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EXPERIMENTAL DETERMINATION OF LOW SPEED DIESEL ENGINE CRANKSHAFT TWISTING

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Abstract

The exhaust gas emission control, together with the demand for specific fuel oil consumption reduction, poses a challenge for constructors. As a possible solution a very accurate fuel injection and combustion process control is considered. Such a control needs a precise timing and synchronization with every piston movement. The understanding of crankshaft deformation phenomena seems to be essential. In this work the experimental methodology of the crankshaft twist angle determination has been presented. The method is based on the cylinder pressure signal and an incremental encoder signal analysis. The presented results show the mean value of crankshaft twist angle under load. The momentary values are presented, too.

Keywords: *transport, marine diesel engines, engine control TDC determination, torsional vibration*

1. Introduction

Today's marine engines face the need of a very accurate control. The latest exhaust gas emission restrictions together with high demands for low specific fuel consumption force constructors to a very fine tuning of the combustion process. Such a tuning is impossible without a very exact control of fuel injection and the exhaust valve timing.

As today's marine engines are characterized by a very high power per cylinder the performance verification by means of cylinder indication becomes of utmost importance. In his work Polanowski [8] recognized a set of error sources of mean indicating pressure determination. Like other authors [2, 3, 6, 8, 10, 11] he defined the TDC determination error as the highest source among others.

The latest marine engines constructions based on the common rail idea (Wärtislä), or electro-hydraulic camshaft (MAN diesel A/G) utilize a precise crankshaft positioning for correct timing of fuel injection and valves timing and for engine's cylinder indication [7]. The positioning sensor is placed on one side of the crankshaft only, usually on its free end. It can be adjusted only when the engine is stopped to indicate static position of the crankshaft. The typical static crankshaft positioning process is done for one cylinder unit only. The other unit positions are calculated based on the shaft geometry and engine's firing order. Such a procedure has some disadvantages as the crankshaft twist due to loading torque and torsional vibration deformation are not taken into account [5].

The crankshaft deformation under load can be estimated with the aid of finite elements method [4]. However it is difficult to calculate it for a real ships' sea-going condition. The verification of calculations by means of experiment results is essential to obtain a correct mathematical model.

The experiment which should allow such verification was carried out on board an ocean going vessel. On the basis of the experiment the following assumptions were made:

- For every cylinder unit the thermodynamic loss angle between the real and thermodynamic TDC is identical under the same load condition,
- The crankshaft rotation speed within the revolution angle less or equal to 1 degree is constant.

The experiment consists in the comparison of the TDC indication read from the sensor mounted on the crankshaft's free end with the thermodynamic TDC read from the cylinder pressure course of every cylinder unit. As the thermodynamic loss angle [10, 11] was assumed constant, the systematic error should be identical for every cylinder unit and should not have influence on the results.

2. The propulsion plant and the measurement conditions

On board of the 1719 TEU container ship, a long stroke low speed MAN 6L70ME-C engine was installed as the prime move. The engine drives the ship's constant pitch propeller directly. The diagram of the propulsion plant is shown on Fig. 1a and its basic technical data are collected in Table 1.

Tab. 1. Basic technical data of the propulsion plant

denomination	value
Engine's nominal power	16980 kW
Engine's nominal speed	98.3 min^{-1}
Engine's firing order	1-5-3-4-2-6
Propeller's shaft diameter	579 mm
Propeller diameter	6700 mm

To eliminate the sea condition influence on the measurement results the experiment was conducted at the calm sea state. A full set of engine working parameters was collected during indication. An extract of the most important is presented in Table 2. The engine torque was measured with the aid of the Maridis torque meter and effective power was given by a genuine MAN digital indicator of PMI type.

Tab. 2. Engines working parameters during measurement

denomination	value
Engine's effective power	8044 kW (47.4% of MCR)
Engine's speed	77 min^{-1}
Engine's torque	997 kNm
Scavenge air pressure	2.12 bar abs
Fuel index	59%

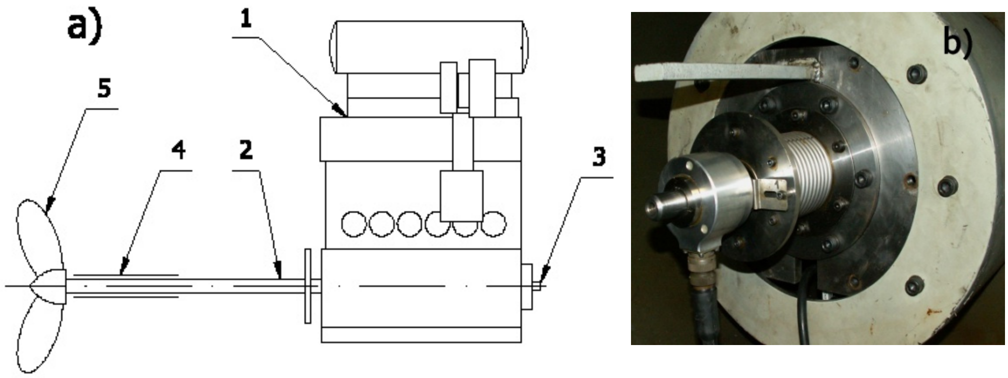

Fig. 1. The propulsion plant: a) Schematic diagram; 1-Main engine; 2-Propulsion shaft; .3-Rotary incremental encoder; 4-Stern tube with stern bearings; 5-Propeller; b) Encoder attached to the crankshaft

The main engine is of the electronic control type. Every fuel injection pump and exhaust valve are hydraulically actuated and controlled by designated digital multipurpose controller (MPC). The MPCs utilize the signal from an incremental encoder for a crankshaft positioning. The encoder is attached to the crankshaft's free end by means of bellows clutch (Fig.1b). The encoder's channel A generates 360 pulses of TTL standard per revolution and the Index channel generates one TTL pulse per revolution. The index pulse is shaped in such a way that it gives low signal for 180 degrees during the 1st unit's compression stroke and a high signal for 180 degrees during the 1st unit's compression expansion stroke. There was not available calibration data of the utilized encoder, so the assessment of the measurement accuracy was rather difficult. However there was made a test of accuracy for another model of incremental encoder of similar standard. On the basis of that test, one can assume that the utilized encoder generates pulses every 1 degree with a standard uncertainty of 0.0032 degree [9]. In Table 3 the utilized measuring instrumentation is presented.

Tab. 3. Measurement instrumentation data

Instrument	type
Measurement board	NI USB-6221, National Instruments, with counter input gate
Encoder	Incremental encoder ITD. 44 A 4 Y134, MAN B&W
Cylinder pressure sensor	F532A8-ACu, Optrand Inc.

During the experiment a simultaneous reading of two described channels from the encoder and in cylinder pressure was carried out. Every cylinder was scanned one by one in stationary working conditions.

3. Methodology

The collected data were processed in two steps. First the pressure signal was prepared, and then the signals were compared in the time domain.

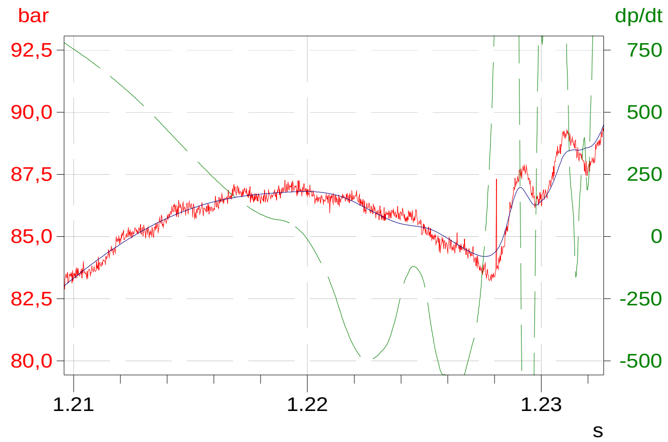

Fig. 2. The denoised signal(solid smooth line) and its differential (dashed line) compared to the raw signal (solid rough line).

Determination of the thermodynamic TDC was carried out by means of differentiation of the cylinder pressure signal. It was assumed that the zeroing of the first derivative of the pressure course reflects the thermodynamic TDC. As the raw cylinder pressure signal is disturbed by a noise of a different origin, the procedure of filtration had to be implemented. As the application of digital low pass filters causes shifting of the processed signal, a wavelet denoising procedure was applied. The Daubechies wavelet of the 7th order was chosen and a decomposition of the original signal into 8 levels was done [1]. On Figure 2 the results of denoising procedure compared to the original signal have been presented. The accuracy of the thermodynamic TDC determination should be no less that 0.05 degree. That could be obtained at a high sampling rate of the pressure signal and after processing with wavelet denoising procedure.

To calculate the angle between the encoder's TDC signal and the thermodynamic TDC the assumption was made that within 1degree of revolution the rotational speed of the crankshaft is constant. Then the number *n* of encoder's channel A full cycles between time t_3 and t_4 was determined (Fig. 3). As the time between consecutive pulses from channel A corresponds to 1 degree of crankshaft revolution, the fractional angle α_1 between the time t_2 and t_3 as well as fractional angle α_2 between the time t_4 and t_5 were calculated:

$$\alpha_1 = \frac{t_3 - t_2}{t_3 - t_1} \tag{1}$$

$$\alpha_2 = \frac{t_5 - t_4}{t_6 - t_4} \tag{2}$$

Then the angle α between encoder's TDC signal and the thermodynamic TDC is equal to:

$$\alpha = \alpha_1 + n + \alpha_2 \tag{3}$$

However the theoretical angle between the crank arms has to be taken into account:

$$\alpha_f = \alpha_1 + n + \alpha_2 - \delta_i \tag{4}$$

Based on the presented methodology the angle α_f for 7 to 9 cycles for every cylinder unit was calculated. The results are presented on Fig. 4.

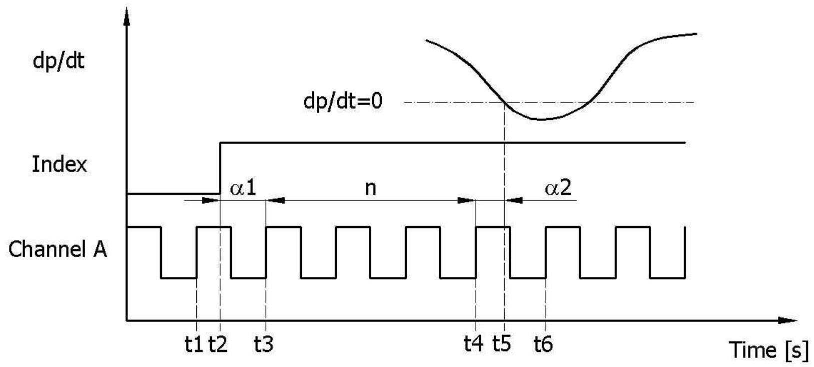

Fig. 3. The principle of data analysis; determination of the angle between encoder's index signal and thermodynamic TDC.

4. Conclusion

One can observe that the thermodynamic TDC mainly delays in relation to the encoder's indication. The mean delay is higher for the units which are placed further from the encoder – units 5 and 6 (Fig. 5). As the engine load during experiment was less than 50% the mean values of the shaft twist reach 0.2º only. However, the momentary values can reach close to 0.5º (Fig. 4). One can presume that according to the mechanics of materials theory the angle should be higher for the engine loaded with higher torque.

Tab. 4. Calculated correction factors for encoder's TDC signal for every cylinder unit at 50% engine load

Cylinder number	Factor value
1	-0.0353°
2	-0.0713°
3	-0.1073°
4	-0.1433°
5	-0.1793°
6	-0.2153°

The experiment shows that the momentary angle between the static and thermodynamic TDC is variable (Fig. 4). As the reason, the torsional vibration of the shaft can mostly be considered. However, the methodology of the thermodynamic TDC determination definitely influences the results, too. In the analyzed engine at 50% load the mean correction factors for the encoders TDC indication can be considered (Table 4). However, it should be kept in mind that the mean values solve the problem only partially. The method for correction of instant angle would be rather more suitable.

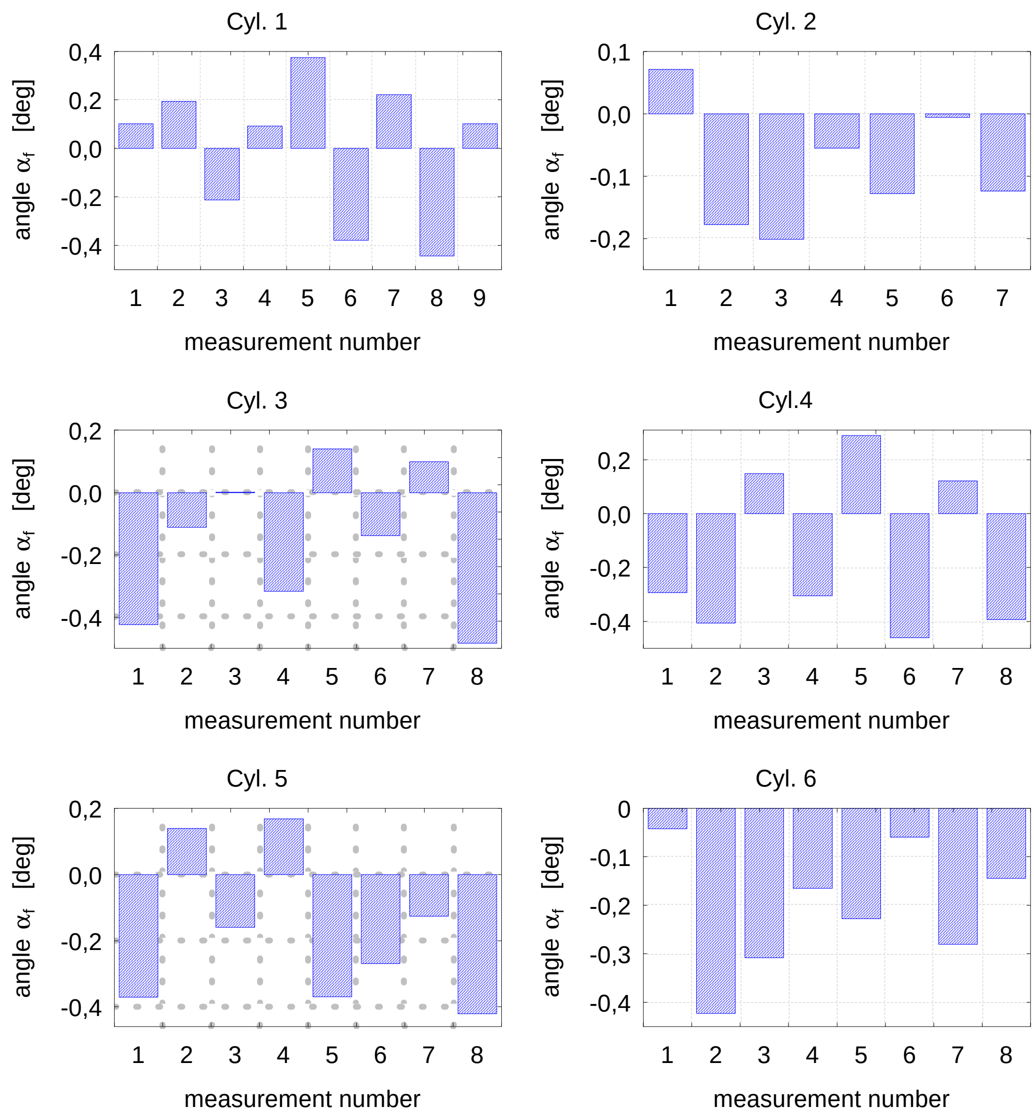

Fig. 4. The results of angle calculation. The values of the individual thermodynamic TDC and encoder's TDC differences in degrees.

The used wavelet denoising procedure has a great advantage compared to the typical low pass filters. The signal suffers no or minimal deformation what is of the uppermost importance when the signal is going to be differentiated. However the use of wavelets is usually time consuming and may be difficult to use for real time calculations.

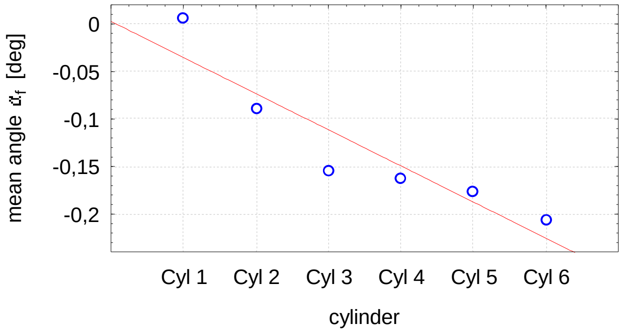

Fig. 5. The mean values of delay angle between the encoder's and thermodynamic TDC and their linear approximation.

Nomenclature

α_1 – fractional angle between the encoder's TDC signal and first rising edge from encoder's channel A

α_2 – fractional angle between the last rising edge from encoder's channel A and determined thermodynamic TDC

α_f – the resulting angle of thermodynamic TDC difference related to encoder's TDC

n – Number of encoder's channel A full cycles between time t_3 and t_4

t_1 – time when the last rising edge from encoder's channel A occurs before encoder's TDC signal.

t_2 – time when encoder's TDC signal occurs.

t_3 – time when the first rising edge from encoder's channel A occurs after encoder's TDC signal.

t_4 – time when the last rising edge from encoder's channel A occurs before thermodynamic TDC.

t_5 – time when the thermodynamic TDC occurs.

t_6 – time when the first rising edge from encoder's channel A occurs after thermodynamic TDC.

δ_i – theoretical angle between first cylinder crank arm and cylinder *i*, calculated from the firing order (δ_1=0°, δ_2=240°, δ_3=120°, δ_4=180°, δ_5=60°, δ_6=300°)

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