IN CYLINDER PRESSURE ASSESSMENT FROM MARINE DIESEL ENGINE

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Abstract

This paper explores the feasibility of using in-cylinder pressure-based variables in terms of certain class of errors, caused by indirect pressure measurement. Experimental direct in-cylinder measurement pressure setup was constructed, to make comparisons with standard indicator channel and indicator valve design. Zero-dimensional model was used for particular combustion system design. Subsequently, paper describes the application of a zero-dimensional combustion model for pressure signal analysis in cylinder head indicating passage of medium speed marine diesel engine. Recording of engine cylinder pressure development and its qualified evaluation is common aid for maintenance.

Especially the outline of experimental test-bed, the injector nozzle equipped with piezoelectric pressure transducer, measurement system specification, B type uncertainty –mean angular speed characteristic, wavelet "denoising" setting and signals comparison, engine cylinder indicator model channel, cylinder pressure traces influenced by engine speed and load, cylinder pressure traces influenced by engine load and speed with two different indicator channel lengths, Indicated engine cylinder power in motored mode of operation are presented in the paper.

Keywords: *marine diesel engine, in-cylinder pressure analysis, combustion pressure measurement*

1. Introduction

High demand to improve engine fuel economy and exhaust gas emission characteristics requires new tools for engine control, also development of engine diagnostic systems calls for improving the cylinder pressure measurement accuracy. One of the methods used to meet these requirements is a convenient and real-time cylinder pressure measuring system. Combustion pressure data is used for assessing engine performance, i.e. specific quantitative values - indicated mean effective pressure and maximum pressure. The final aim is a calculation of indicated power for each cylinder and total engine power. In addition, collected and calculated data expand the engine trend diagnostic system capabilities. Balanced combustion in reciprocating engines is important for reliable operation. Throughout the years, a number of engine balance techniques have been developed and utilized. Engine cylinders balance and tuning is carried out by means of valve and fuel injection timing as well as fuel rate and compression. Marine engine cylinder diagnostics is performed through the indicator valves and a long passage that is drilled in the cylinder head. Particularly, these engines are equipped with different, sometimes complicated cylinder pressure indicating passage system. It helps to maintain pressure transducer for mechanical reasons or for protecting from the harsh temperature, particulate and even combustion environment. In such cases – long measurement pressure pipe, might likely distort the pressure signals and even induce resonance. From the other hand, reliable calculation of mean indicated pressure and other performance data requires accurate simultaneous measurement of cylinder pressures and detection of the crankshaft position. Measurement of the cylinder combustion pressures shows substantial cycle-by-cycle variations. Unfortunately, the measurement of the cylinder pressure and piston position inside of a running engine, with easy to handle long lasting instruments is not a simple task. An important requirement for such pressure analysis is the ability to perform measurements with the highest accuracy of 0.1 bar in combustion phase and 0.1 deg, regarding crank angle position. In most cases however, cylinder pressure measurements generate data containing considerably greater errors. In order to minimise these problems, an analysis of the complete cylinder pressure measurement method was undertaken.

2. Experimental cylinder pressure system

Cylinder pressure measurements were carried out on a multi-cylinder marine diesel engine coupled to hydraulic brake – Fig. 1. This installation enabled keeping one cylinder under motoring operation. The idle engine cylinder combustion chamber was provided with two pressure transducers.

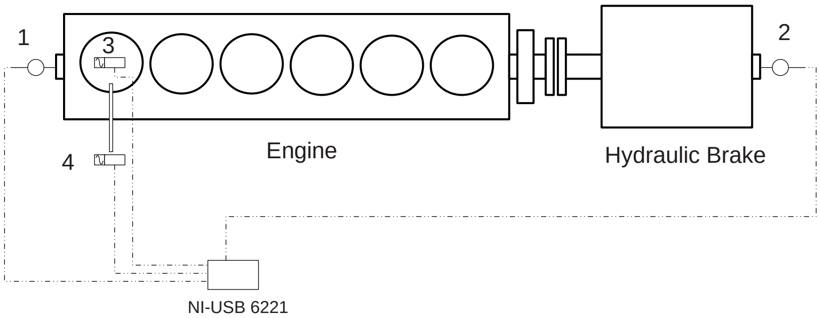

Fig. 1. The outline of experimental test-bed; (1) crankshaft encoder – free end, (2) crankshaft encoder – hydraulic brake, (3) combustion chamber pressure transducer, (4) pressure transducer at indicator pipe

The essential combustion pressure transducer was installed directly inside the engine combustion chamber. For that reason injector nozzle was adopted and miniature piezoelectric transducer has been inserted, as to replace the nozzle needle. Injector nozzle adaptation for straight combustion pressure measurement arrangement is shown in Fig. 2. The piezoelectric transducer system typically comprises of the quartz pressure element and charge amplifier. The other pressure transducer is placed in standard design on indicator cock.

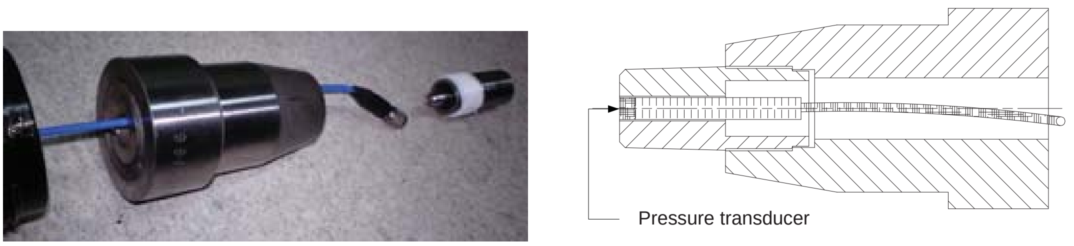

Fig. 2. The injector nozzle equipped with piezoelectric pressure transducer

The crank degree signals required were obtained from two shaft encoders providing a resolution of 0.1° and 0.5° respectively. Two encoders were installed on the test bed engine shaft-line. One of them was installed on the free end of the engine crankshaft and the other one on the free end of the hydraulic break. Both encoders were set in such a way that their reference pulses were generated simultaneously at the cylinder 1 and 6 in TDC position. To ensure the encoder index pulse was properly aligned with engine crankshaft and piston position supplementary settings and checks were done statically, according to the factory mark and manufacturer recommendation. Additionally, the static crosscheck of the TDC position was done. The accuracy of the static calibration was evaluated as 0.25 deg. During the principal experiment six signals: four from encoders and two from the pressure sensors, were recorded simultaneously by means of fast digital acquisition system. The system allowed for several engine cycles to be measured sequentially with fine resolution.

To develop relatively low cost PC-based data acquisition and analysis system, commercial DAQ - NI-USB 6221 board with the LabVIEW software were employed. Optical encoders triggers and clocks the acquisition board. Specification of experimental cylinder pressure system transducers is presented in Tab. 1.

Tab. 1. Measurement system specification

	Sensor	Maker	Assignment
1	Incremental rotary; M590	P.P.H. WObit	Engine crankshaft free end (fore-side)
2	Incremental rotary; PFI 60A0720TPT	Introl Sp. z o.o.	Hydraulic brake free end (aft-side)
3	Piezoelectric, 112B10	PCB Piezotronics, Inc.	Combustion chamber
4	Fiber-optic; F532A8-ACu	Optrand, Inc.	Indicator cock

3. Measurement and calculation methodology

3.1. The engine crankshaft speed evaluation

The incremental encoders have had a practical application in automation for many years. In modern marine engines they can be engaged as crankshaft angle position transmitters. In recent experiment a suitable angle measurement standard had to be chosen. The encoders used in measurement system are: 3600 and 720 pulses per resolution. The most adequate standard should be a high precision at a very small nominal value of the angle. Acquiring such a high standard and reading multiple samples for every position of the encoder line is very demanding. In consequence, the uncertainty of the crank angle measurement was scrupulously estimated. For crank angle base measurement, the 3600 pulse encoder was chosen. It was assumed that the encoder's shaft rotated at exactly 2π angle allowing for nominal amount of pulses to be counted. Since the mean value of a measured angle is equal to: $\bar{\alpha} = \frac{2\pi}{n}$, and it is assumed that the encoder's shaft rotates uniformly with constant angular speed $\bar{\omega}$, the angle scale can be transformed into a time scale. The advantage of that assumption is that the time scale is much easier to measure than angle scale. In such case, the angle of shaft's rotation comprising of *i* pulses is expressed as:

$$\alpha_i = T_i \cdot \bar{\omega} \quad . \tag{1}$$

The mean value of an angle $\bar{\alpha}$ can be determined by means of equation,

$$\bar{\alpha} = \frac{\bar{\omega}}{n} \cdot \sum_{i=1}^{i=n} T_i = \bar{\omega} \cdot \bar{T} \quad . \tag{2}$$

The difference between the value of a singular angle α_i, and the mean value $\bar{\alpha}$, can be recognized as the unknown value of the error,

$$\delta_i = |\alpha_i - \bar{\alpha}| \, . \tag{3}$$

If the set of values α_i is treated as *n* independent observations of the expected value $\bar{\alpha}$, the experimental standard deviation of singular measurement can be determined,

$$s(\alpha_i) = \sqrt{\frac{1}{n-1} \cdot \sum_{i=1}^{n} (\alpha_i - \bar{\alpha})^2} \, . \tag{4}$$

And the experimental standard deviation of a mean value can be expressed as:

$$s(\bar{\alpha}) = \sqrt{\frac{s^2(\alpha_i)}{n}} \, . \tag{5}$$

Those two quantities express, in turn, uncertainty of the angle α_i measurement. In order to get the full view of the measurement uncertainty, the systematic uncertainty of the measurement was evaluated. The ISO recommendations regarding determination of uncertainty type B were applied [1]. According to propagation of uncertainty rule, the same scale interval refers to both, singular value measurement T_i and to the average $\overline{T}$. That enables the equation (2) to be differentiated with respect to T_i and $\overline{T}$:

$$u_{\alpha} = \sqrt{\left(\frac{\partial\overline{\alpha}}{\partial T_i}\cdot\Delta T\right)^2 + \left(\frac{\partial\overline{\alpha}}{\partial\overline{T}}\cdot\Delta T\right)^2} \quad . \tag{6}$$

For a certain range of n values of T_i, the type B uncertainty u_{α} was determined and presented in Fig. 3. The systematic uncertainty of type B depends linearly on the angular speed. This is the result of constant scale interval of time measurement ΔT at decreasing period T_i. During experiments it is important to take into account the systematic uncertainty at the angular speed corresponding to the speed in specific application – reciprocating engine.

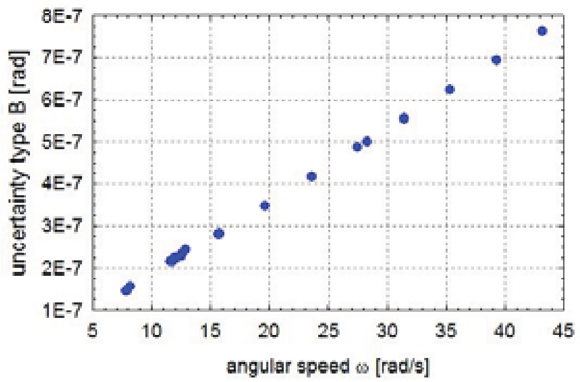

Fig. 3. Type B uncertainty –mean angular speed characteristic

3.2. The cylinder pressure assessment

Evaluation of in-cylinder pressure record offers a wide variety of information. However, piezoelectric transducers used for measuring cylinder pressure cause a drift in the pressure trace, i.e. the absolute pressure level is unknown and it is varying. This drift is so slow that it is considered to be constant during one engine cycle. The offset can be estimated with a range of methods. Here, the offset in the measured cylinder pressure is determined by comparing it to the intake manifold pressure. Due to standing waves in the intake runners and flow losses over the valves at certain operating points, the reference pressure value might be insufficient. This could be solved by including another estimation method, described in the next section. In the first stage of experiment, the cylinder pressure records were done at five different engine speed settings (364, 452, 540, 646 and 718 rpm) and idle engine load. Due to temporary cylinder retrofitting, only motoring mode of operation for cylinder No. 1 was available. The cylinder injection assembly consequently was cut off. Additionally, two records at 718 rpm and 25% and 50% of nominal load were performed in order to attain higher compression ratio and to establish the engine load influence on crank shaft dynamic twist.

For each recorded and analyzed engine revolution, the average angular speed was calculated based on the assumption that the rotating speed irregularity is small enough and can be neglected. Then, based on the time differences, the influence of the engine speed setting and load on the shaft-line (engine crankshaft, output shaft and brake impeller) dynamic twisting was estimated according to the formula:

$$\Delta\alpha = (t_i - t_w)\cdot\frac{360\cdot\bar{n}}{60}, \tag{7}$$

where:
$\Delta\alpha$ - shaft-line angle dynamic twist [deg],
$\bar{n}$ - average engine speed [rpm],
t_i - time of encoder 2 TDC position [s],
t_w - time of encoder 1 TDC position [s].

The characteristic cylinder pressure trace values were detected by means of differentiation of the pressure signal. The directly recorded pressure course was strongly influenced by noise and, consequently, was processed before differentiation. There are many unique methods of noise reduction and signal smoothing. Typically, a few types of filtering are applied. Whenever it is applicable, an approximation by a known theoretical curve gives excellent results. In presented experiment the "wavelet de-noising" procedure was chosen as the original signal suffered only minimum deformation. After a few tests the Daubechies family wavelet was chosen – DB 8 (Fig. 4). The wavelet toolbox of the Matlab software has been used for filtering.

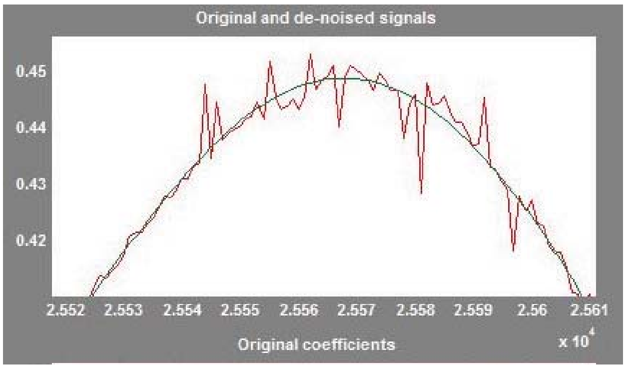

Fig. 4. Wavelet "denoising" setting and signals comparison

After de-noising, the pressure signals might be differentiated. The zero-crossing points of the derivatives in the vicinity of the static TDC were accepted as the thermodynamic TDC. Consequently, the angle between static and thermodynamic TDC was calculated.

3.3. The cylinder pressure and indicator channel flow modelling

A simple and efficient 0-D compression model was used for describing the cylinder pressure trace referred to as the polytropic model,

$$p(\Theta)(V_d(\Theta) + V_c)^n = C\ , \tag{8}$$

where:
p(Q) - cylinder pressure,
V_d (Q) - volume function,
V_c - clearance volume,
N - polytropic exponent,
C - constant.

The model describes the compression and expansion phase between inlet valve closing (IVC) and exhaust valve opening (EVO). In some cases at a particular location the stagnation pressure at the section is specified. For example, in engine cylinder indicator passage, where the inlet stagnation pressure and the exit pressure are fixed, the pressure ratio becomes a determining parameter for the flow. For this purpose, a relation for mass flux in terms of pressure ratio is derived and a simple form of steady flow energy for an isentropic flow is used. The relation expressing mass flux at any section of model pressure channel (Fig. 5) can be written in terms of the throat area and pressure:

$$\dot{m} = \frac{p_0 A_t}{\sqrt{\gamma R T_0}} \left[\frac{2\gamma^2}{\gamma - 1} \left(\left(\frac{p_t}{p_0} \right)^{2/\gamma} - \left(\frac{p_t}{p_0} \right)^{(\gamma+1)/\gamma} \right) \right]^{1/2} \tag{8}$$

where:
γ - ratio of specific heats,
R - gas constants,
A_t - channel throat area,
p_t - pressure at the throat,
p_0 and T_0 - stagnation conditions.

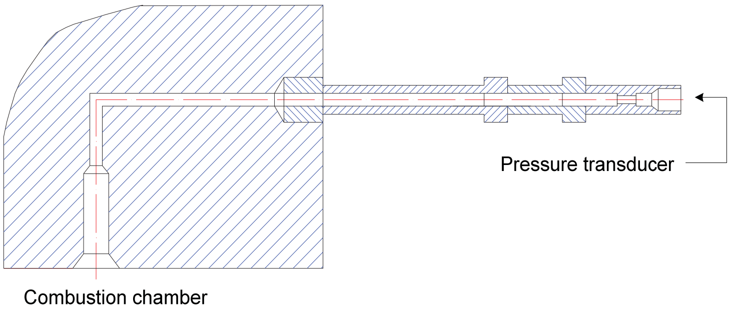

Fig. 5. Engine cylinder indicator model channel

4. Results and discussion

In the main part of the investigation process, the pressure sensor mounted on indicator pipe was considered a standard engine design option. The recording of in-cylinder pressure in the combustion chamber (straight admittance) by means of indicator pipe was carried out continuously during the whole process of experiment on the engine test bed. Additionally, timing traces were recorded for Top Dead Centre and crankshaft definition. The dynamic pressure data, measured at four different engine speed, was used to evaluate indicator passage correction method. For each engine condition, pressure data was compared as shown in Fig. 6.

To determine the passage frictional and volume effects, the gas flow is regarded as laminar, since gas velocity varies from zero at the transducer to the same value as inside the cylinder. As shown in Fig. 7, the summarized cylinder pressure trace angle is strongly influenced by engine speed and less so by the level of effective load.

The indicator passage of two different lengths was designed, where the pressure transducer was to be placed. As shown in Fig. 8, the results obtained by including the additional passage distance effects agree reasonably well with the cylinder pressure distributions measured directly inside the cylinder and at the end of indicator channel. This would indicate that for the standard indicator passage used in the experiment, the engine speed effect is substantial and have to be accounted for in the diagnostic procedures.

When congregated corrective angle functions examined it was possible to calculate the key parameters of the engine individual cylinder. Conventional calculation of IMEP has been done by numerical integration of the measured pressure (filtered) to ensure high accuracy. As a result, estimated values of the indicated power for single cylinder are presented in Fig. 9.

5. Conclusions

Recording of engine cylinder pressure development and its qualified evaluation is common aid for maintenance. However, is associated with serious pressure signal distortions and even if some thermodynamic properties of power cycle can be identified properly but accurate power estimation is affected. Experimental results achieved from marine medium speed engine, provide basic information on engine cylinder pressure development, measured straight in combustion chamber and by means of standard indicator pipe. The thermodynamic analysis of selected successive cylinder pressure data shows reasonable pressure deviation caused by indicator passage design. As a result, the fluctuation of pressure trace shift has been found acutely depended to engine speed – 3.3 deg within the 300 to 750 rpm range. Due to cylinder motored mode of operation, the measured pressure distortion against engine effective load were estimated only up to 50% nominal value and found as effective pressure trace shift ~ 1 deg.

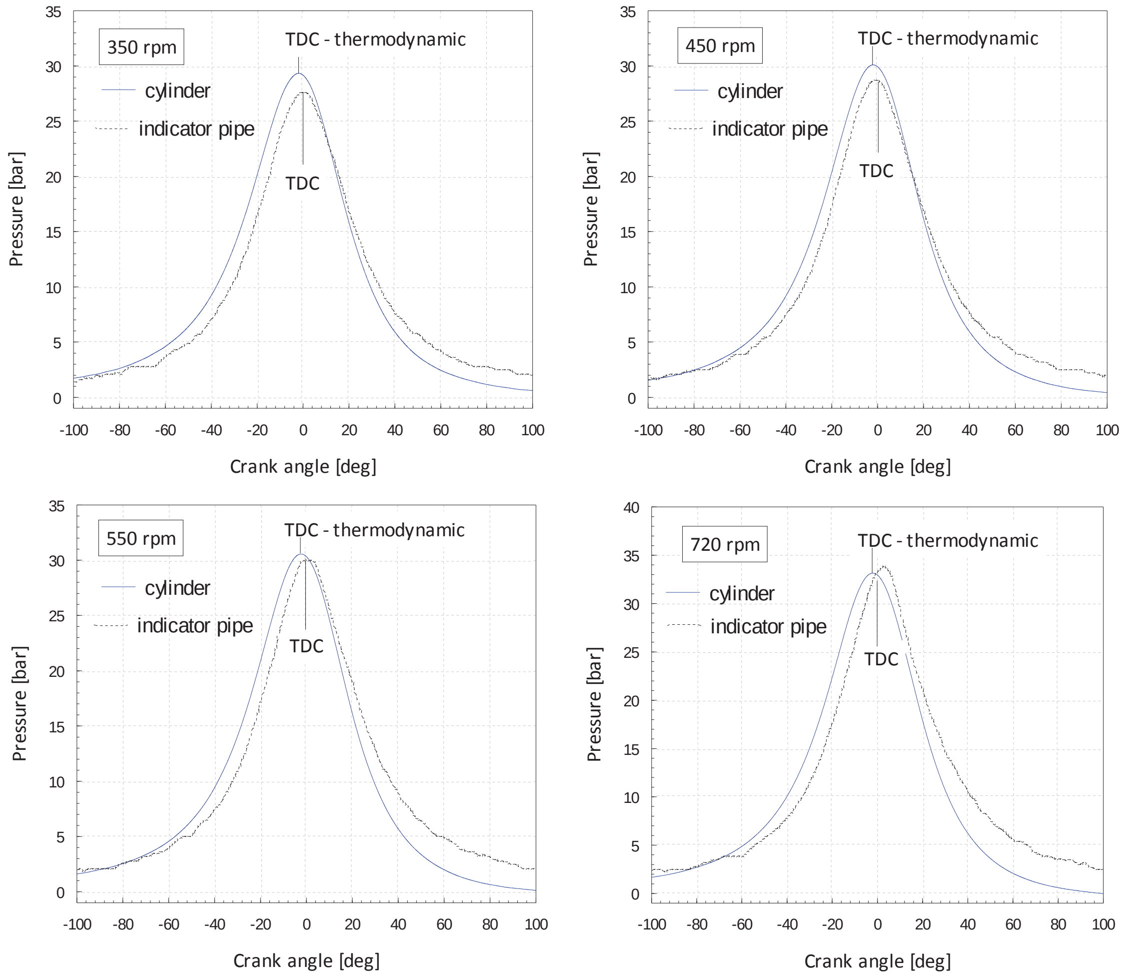

Fig. 6. Measured cylinder pressure comparison at different engine speeds

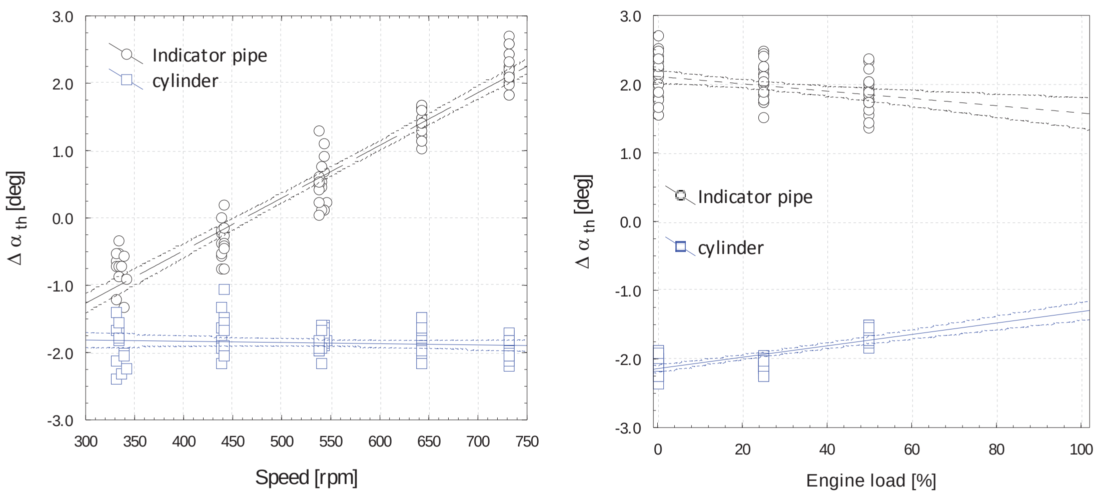

Fig. 7. Cylinder pressure traces influenced by engine speed and load

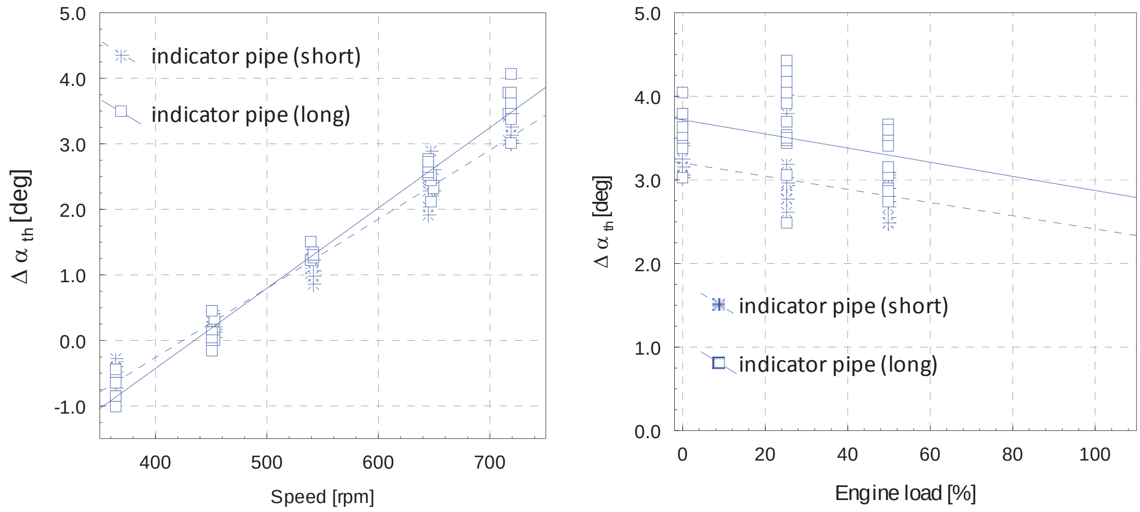

Fig. 8. Cylinder pressure traces influenced by engine load and speed with two different indicator channel lengths

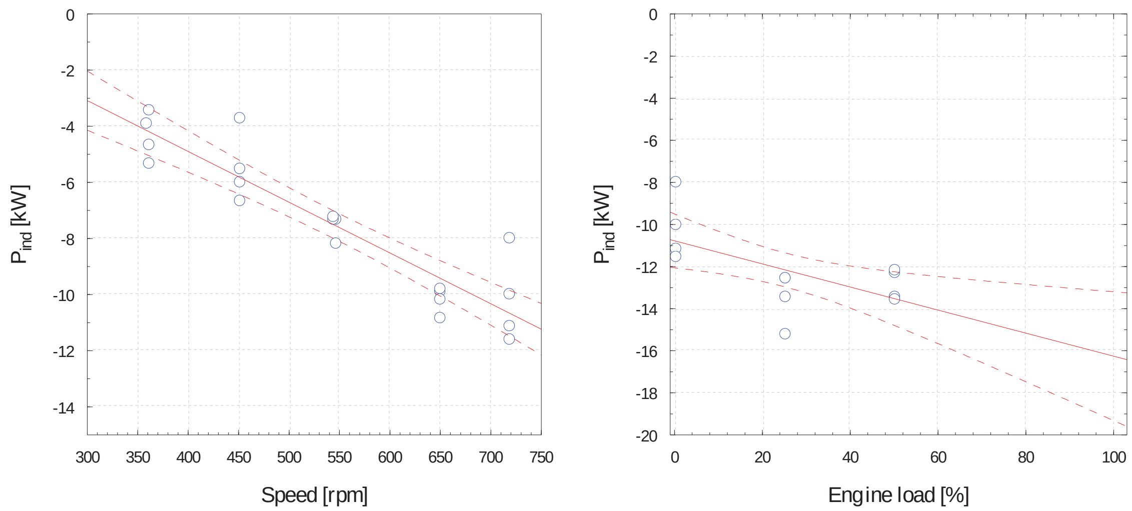

Fig. 9. Indicated engine cylinder power in motored mode of operation

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