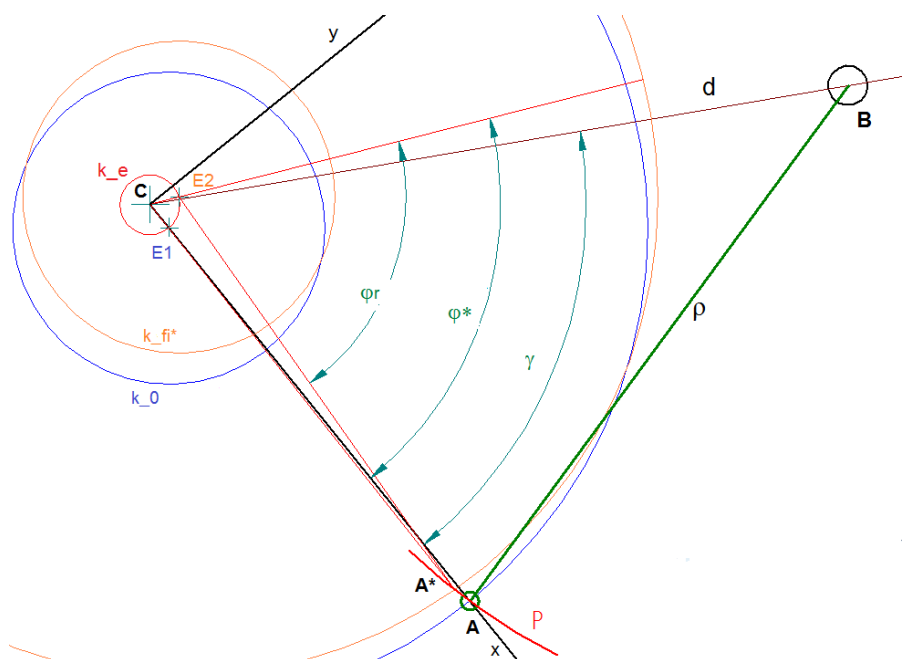


Figure 5 Kinematics of angle crank encoder exposed to run-out motion

When the eccentricity is present, the movement of the encoder's stator can be explained as a complex virtual combination of two simpler motions:

- The primary one, during which encoder body follows and rotates together with its rotor (optical disk) and crankshaft. During that angular movement, described with angle φ^* whole encoder rotates to a new position where its center takes position at point E2. Both E1 and E2 points belong to circle "k_e" i.e. they are equidistant from the crankshaft centre point „C" by eccentricity radius e ($CE_1 = CE_2 = e$).
- Secondary motion through which the encoder's stator revolves back, with fixed rotor, around point "E2" for angle φ_r , in order to "connect" point A of the secure arm with the end of the linkage rod BA. It is assumed that the linkage rod length ρ and secure arm length R are constant and not deformable. The consequence of the radial run-out of the complete encoder body is the reposition of the spherical joint from point "A" to point "A*" which forces a rotation of the linkage rod BA around point B.

From one side, connecting point "A" is able to rotate along arc "p" with radius ρ (circumscribed from the point "B"). On the other side, belonging to encoder's secure arm, the point "A" has to be a part of the circle circumscribed from the centre of encoder – point "E" with radius R. Thus the position of the encoder stator is defined by the intersection of this circle and the arc p. The direct effect of the run-out of encoder body is the inequality of above described angles φ^* and φ_r with a difference which oscillates between minimum and maximum value over each crankshaft revolution. The angle difference is converted into the relative motion between the encoder's stator (which contains

the optical sensors) and its rotor (optical disk) and this angle difference time derivative is an angular speed component which interferes with the measured speed and appears as an error (figure 3).

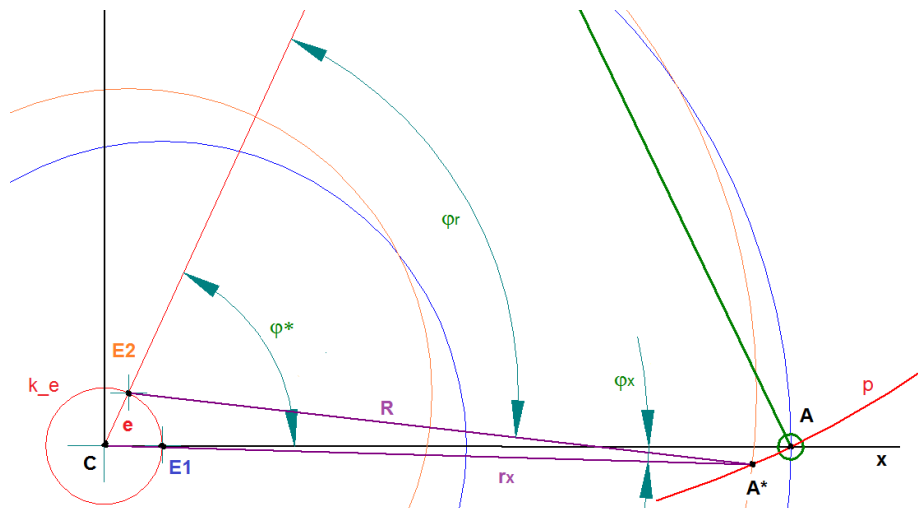


Figure 6 Relation between the angular movements of encoder's stator and rotor

If the point C is an origin of the polar coordinate system in which the polar distance can be designated as r and position angle as φ (measured from half-line Cx) than the arc p can be described with the following equation (fig. 6):

$$-r^2 + 2 \cdot r \cdot d \cdot \cos(\varphi - \gamma) = -\rho^2 + d^2 \quad (8)$$

The equation of the encoder body circle, rotated for the angle φ^* in the "new" E2 position states:

$$r^2 - 2 \cdot r \cdot e \cdot \cos(\varphi - \varphi^*) = R^2 - e^2 \quad (9)$$

With an assumption that the run-out value e is already known, it is possible, by eliminating the variable r from the system of equations (8) and (9), to solve the system by variable φ and to identify the exact angular position φ_x of point A*.

Angular difference, defined as $\Delta\varphi_{err} = \varphi_r - \varphi^*$, corresponds to the relative angular movement of encoder's stator due to encoder run-out. Evaluation of its first order time derivative gives a value which should be used for measured angular speed correction through simple addition. Geometrical relations, which can be spotted on figure 6 give the following equation:

$$\Delta\varphi_{err} = \varphi_x + \arcsin\left(\frac{e}{R} \cdot \sin(\varphi^* + \varphi_x)\right) \quad (10)$$

By using eq. (10) it is possible to evaluate the "error" in measured crankshaft angular position for each value of crankshaft position φ^* on complete revolution period, i.e. for $\varphi^* \in [0 \dots 2\pi]$.

The run-out value e as well as the angular position φ_{ref} of the polar coordinate system used during equation derivation cannot be determined in advance – it should be identified. For an illustration the angle φ_{ref} used in figure (5) have a value of zero, but in practice it can take a numerous values from an angular interval which is specific for each particular encoder mounting disposition.

In order to identify the unknown parameters e and φ_{ref} , it is necessary to provide representative or reference angular speed measurement data. This can be obtained through measurements on particular engine regimes (like those with shortly introduced misfire or steady-state cranking). Data from other regimes can be used also but only by relying on the assumption that the energy contributions from all engine's cylinders are equal. The latter approach can be very practical and useful, especially when used on engine's working points with low COV (*coefficient of variation*) values.

CASMA filter as tool for run-out parameter identification

The evaluation of the energy contribution of each engine cylinder is a common practice often used in engine diagnostics. It is used for detecting of unequally injected fuel quantities among different cylinders, or to detect a

misfire on a particular cylinder. One of very efficient methods for evaluating the cylinder contribution is developed by Schmidt [9] and called CASMA (Crank Angle Synchronous Moving Average) filter.

CASMA calculations are based on the assumptions that the crankshaft is rigid and that entire crankshaft mechanism can be simplified as a single rotating mass with the equivalent moment of inertia J_{ekv} . Upon these assumptions the crankshaft motion equation can be presented in simplified form:

$$J_{ekv} \cdot \frac{d\omega}{dt} = \sum M_i \quad (11)$$

Where M_i are torque components which originate in gas and friction forces, external and auxiliary loads as well as in inertial torques. Every of this torque components can be evaluated as the sum of two superposed signals – one of which is periodical and other with a constant value:

$$M_i(t) = \bar{M}_i(t) + \tilde{M}_i(t) \quad (12)$$

The characteristic torque period in a 4-cylinder engine is 180° i.e.:

$$\tilde{M}_i(\varphi(t)) = \tilde{M}_i(\varphi(t) + \pi) \quad (13)$$

The mean value of the torque, on the π length interval can be calculated as:

$$\frac{1}{\pi} \cdot \int_{\alpha}^{\alpha+\pi} (\sum M_i) \cdot d\varphi = \frac{J_{ekv}}{\pi} \cdot \int_{\alpha}^{\alpha+\pi} \dot{\omega} \cdot d\varphi = \frac{J_{ekv}}{\pi} \cdot \int_{\alpha}^{\alpha+\pi} \omega \cdot d\omega \quad (14)$$

A Work of oscillatory torque components, over an interval of periodicity is equal to zero i.e.:

$$\int_{\alpha}^{\alpha+\pi} \tilde{M}_i \cdot d\varphi = 0 \quad (15)$$

That, actually, simplifies the left side of the equation (14) to the difference between works of mean value gas torques W_g and "constant" value torques originating from the friction forces and external loads W_{const} :

$$W_g \Big|_{\alpha}^{\alpha+\pi} - W_{const} = \Delta E_{kin} \Big|_{\alpha}^{\alpha+\pi} = \frac{J_{ekv}}{2} \cdot (\omega^2(\alpha + \pi) - \omega^2(\alpha)) \quad (16)$$

The difference, defined by the equation (16), will be equal to zero if the working process on that interval is quasi-stationary i.e. if its mean value is constant. If the energy contribution of one or more cylinders differs that will distort the mean value of the torque and it will be not constant any more on the periodic interval. Thus, the value of the expression (16) can be used as an indicator of the work balance between cylinders. The greater is the value of this indicator the greater is the difference of works between cylinders.

According to the equation (14), and the fact that the time derivate of the angular speed can be treated in the same way as the torque in equation (12), the difference (16) can be expressed as:

$$\Delta E_{kin} \Big|_{\alpha}^{\alpha+\pi} \sim \frac{1}{\pi} \cdot \int_{\alpha}^{\alpha+\pi} \bar{\omega} \cdot d\varphi \quad (17)$$

In other words, the balance indicator is proportional to the area beneath the angular speed on the interval of periodicity, filtered with the moving average. This conclusion implies that the indicator can be calculated even in dynamical regimes during which the cycle averaged speed is not constant.

If the sampling interval has a length of Φ_s , the number of sampled points on the angular interval $\Phi_w = 180^\circ CA$, equals $M = \Phi_w / \Phi_s$. Therefore, the equation (17) when discretised into the sampling resolution domain can be evaluated as:

$$\Delta E_{kin} \Big|_{\alpha}^{\alpha+\pi} \sim \frac{1}{\pi} \cdot \sum_{v=0}^{M-1} \dot{\omega}((k-v) \cdot \Phi_s) \cdot \frac{\pi}{\Phi_s} \cdot \Phi_w \quad (18)$$

$$= \frac{1}{M} \cdot \sum_{v=0}^{M-1} \dot{\omega}((k-v) \cdot \Phi_s)$$

CASMA algorithm calculates the angular acceleration $\dot{\omega}$ as:

$$\bar{\omega}(k) = \omega(k-1) + \frac{\omega(k) - \omega(k-M)}{M} \quad (19)$$

When the argument is the measured angular speed, according to eq. (19), the acceleration can be evaluated as:

$$\dot{\omega}_{meas}(k) = \frac{\sum_{v=0}^{M-1} \omega_{meas}(k-v) - \sum_{v=0}^{M-1} \omega_{meas}(k-v-M)}{\pi} \cdot \sum_{v=0}^{M-1} \omega_{meas}(k-v) \quad (20)$$

Since the angular speed is determined by period measurement (counting the number of reference clock impulses arrived during the period interval), the time duration of the period over 180° CA can be evaluated as:

$$\hat{T}(k) = \frac{\sum_{v=0}^{M-1} n_{impref}(k-v)}{f_{ref}} \quad (21)$$

Where f_{ref} is the reference clock frequency and $n_{impref}(k)$ is the number of reference impulses arrived during k -th moving period of 180° CA.

Thus the mean value of the angular speed over k -th period can be expressed as:

$$\hat{\omega}(k) = \frac{\pi}{\hat{T}(k)} \quad (22)$$

The overall result of a CASMA filter is a residue $r(k)$ which indicates the existence of energetic contribution differences between cylinders:

$$r(k) = \frac{\hat{\omega}_{meas}(k) - \hat{\omega}_{meas}(k-M)}{\pi} \cdot \hat{\omega}_{meas}(k) \quad (23)$$

Now, having the all needed tools, the crank angle encoder run-out can be determined using the following procedure:

1. For initially assumed values of $[e_0 \ \varphi_{ref_0}]$ the calculation of the corrective angle $\Delta\varphi_{err}$ is done and this value is used for the initial correction of the measured angular speed.
2. The corrected angular speed has to pass further as an argument to the CASMA algorithm in order to calculate the residue vector r
3. The residue value contains the most important information about differences between cylinders energy contribution. Since it is assumed that there is a minimum difference between cylinders energy contribution the value of this residue indicates indirectly how good was the guess of values $[e \ \varphi_{ref}]$.
4. In order to estimate the correct run-out values of the encoder the residue function is minimised through an iterative process by means of Levenberg – Marquardt algorithm [10]. Final result gives the identified run-out parameters $[e^* \ \varphi_{ref}^*]$.
5. Finally, the identified run-out parameters can be used for measured angular speed correction.

CONCLUSIONS

The crank angle encoder, used in this research had an identified run-out value of $e^* = 0.07 \text{ mm}$. Despite the fact that all possible precautionary measures were applied during encoder installation it was very difficult to accomplish near zero run-out condition, mostly due to a specific configuration of the crankshaft free-end shape and its front pulley design. Although the run-out was minimised and through monitored vibration of the encoder, almost unnoticeable, the produced error substantially distorted the information about instantaneous crankshaft angular speed. It is also interesting to note that detected run-out value was below mounting tolerance limits stated in some mounting manuals of the very similar equipment [11].

Kinematic analysis of the crank angle encoder assembly showed that it is possible to derive expressions for the evaluation of the angle encoder measurement errors. The applied CASMA algorithm also proved itself as a useful tool in the process of run-out error parameter identification. Also to have on mind – run-out parameter identification method used strongly relies on the assumption that there is not any significant difference in cylinder overall energy contribution.

It is shown that even a small eccentricity value, as a mounting error, can lead to significant errors in the measurement of the instantaneous angular speed. This is particularly true not only when less sophisticated singular speed measurement devices are used but also in the case of application of single body optical encoders. Special attention should be paid to the mounting and the evaluation of data gathered from the directly mounted encoders by means of secure arm and linkage rod towards the crankcase since the potential error depends not only on mounting eccentricity error but also from other parameters defining the kinematics of crank angle encoder.

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