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TEST STAND FOR MONITORING OF TORSIONAL VIBRATION OF ENGINE'S CRANKSHAFT BY INSTANTANEOUS ANGULAR SPEED MEASUREMENT

Continuous monitoring of diesel engine performance under its operating is critical for prediction of malfunction development and subsequently functional failure detection. Analysis of Instantaneous angular Speed (IAS) of the crankshaft is considered as one of non intrusive, and effective method of detection of combustion quality deterioration.The paper contains presentation of attempt of monitoring of piston engine's crankshaft torsional vibrations by measurement of Instantaneous Angular Speed at free and power output ends of the engine's crankshaft. It is assumed that calculation of differentia value between both ends shall give the picture of torsion angle magnitudes and phases of the peak values. The angular speed measurements is to be done utilising two optical sensors for reading and two perforated discs mounted at shaft's ends playing the role of speed signal emitters. In the paper is presented description of the measurement system and explanation of its mode of work. It is also shown analysis of measurement accuracy, way errors elimination and method of signals runs filtration. Presented results of experiment derives from test cycle carried out using laboratory stand of Gdynia Maritime University equipped with 3-cylinder self – ignition engine, powering electric generator.

***Keywords**: torsional vibrations, crankshaft angular speed, diesel-generators, optical method of measurement*

INTRODUCTION

Diesel engines are one of the most critical mechanisms having impact on safety of shipping. Unpredicted failures of engines, installed on board as main propulsion or electro-generators units can result with serious consequences, jeopardizing human life and environment. One of the most common problems occurring during diesel engines operation are malfunctions of fuel injection systems and subsequently problems with main and journal bearings lifetime [1, 2]. Moreover, as far as electro-generators are concerned, irregularity of rotational speed can affect quality of delivered electric energy (frequency stabilization). Detection of the piston – sleeve set giving no proper contribution to the total engine's torque value is possible in way of measurement of in-cylinder pressure using electronic or mechanical indicators. That method, although very effective has some inconvenience. Firs one is related to an engine construction – one has to have indicator cocks at every cylinder, second one is due gauges vulnerability, high temperature and exhaust gases pollution do not let continuous monitoring. Above presented facts leads to the conclusion that any other way, free from presented inconveniences shall be accepted for practical implementation [4].

In internal combustion piston engines, reciprocating movement of pistons is converted to rotary movement of the crankshaft. The angular speed is strongly affected by tangential force coming from gas pressure and vertical imbalance inertial forces induced by reciprocating masses of piston and connecting rod. The character of acting forces let assume that IAS can be utilized for detecting engine faults related to combustion process. Because of sequential ignition in cylinders and differences of combustion quality (i.e. burning process speed and duration, heat emission, pressure expansion) occurring between cylinders, angular speed of a crankshaft is not uniform. Variations of instantaneous angular speed value is reflecting level of unsteady character of subsequent pistons contribution. Pressure of combustion gases is transformed to a crankshaft in form of variable force, which depends on pressure value and crankshaft position. That force creates torque necessary for attached driven mechanism movement. In our case that was electro-generator producing alternate current. Superposition of instantaneous values of torques coming from driving engine and driven generator, with opposite vector direction, are reason of torsional vibrations of the shaft. Due to origin of occurring forces, character of torsional vibrations shall reflect quality of combustion forces.

The main goal of carried out investigation was answer the question whether IAS signal coming out from both ends of the crankshaft can be a source of information about combustion irregularity and whether level of torsional vibration can be controlled using IAS measuring method.

1. DESCRIPTION OF TORSION MEASUREMENT SYSTEM

General idea of torsional vibrations observation relays on simultaneous measurement of the distance done by opposite ends of the shaft. Twist of the crankshaft is result of difference of the angular distance done by the two random points placed on shaft's opposite ends circumference (see fig. 1). In order to determine the differential value of the distance, simultaneous measurement of angular speed of both ends of the shaft is necessary. Value of angular shift at perpendicular plains is to be calculated by formula (1):

$$\varphi = (\omega_A - \omega_B) \cdot \tau \,, \tag{1}$$

where:

ω_A, ω_B – angular speed of points A and B,
τ – time of speed measurement.

Accuracy and reliability of measurement depends on relation between torsions frequency and sampling frequency. In the worst case, torsional changes can take place in the time between subsequent samples and torsion won't be detected.

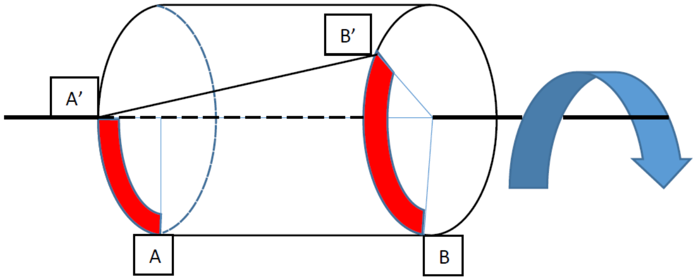


Fig. 1. General idea of torsion measurement as differential value of displacement of points A and B

In presented system, the disc A mounted at free end of the crankshaft plays the role of "master" gage and is reference for time counting. The angular speed of the second one, in "slave" position was calculated using time interval determined by master disc. It was assumed that basic measurement sample is a pair of slot-tooth, and number of impulses registered for such pair is the base of time calculation.

For measurement of Instantaneous Angular Speed was implemented encoder in form of perforated disc with windows placed at edge of the disc. All windows have the same dimensions and angular distance between them is 2°. Number of windows is 90, then windows are separated by 90 "teeth" with the same span 2° (see fig. 1). Rotating disc cuts laser ray pointed at disc's edge in middle of windows' zone. Measurement head, "u" shaped form, consist of a laser from one side and a photodiode at opposite arm (fig. 1) Laser impulses, emitted with frequency of 16 MHz going through windows creates signal with value "1" and stopped by tooth gives blind signals value "0". This mode of operations lets measure time of passing angular width of tooth and window, in form of a number of registered impulses. Above presented mode of operations must lead to the conclusion that accuracy of preparation of windows and equality of their dimensions are crucial for accuracy of measurements. In practice, that inconvenience can be partly omitted by taking for calculation an angle of a pair window-tooth, eventually bigger or smaller window's size is compensated by tooth dimension because total angular distance must be equal to 360°.

In order to mark the position of the disc in correspondence to the crankshaft position, the trigger in a form of one additional window (slot) narrow and asymmetric is to be placed in the zone of one tooth. That slot must be placed at a position "cutting" laser ray when piston in first cylinder is in TDC (Top Dead Centre). It lets allocate every part of IAS record to the crankshaft angle and specific cylinder, what is absolutely necessary for further diagnostic analyses.

Number of impulses registered for every slot-tooth pair is different and is in functional relation with instantaneous angular speed of the disc. In table 1 are presented examples of records of registered impulses transformed to MS Excel format. Values in the colon signed with letter A coming from disc mounted at free end of the crankshaft and values B respectively from end of generator.

The angular speed calculation can be explained using data from table 1. The sum of impulses for pair AZ1 and AS1 is:

$$i = \text{AZ1} + \text{AS1} = 4047 + 3078 = 7125, \quad (2)$$

instantaneous angular speed is presented in form of formula (3):

$$\omega = \frac{\frac{2\pi}{w}}{i \cdot f} \left[\frac{rad}{s}\right], \quad (3)$$

where:

- ω – angular speed,
- w – number of pairs “slot-tooth”,
- i – number of impulses register for pair window-tooth,
- f – frequency of laser emitter.

For our example, width of the pair is α = 0.0349 rad, and calculated time is:

$$\tau = i \cdot f = \frac{7125}{16} \cdot 10^{-6} = 0.000445\ s, \quad (4)$$

and angular speed:

$$\omega = \frac{\alpha}{\tau} = \frac{0.0349}{0.00045} = 77.555 \frac{rad}{s}. \quad (5)$$

Table 1. Example of record of angular speed both ends of the shaft rotating with mean speed of 753 rev/min

Teeth:90	**Rey:10**	**Samples:3600**					
A Z1	**A S1**	**A Z2**	**A S2**	**B Z1**	**B S1**	**B Z2**	**B S2**
4047	3078	4046	2999	3639	3497	3629	3462
4073	3025	4049	3095	3650	3450	3699	3379
4138	3031	3976	3099	3609	3485	3639	3469
4019	3004	4061	3062	3563	3453	3635	3517
4132	2991	4143	2998	3563	3459	3666	3446
4120	2935	4038	3009	3601	3490	3652	3443
4097	2988	4108	3026	3621	3475	3628	3476
3899	3097	3955	3005	3750	3337	3646	3435
3990	3001	3966	3098	3784	3362	3630	3461
4028	2998	4044	3015	3657	3444	3639	3403
4000	3052	3877	3042	3743	3328	3711	3334
4014	3001	3989	3067	3807	3350	3738	3336
3899	3068	3989	3058	3830	3373	3834	3307
3956	3106	4000	3060	3766	3204	3833	3344
3990	3090	4022	3017	3774	3405	3682	3239
4029	3011	4082	3067	3813	3355	3804	3273
4071	3033	4095	3069	3882	3244	3880	3265

The angular speed of the second disc is calculated the same way, using data BZ1 + BZ2. Both speeds were calculated using different time intervals τ_A and τ_B determined for disc A and disc B.

It would be very complicated to install two disc in the exactly the same position against TDC at both ends of the shaft. Most accurate installation would bear the error with value of slit dimension. To avoid above problem, displacement of master disc was referenced to "start slot" but angular instantaneous displacement of the second disc was calculated by multiplication of average speed resulting from impulses calculation (tab. 1) by time when master disc does the angular way of slot and tooth. Of course that approach to the problem requires verification of accuracy.

It was assumed that basis for angular displacement calculation shall be time coming from "master" disc A, and formula of the angular shift distance is:

$$\phi = (\omega_a - \omega_b)\tau_A. \tag{6}$$

Of course that mode of thinking is correct only for first pair, repetition of the way for second pair slot-window, subsequent calculation must take under consideration already done angular distance and is described by formula:

$$\phi_n = \sum_{i=1}^{n}(\omega_{Ai} \cdot \tau_{Ai}) - \sum_{i=1}^{n}(\omega_{Bi} \cdot \tau_{Ai}). \tag{7}$$

Above formula provide us with data set containing number of i values of angular shifts between shaft ends in domain of crank angle, what is the measure of torsional vibration. Figure 3 presents wavelets of instantaneous angular speed of two opposite discs and figure 4 presents torsional displacement of the shaft in domain of crank angle. It can be observed that the wavelet of the angular speed of both ends of the shaft are different and it is reflected by angular position of shaft's ends.

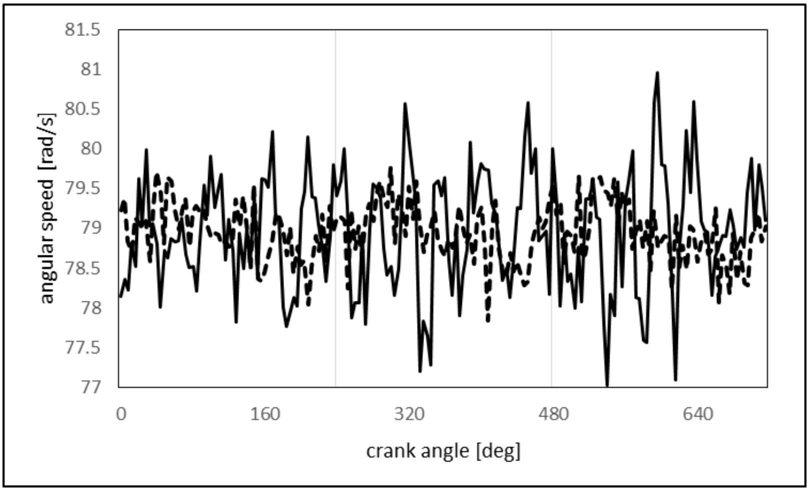


Fig. 2. Wavelets of instantaneous angular speed in domain of crank angle. Solid line refers to "master "disc A and dotted one to the disc B

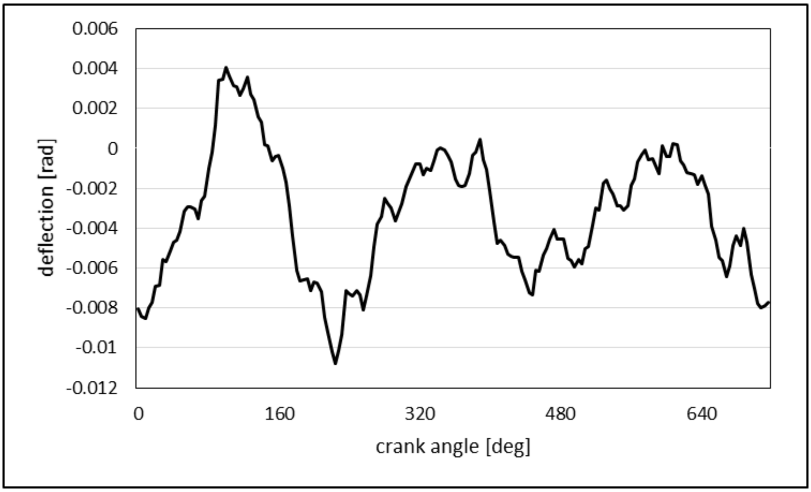


Fig. 3. Torsional deflection of the shaft in time of two revolutions (720° CA)

The form of the torsional displacement presented in figure 3 shows the character of excitation due to action of three cylinder. The peak values interval is 240° what is equal to ignition intervals of 3-cylinders, 4-stroke engine. Value of deflection together with data about shaft's dimensions and material enables calculation of share stress and twisting force.

In figure 4 are presented measurements discs and heads mounted at free end of the engine, and free end of the generator. Data collecting block starts measurement simultaneously for both disc what gives possibility of speeds comparison.

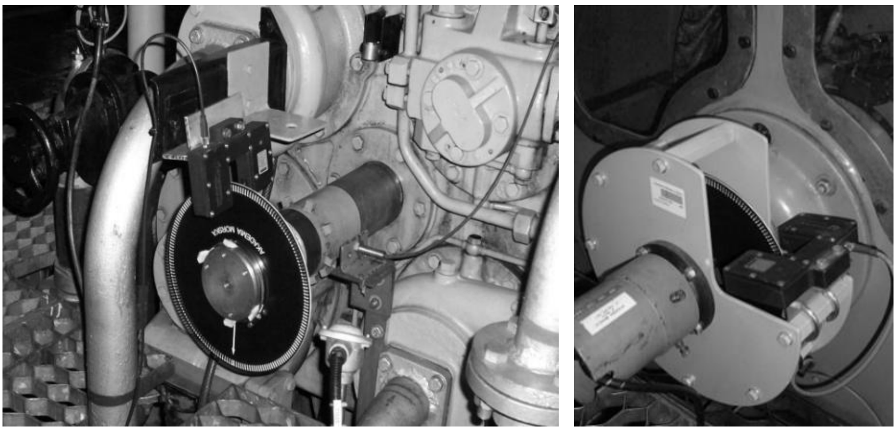


Fig. 4. Discs and measuring heads mounted at engine's crankshaft end

2. VERIFICATION OF METHOD'S RELIABILITY AND ACCURACY

Presented method has two hesitation points which has to be verified, and its influence at results of measurements must be clearly determined. The firs weak point is discrete nature of measurement discs. Having data presented in table 1, one can only calculate mean angular speed in distance of slot and tooth pair, what in our case is 2^o. Because of comparative nature of measurements, is absolutely necessary to verify level of potential deviations between data coming from both ends of the shaft. Despite of torsional character of revolutionary speed, average angular speed of two ends of the shaft must be equal and arithmetical sum of instantaneous torsional deflections must be close to 0, it means that level value of Φ_n for $n = 90, 180, \ldots$, i.e. for full revolutions, must be at level of single torsion. In order to verify above, several measurements under various loads were carried out. Results are presented in form of irregularity deviation index, calculated as proportion of differential value to mean value.

$$\gamma = \frac{2(\omega_A - \omega_B)}{\omega_A + \omega_B}, \tag{8}$$

where:

ω_A – angular speed of the disc A,
ω_B – angular speed of the disc B.

Obtained results of angular speed deviations between front and rear discs enable coming out with conclusion that error is very low and can be omitted (table 2).

Table. 2. Values of deviation index calculated for mean values of 1 revolution and 10 revolutions

Load	Angular speed A		Angular speed B		Deviation index γ	
	mean of 1 rev.	mean of 10 rev.	mean of 1 rev.	mean of 10 rev.	mean of 1 rev.	mean of 10 rev.
100 kW	79.13621	79.10985	79.14560	79.10509	**1.2E-4**	**6.01E-5**
140 kW	79.06436	79.04746	79.05816	79.04055	**7.8E-5**	**8.7E-5**
220 kW	78.94818	79.00442	78.91894	79.00359	**3.7E-4**	**1.05E-5**

Calculated values of angular distance according to formula 6, of discs A and B, are presented in table 3. It is worthy of underlining that after two revolutions (row number 2 in table 3), under every considered load, torsional displacement as differential value (A-B) is equal to null, what complies with four stroke engine's cycle of excitation.

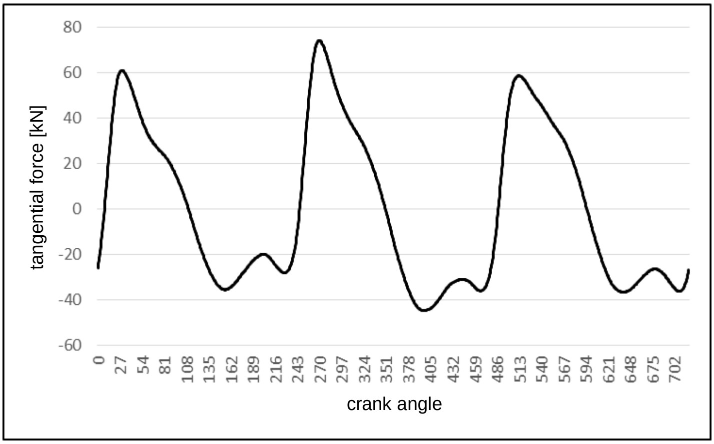
Above presented verification is evidence of correctness of implemented method of calculations, and proving that averaging of instantaneous angular speed based on 2^o sequence is sufficiently accurate.

Moreover, the shape of the torsion wavelet (fig. 3) can be compared with the model of excitation tangent forces function. The model wavelet of that force elaborated for three cylinder laboratory engine is presented in figure 5. Must be noticed the coherence between angular position of the shaft engine the peak of model wavelet of excitation force and maximum peak of measured torsion was high.

Table. 3. Comparison of calculated angular distance for subsequent revolutions

Engine's load	Number of subsequent revolution	Distance of disc A [rad]	Distance of disc B [rad]
100 kW	1	6.287	6.283
	2	12.566	12.566
	3	18.854	18.850
120 kW	1	6.278	6.283
	2	12.566	12.566
	3	18.845	18.850
220 kW	1	6.284	6.283
	2	12.566	12.566
	3	18.851	18.850

a)



b)

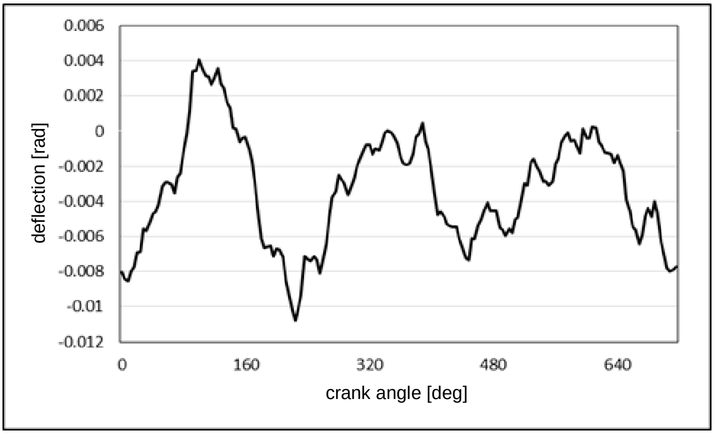


Fig. 5. Model of tangential force wavelet of 3-cylinder 4-stroke engine (a) and measured torsion wavelet (b)

CONCLUSION

1. Proposed method of torsional deviations measurement has big advantage due to easy way of mounting of measurement system elements. Both discs are very light and have moderate diameter. Installation of measurement heads at outer surface of engine body using simple grips is possible.
2. Data processing is not complicated and can be proceed with MS Excel program.
3. High Frequency of laser emitter gives high level of accuracy and ensure broad range of implementation looking from revolutionary speed point of view.
4. Proposed for disc elaboration width of slot and tooth giving measuring unit span covering 2 degrees was proved as sufficient, what gives typical configuration with 90 teeth and slots around disc's circumference.
5. Proposed method enable torsion analysis basing on one node mass model.
6. Further steps of method development shall be orientated on discovering relations between registered torsion variations and engine malfunctions, what shall enable formulating diagnostics conclusions [3].
7. Subsequent experiments shall relay on measurements of torsional deflection of the shaft working under simulations of fuel pump and injectors malfunction and determination of influence of irregularity of injection at torsional twists wavelets.
8. Interesting would be determination of influence of load fluctuation of electrogenerator as a source of torsional characteristic changes.

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