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Methods of Real-Time Parametric Diagnostics for Marine Diesel Engines

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ABSTRACT

Using modern high-performance microcontrollers with wireless interfaces, built-in ADCs and low overall consumption, we develop a portable, real-time parametric diagnostic system for marine engines. The system is based on the use of modern Android/iOS gadgets that receive information from sensors via Bluetooth and then carry out the necessary calculations and display charts and data in real time. The system developed here uses a combination of a gas pressure sensor in the working cylinder and a vibroacoustic sensor, which expands the diagnostic capabilities of marine diesel engines under operating conditions. This solution allows for diagnosis of the fuel injection system, the valve train mechanism, and several other engine systems. In order to develop a portable diagnostic system for marine diesel engines, it is first necessary to solve the problem of analytically determining top dead centre (TDC), since such a system does not use special sensors for this. An algorithm for determining TDC is proposed here, based on an analysis of the measured pressure diagram rather than its derivative, which minimises the influence of digital and analogue noise. Our algorithm for determining TDC and subsequent data synchronisation is applicable in the absence of information about the actual compression ratio in the cylinder, which is a typical scenario for modern engines with variable valve timing. The algorithm also works under conditions of only approximate data on the charge air pressure, which are refined during the iteration process. A formula is proposed for determining the initial TDC position. Parameters for irregular operation of the engine are considered, and can be calculated in real time using time diagrams of pressure and vibration. Methods for expressly assessing the stability of the functioning of the main engine systems by monitoring and analysing a number of successive operating cycles are considered. To assess the unevenness of operation of the engine, a dispersion estimate of the deviations in the main parameters is used. To enable a comprehensive assessment of the engine stability in real time, the CII (cycle irregularity index) criterion is developed. The data processing methods described in this article provide an accurate estimate of the indicated power, due to the precise determination of TDC, thereby enabling an analysis of the stability of the operating cycles, optimal tuning of the engine systems, and monitoring of the results during operation.

Keywords: marine diesel real-time diagnostic; engine cycle parameters; vibration diagnostics; cycles irregularity index; injection and valves timing

INTRODUCTION

The strengthening of the requirements set out by the International Maritime Organization (IMO) is associated with strict restrictions on the amounts of toxic compounds emitted into the atmosphere during the operation of marine engines. The IMO has set a broad target in terms of decarbonising marine energy [1]; low-sulphur and alternative fuels therefore have become widely used in maritime transport, and special measures have been introduced to improve the energy efficiency of marine engines. Among the other approaches that have been developed to solve the problems set out by the IMO, one urgent task is to increase the efficiency of monitoring and diagnostics of the operating process of marine engines, which is directly related to the assessment of the quality of fuel combustion in the cylinders and the emissions of toxic substances during the operation of sea vessels [2,3].

A comparison of stationary and portable systems for the parametric diagnostics of marine engines shows that both have about equal numbers of working process parameters, with leading roles being played by the mean indicated pressure (MIP) and indicated engine power (IPOWER) [3,4]. However, one important difference between stationary and portable systems for marine engine diagnostics is the ability of stationary systems to operate in real time. Stationary systems include top dead centre (TDC) position sensors (for example, Type 2629DK [5]), which use quick data transfer from the time function into a crank angle function [4,6], thus allowing for computation in real time.

In modern portable systems, the position of TDC is computed by means of data analyses of the operating cylinder gas pressure [6-11]. This is why most modern portable systems for marine engine diagnostics operate in a delayed analysis mode, which involves sequential measurements of data on all cylinders, saving these data, calculating TDC, and then calculating the parameters of the work process in offline mode [4,11-13]. In the following, we summarise the features of the main parametric diagnostic systems that are currently used.

Probably the most carefully designed software in use is DK-20 Portable Analyzer (Doctor Analysis by Icon Research Ltd [14]). This system includes capabilities for analysing engine data and reporting on the results of engine indication. There an analysis foreseen for the propeller curves and comparison of factory tests and sea trials. The system is based on the use of a TDC apparatus sensor, meaning that preliminary preparation of the engine is needed. Setting of the TDC position is done with an accuracy of 0.1°CA (according to the pressure diagram step) by means of pressure derivative diagrams *P′(deg)* (TDC adjustment, Chapter 4 of [14]). We note that such a setting allows us to analyse the effect of an error in TDC on the calculated parameters for the working process using real measured data. This type of analysis illustrates the significant impact (up to 10%/°CA) of an error in TDC on the calculated value of IPOWER for some engine types, consistent with the findings in [4,6-11].

However, the engine analysis and diagnostics capabilities of the DK-20 Portable Analyzer system are limited by the use of only one gas pressure sensor in the working cylinder, which does not allow for a direct analysis of the fuel injection system and the mechanism for valve timing control. According to the authors, Doctor Analysis is the best software product for parametric diagnostics of marine engines when only gas pressure data is available for the working cylinder.

In portable systems such as EPM-XP and EPX-XPplus (IMES GmbH [12-16]), high-quality IMES pressure sensors [17] are used, which have high accuracy and stability under prolonged continuous monitoring of the working process. The stability and high accuracy of IMES sensors are partly responsible for the fact that these systems have become the most widely used method of diagnosis in the maritime fleet. These systems do not operate in real time.

LEMAG PREMET XL (Lehmann & Michels GmbH [18]) diagnostic systems are also widely recognised for marine engine diagnostics. These are robust and reliable diagnostic systems, primarily designed for operation on main low-speed engines due to their design features. LEMAG PREMET XL systems were the first to apply a graphical method of TDC correction, based on the principle that the rate of pressure changes near TDC *P′*(*deg*) has characteristics that are close to linear.

The present author has developed the idea of extrapolating the *P′*(*deg*) curve to determine TDC; this means that instead of using a linear model as in LEMAG, *P′*(*deg*) itself can be used in analytic form [3]. This method of finding TDC is used in DEPAS and EPX-XPplus (IMES) systems [15]. LEMAG PREMET XL diagnostic systems [18] are typical marine analysis systems in which a single gas pressure sensor is used in the working cylinder.

In MarPrime diagnostic systems (Maridis GmbH) [19], an ultrasonic sensor is employed alongside the gas pressure sensor in the working cylinder. This type of system finds TDC dynamically and then obtains the main parameters of the working process, including MIP and IPOWER, injection and valve timing, and valve leakages. In the present author's opinion, it is not very convenient to use diagrams created based on the MarPrime ultrasound sensor to determine injection and valve timing or for the diagnosis of mechanisms, due to the influence of significant noise.

A similar set of working process parameters is determined by the portable diagnostic system Depas 4.0HT (Depas Lab. ONMU) [4,11,13], in which a vibroacoustic sensor is employed in parallel with a pressure sensor. The Depas system employs the authors' custom algorithm for TDC determination, based on modelling of the *P′*(*deg*) curve during the compression stroke. According to the authors of the paper, Depas vibrograms allow for accurate determination of the injection and valve timing due to the use of a vibroacoustic signal processing algorithm, which involves digital filtration up to 20 kHz and demodulation of the obtained signal. Depas systems also include turbocharger diagnostics and enable diagnosis of additional engine systems [20,21].

The problem of determining TDC has been addressed by many authors [4,6-11], all of whom have provided data on the significant influence of an error in TDC on the calculated engine parameters. Professor Stanisław Polanowski has evaluated the impact of a ±0.3°CA error in the location of TDC on the calculated compression ratio in the cylinder as within ±10% [6].

Per Tunestal has offered a method for determining TDC based on a thermodynamic model of the gas inside the combustion chamber, and has provided data on the effect of TDC error within the range of up to 10%/°CA on IMEP and on the heat release characteristic in the cylinder [7]. In this work, he has demonstrated that the TDC error should not exceed 0.1°CA.

Emiliano Pipitone and Alberto Beccari used a special capacitive sensor to assess the accuracy of TDC determination [8], and reported that a 1°CA error in the position of TDC could lead to a calculation error of up to 10% in IMEP and up to 25% in the heat release characteristic. In [8], it was shown that the crankshaft position (and thus the volume inside the cylinder) needs be known to an accuracy of 0.1°CA, which is a challenging task.

Thermodynamic methods for determining TDC have been discussed in the works of Staś [9], Tazerout, Le Corre, Rousseau [10], and Varbanets [11]. All of these authors have emphasised the importance of precise TDC determination (0.1–0.3°CA) and the significant impact of an error in TDC on power calculation (6–10%/°CA).

In addition to the pressing task of determining TDC, another crucial aspect for real-time diagnostic purposes is identifying the stability criteria of working cycles. In a paper by Shi et al. [22], the significance of detecting the instability of the working cycle for diagnosis of the engine condition was demonstrated, and methods for calculating the cyclic variation characteristics for the parameter P_{max} were provided. In this study, the criterion for instability of the working cycle was determined overall for the entire working process diagram *P'*(*deg*). In this case, a separate analysis of the section of the diagram before the ignition point (valve operation stability) and after the ignition point (fuel injection system stability) is feasible. In addition, an integral criterion for the stability of the working cycle as a whole was proposed in this study.

The systems and methods for parametric diagnostics of marine engines described above operate on deferred analyses. Several tasks can therefore be identified that are relevant for real-time parametric diagnostics systems for marine engines. Real-time monitoring and diagnostics would enable the analysis of the working process and key parameters during operation and engine tuning.

Further development of portable parametric diagnostic systems for marine diesel engines will therefore require operation in real time. However, the implementation of such a system is possible only by solving a number of scientific and practical problems.

The first of these is the need for a noise-resistant and robust algorithm for TDC determination that is capable of operating with imprecise input data. Another crucial task is the development of criteria for assessing the stability of working cycles based on pressure and vibration time diagrams, thereby enabling the effective diagnosis of the performance of fuel injection systems and valve timing control. Modern high-performance microcontrollers with built-in ADCs and low overall power consumption [23] provide sufficient computational power to handle tasks such as data preprocessing and the preliminary calculation of certain working process parameters in real time.

Secondly, the diagnostic systems mentioned earlier rely on wired data transmission from sensors [13-15,18,19], which limits the mobility of the engineer during engine diagnostics and reduces data stability against electrical interference. The authors suggest that the use of wireless data transmission from sensors could address these issues.

Thirdly, it is worth noting that developers of diagnostic systems use built-in microcontrollers for parameter calculation, and the results are displayed on the built-in mini screens of the systems [14,15,18,19]. However, the computational capabilities of modern Android/iOS devices significantly surpass those of the microcontrollers used in diagnostic systems, and their display quality far exceeds that of the built-in graphical screens of existing diagnostic systems. In addition, end users of diagnostic systems are paying not only for sensors and measurement modules, but also for calculation and display modules, since virtually all engineers now have mobile devices.

According to the authors, a marine engine diagnostic system should be based on the use of modern Android/iOS devices that receive information from sensors via a wireless interface, and then perform the necessary calculations and display the results on high-quality displays in real time.

DESIGN OF A REAL-TIME DIAGNOSTIC SYSTEM

The concept proposed in this article for diagnosing marine diesel engines uses a cylinder gas pressure sensor, an additional vibration sensor, and wireless data transmission to enable real-time cycle diagnosis, as shown in Fig. 1. The temperature sensor depicted in Fig. 1 can be considered optional, but can be integrated into the real-time diagnostic system to determine the temperatures at key engine points.

The calculations and the display of indicator diagrams and process parameters (Table 1) are performed on an Android or iOS smartphone or tablet, as their current performance allows for complex calculations such as FFT data filtering, computing of the average indicator pressure, and phase analysis of vibration diagrams. The quality of the graphic display on modern gadgets is notably higher than that of the specialised built-in screens of the aforementioned diagnostic systems operating in postponed analysis mode.

Tab. 1. *Engine operating parameters determined in real time and in analysis mode*

Real-time quick scan mode	
RPM	Engine revolutions per minute, min^{-1}
P_{max} (P_z)	Maximum combustion pressure at corresponding crank angle, bar, °CA
COV_{Pi}	Coefficient of pressure variation (COVpi diagram), %
COV_{Pmax}	Coefficient of variation of peak pressure, %
CII_p	Cycles irregularity index (pressure), %
$\Delta\varphi_{adv}$	Variation in the fuel injection timing, *°CA*
$\Delta\varphi_{inj}$	Variation in the duration of fuel injection in the cylinder, *°CA*
$\Delta\varphi_{valve}$	Variation in the valve closing angles, *°CA*
Analysis mode	
Pmax, φPmax	Maximum combustion pressure at corresponding crank angle, bar, °CA
P_{comp} (P_c)	Compression pressure, bar
IMEP (MIP)	Indicated mean effective pressure, bar
IPOWER (N_i)	Cylinder indicated power, kW
$P_{ignition}$ (P_{ign})	Firing pressure and corresponding crank angle, bar, °CA
P_{exp} (P_{36})	Pressure at 36°CA after TDC, bar
Fuel Injection timing	Actual and geometrical fuel injection timing, °CA
Valve timing	Valve opening and closing timing, °CA
Fuel ignition delay	Ignition delay between fuel injection and combustion start, °CA, ms
DeltaG	Difference between actual and geometrical fuel injection timing, °CA

The use of Bluetooth data transmission would also resolve many issues associated with the inconvenience of wired data transmission during engine diagnostics. The emergence of controllers with built-in ADC modules and wireless data transmission capabilities has enabled the design of marine engine diagnostic systems that are capable of measuring and analysing engine data with steps of 0.1°CA (for four-stroke diesels) and 0.05°CA (for low-speed two-stroke diesels). A real-time system can therefore be designed for diagnosing a wide range of marine engines, as well as locomotive and shore station engines.

The values for the discretisation step mentioned above are currently adopted by most modern diagnostic systems, as this enables more precise phase estimation of the indicator parameters and, importantly, assessment of the detonation coefficient during fuel combustion in cylinders through an FFT analysis of gas pressure data *P*(*deg*).

The high performance of modern microprocessors, and the high ADC frequency (up to 500 ksps and above), built-in wireless data transmission module, and low total power consumption make it possible to design a universal marine diesel diagnostic system with autonomous power operation in real-time mode, as shown in Fig. 1.

The main advantage of the proposed marine engine indication and diagnosis system over the systems described above is the ability to adjust the engine during operation in real time. Real-time monitoring allows for control over the cylinder load (IPOWER), monitoring of the mean indicated pressure (IMEP), and the main pressure points P_{max}, P_{comp}, and P_{ign}, as well as evaluation of irregularities in the fuel delivery and fuel combustion processes in working cylinders, as shown in Table 1.

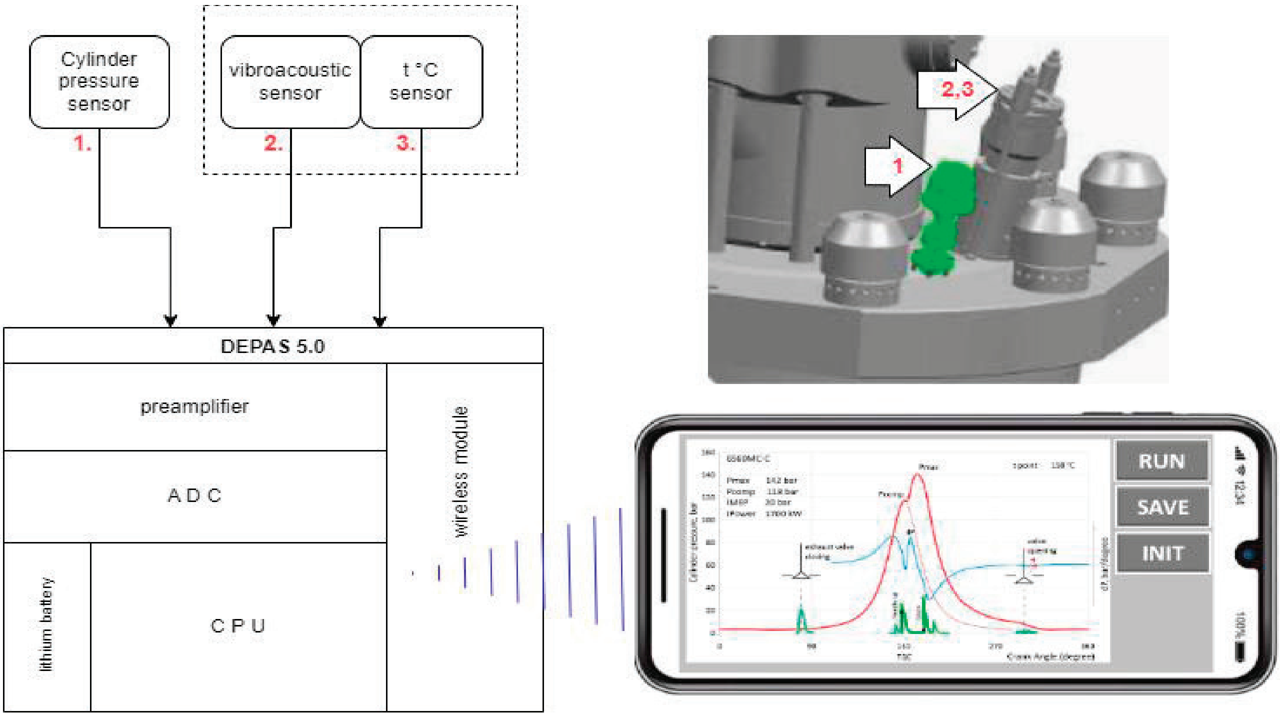


Fig. 1. Proposed real-time diagnostic system for marine diesel engines using a vibration sensor and wireless data transmission

The amplifiers for the sensor signal and a microcontroller are housed in a small box, along with lithium power sources and a charging controller. The computational and graphical functions are executed on a modern smartphone or tablet. In this way, an important problem is solved, in that control over the marine engine operating process is now available to a wide range of marine engineers. In fact, the engineer receives only the necessary set of sensors and one measuring unit. All calculations and diagram displays are performed on a personal device, which communicates via Bluetooth. This solution minimises the volume of the hardware involved and the potential cost of the diagnostic system.

METHOD OF DETERMINING THE PISTON TDC BY SOLVING THE EQUATION $PV^n = const$

Data synchronisation (i.e., the translation of pressure data in the cylinder from a function of time to a function of the crankshaft angle [3,10,11]) is often necessary when no information is available about the actual degree of compression in the cylinder $\varepsilon = V_a/V_c$, where ε is the compression ratio in the cylinder, V_a is the working volume of the cylinder, and V_c is the compression chamber volume. This is typical for modern engines with variable valve timing [1].

$$P(t, ms) \rightarrow P(deg, °CA) \tag{1}$$

The problem of analytically determining TDC has been addressed by many authors [3,10,16,17]. We note that for certain types of diesel engines, this problem is relatively straightforward. These are low-speed and medium-speed diesel engines with autoignition after TDC, as shown in Fig. 2. In this case, the curve of the rate of change in pressure during compression $dP/d\varphi$ crosses zero at the position of TDC (excluding minor thermodynamic shifts [3,4]). In other cases, the curve $dP/d\varphi$ closely approaches zero, as shown in Fig. 2, and the extrapolated curve intersects zero at the position of TDC, with a very small extrapolation error.

Two-stroke low-speed diesel engines are widely used in marine vessels as main engines, and for these, the problem of analytical data synchronisation in Eq. (1) is relatively easily solved. Other commonly used marine engines (such as two-stroke low-speed, and medium- and high-speed four-stroke engines) operate with initiation of ignition in the cylinder before TDC. In this case, the curve for the rate of pressure change $dP/d\varphi$ does not intersect zero at the TDC position. Extrapolating this curve using a mathematical model can introduce errors associated with significant noise in $dP/d\varphi$, which arise as a result of numerical differentiation of the pressure curve [3,8-15].

We propose to determine TDC and to solve the data synchronisation problem in Eq. (1) by analysing the recorded pressure curve, for which Eq. (2) holds true.

The method introduced below is based on the law of polytropic compression in the cylinder with a conditionally constant average value of the polytropic compression coefficient n [3,4]:

$$PV^n = const \tag{2}$$

where P and V are the pressure and volume in the working cylinder, respectively, and n=1.33–1.37 [3].

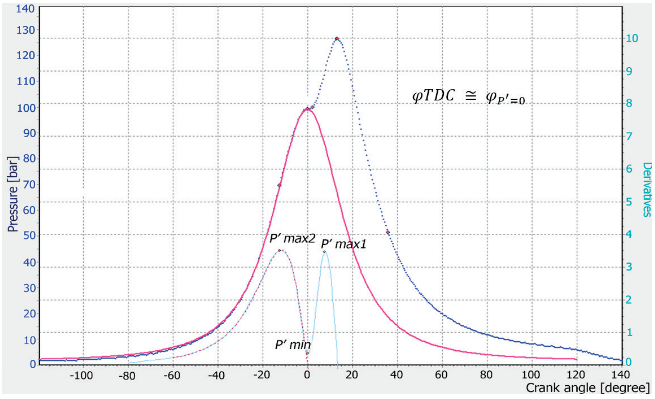


Fig. 2. Process of operation of a two-stroke low-speed diesel engine 6S50MC-C with fuel ignition after TDC

In addition to a lack of information about the actual degree of compression ratio (ε), the scavenging air pressure P_{scav} is often unknown or very approximately known (since the scavenging pressure has a small absolute value, it is usually measured with a large error in practice [10]).

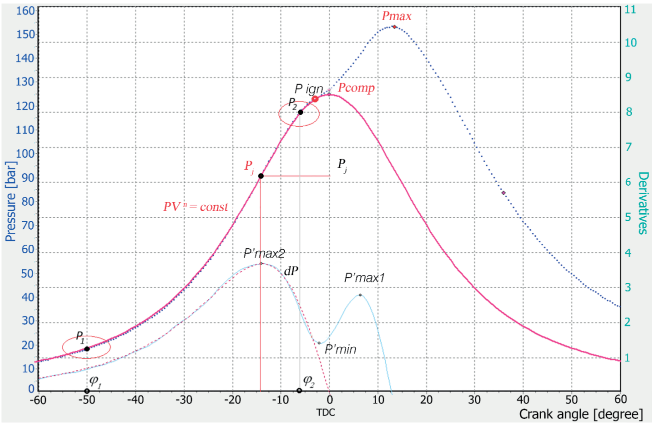


Fig. 3. Process of operation of a four-stroke YANMAR 6EY18ALW diesel engine with fuel ignition prior to TDC

Most modern marine diesel diagnostic systems designed for practical use [4,14,18,19] rely on uncooled sensors, in which the constant level changes when heated during measurement, meaning that they not intended for measuring constant pressures; in other words, they do not measure the scavenging air pressure. When engine indicating, P_{scav} must be measured by an independent sensor or set manually, which is inconvenient during diagnostics. An inaccurate setting of P_{scav} can significantly distort the calculation results and introduce errors into the calculated values of IMEP and IPOWER.

Thus, the task of determining TDC with a given accuracy and the subsequent synchronisation of data using Eq. (1) with unknown values for the parameters ε and P_{scav} is important.

Here, we consider solving this problem for the section $[P_1–P_2]$ on the compression curve, based on the assumption that the expression in Eq. (2) holds true with appropriate accuracy.

For any two points on the compression curve, we have

$$P_1V_1^n = P_2V_2^n = const \tag{3}$$

To eliminate the influence of noise on the pressure curve, the values of P_1 and P_2 are determined as the values of quadratic polynomials approximating the corresponding enclosed segments near points P_1 and P_2, as shown in Fig. 2:

$$P_1 = a_1\varphi_1{}^2 + b_1\varphi_1 + c_1$$
$$P_2 = a_2\varphi_2{}^2 + b_2\varphi_2 + c_2$$

where the coefficients a_1, b_1, c_1 and a_2, b_2, c_2 of the polynomials approximating the pressure curve are determined using the method of least squares on segments $\varphi_1 \pm 5–10°$CA, $\varphi_2 \pm 5–10°$CA depending on the noise level of the data (see highlighted segments in Fig. 2).

Given the expressions for the volumes of the cylinder during the crankshaft rotation phases φ_1 and φ_2, Eq. (3) can be written as:

$$P_1(V_{S1} + V_c)^n = P_2(V_{S2} + V_c)^n;$$

$$\frac{V_{S1} + V_C}{V_{S2} + V_C} = \sqrt[n]{\frac{P_2}{P_1}} \tag{4}$$

where V_c is the compression chamber volume, $V_{S1} + V_c = V(\varphi_1), V_{S2} + V_c = V(\varphi_2)$, and the expression for the current cylinder volume V(φ) is written as $V(\varphi) = V_c + \frac{\pi D^2}{4}s(\varphi)$, where D is the cylinder diameter.

$$s(\varphi) = R\left[\sqrt{\left(1 - \frac{1}{\lambda_{CR}}\right)^2 - k_p{}^2} - \frac{1}{\lambda_{CR}}\sqrt{1 - \lambda_{CR}{}^2\left(sin(\varphi) - k_p\right)^2} - cos(\varphi)\right]$$

where R is the crank radius;
$\lambda_{CR} = {}^R/_L$ is the connecting rod (or crank) ratio;
$k_p = {}^e/_R$ is the relative eccentricity;
e is the eccentricity of the crank [3,24].

We can denote the expression under the square root in Eq. (4) as $\sqrt[n]{\frac{P_2}{P_1}} = A$, and finally write the expression for the compression chamber volume

$$AV_c - V_c = V_{S1} - AV_{S2} \tag{5}$$

$$V_s = \frac{V_{S1} - AV_{S2}}{A - 1} \tag{6}$$

Thus, obtaining the pressures as a result of engine indexing at two points on the compression curve, P_1 and P_2, and approximate values of the initial phases φ_1 and φ_2 relative to the unknown TDC, enables us to determine of the compression chamber volume $V_C = f(V_{s1}, V_{s2}, A)$ using an iterative method, which will be necessary for further modelling of the compression curve in the cylinder.

$$P_1V_1^n = P_jV_j^n \tag{7}$$

$$P_j = P_1\left(\frac{V_{S1} + V_c}{V_{Sj} + V_c}\right)^n \tag{8}$$

Since temperature changes affect the constant level of the non-cooled pressure sensors used in modern marine diesel diagnosis systems [4,14,18,19], this constant level is subtracted during the measurements, thus excluding even an approximate estimate of the pressure at the beginning of compression P_a. The pressures on the compression curve P_1 and P_2, taking into account the unknown pressure at the beginning of compression, can then be recorded as follows:

$$P_1 = \bar{P}_1 + P_a \tag{9}$$

$$P_2 = \bar{P}_2 + P_a \tag{10}$$

where $\bar{P}_1$, $\bar{P}_2$ are the pressure values after subtraction of the constant temperature level.

As an initial approximation for P_a, a value of $P_a \approx P_{ign}/\varepsilon_{pass}$ can be used (see Fig. 2), where ε_{pass} is the passport or approximate value of the compression ratio in the cylinder. As an initial approximation of the position of TDC, the phase P_{max} can be conditionally accepted; the variable value of φ then includes the desired correction to the calculated value of the piston TDC:

$$\varphi = \bar{\varphi} + \Delta\varphi_{TDC} \tag{11}$$

The initial approximation $\Delta\varphi$ for TDC is determined using the method described below.

To build the compression curve model in Eq. (6), based on the polytropic compression equation in Eq. (2), we use the pressure data in the working cylinder obtained during engine indication with a subtracted temperature constant level $\bar{P}_j$ and with initial approximate estimates P_a and $\Delta\varphi_{TDC}$. Thus, the compression curve model P_j is a function of two unknown parameters, and $\Delta\varphi_{TDC}$:

$$P_j = f(P_a, \Delta\varphi_{TDC}) \tag{11}$$

Modelling is performed using the least squares method with Powell's non-gradient nonlinear n-parameter minimisation method [27]. The minimised functional is expressed as:

$$F = \sum_{j=\varphi 1}^{\varphi 2} \left[\bar{P}_j - P_j(P_a, \Delta\varphi_{TDC}) \right]^2 \rightarrow min \quad (12)$$

where φ_1, φ_2 are the boundaries used to model the section of the compression curve.

This method allows us to determine the minimum of a nonlinear function of n variables by conducting successive searches along a system of conjugate directions [28]. Powell's method does not use derivatives for the search, which is convenient for practical calculations. This approach is effective not only for quadratic functions, but also for general nonlinear n-parametric functions.

As a result of minimising the functional, we obtain a compression stage model [P_1–P_2] (see Fig. 2) constructed based on engine indication data, and most importantly, we obtain values for P_a and $\Delta\varphi_{TDC}$ with the specified accuracy, which enables us to refine the TDC position and solve the data synchronisation problem in Eq. (1) without requiring the compression ratio in the cylinder. The final analysis of the indication data, after solving the problem of analytical data synchronisation, allows us to determine MIP, IPOWER, and the main parameters of the working process shown in Table 1.

DETERMINING THE INITIAL APPROXIMATION OF TDC BY ANALYSING DP/DT DIAGRAMS

An initial value for TDC is necessary to implement the method described above for determining the position of TDC and the boost pressure P_a. However, rapid determination of the initial value of TDC, which is calculated from the temporal diagrams $P(t, \text{ms})$, is required for the operation of marine diesel engine diagnostic systems in real time, as shown in Fig. 1. In this section, we introduce an empirical formula for determining TDC, based on an analysis of *dP/dt* diagrams obtained by numerical methods and processed by a digital filter [4,16], as shown in Fig. 4.

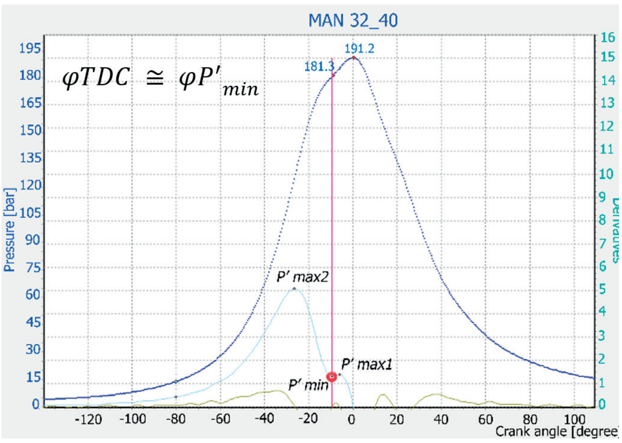
The initial phase of TDC is conventionally accepted as the point P_{max}, and the displacement to the initial approximation is calculated; the position of TDC is then refined by the method described above. An analysis of data obtained from marine engines using IMES sensors [17] and from working process models obtained in the Blitz-PRO environment [24] showed that the relationship between the magnitudes and corresponding phases of the extrema of the pressure change rate diagrams in the working cylinder enabled the initial approximation of TDC to be determined. An analysis of marine engine indication data of different types [3,4] showed that in most cases, depending on the nature of the working process and the type of indicator diagram, the TDC position lies between the phases [$\varphi P'_{min}$, P'_{max1}] of the minimum of the first derivative and the maximum of the first derivative of the combustion stroke.

The phase relationships between the maxima of the first derivatives during combustion $\varphi P'_{max1}$ and compression $\varphi P'_{max2}$, and the position of TDC, are experimentally obtained as shown in Figs. 4 and 5. It is found for various types of marine engines that in the case of 'soft' combustion, where the rate of pressure change during combustion is lower than during compression, the position of TDC is closer to the phase of the minimum of the first derivative $\varphi P'_{min}$, which occurs between compression and combustion, as illustrated in Fig. 4(a) (red point).

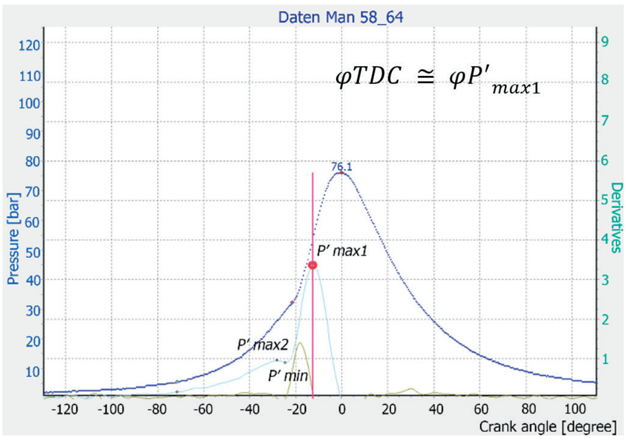
For engines with 'intense' combustion, where the maximum of the first derivative during combustion is greater than during compression, the position of TDC is closer to the phase of the maximum derivative during combustion, $\varphi P'_{max1}$, as depicted in Fig. 4(b) (red point). Two extreme cases are identified depending on the ratio of the magnitudes of P'_{max1}/P'_{max2} (these ratios are empirical, and need to be specified for a given type of engine).

$$\frac{P'_{max1}}{P'_{max2}} < 0.5 \rightarrow \varphi\, TDC \cong \varphi_{P'min1} \quad (13)$$

$$\frac{P'_{max1}}{P'_{max2}} > 3.0 \rightarrow \varphi\, TDC \cong \varphi_{P'max1} \quad (14)$$



a)



b)

Fig. 4. Initial approximation determination of VMT for (a) MAN 32/40 and (b) MAN 58/64 engines

For modern engines, the ratio of the rates of pressure change during combustion and compression P'_{max1}/P'_{max2} may be less than 1.0; in other words, the rate of pressure increase during combustion is less than during compression, which characterises 'soft' combustion [3] without significant dynamic loads on the crank-connecting rod mechanism and bearings, as shown in Fig. 4(a). In this case $\varphi TDC \cong \varphi P'_{min}$ In general, the relationship between the position of TDC and the extrema of the first derivative can be represented by the following empirical formula:

$$\Delta\varphi_{TDC} = \varphi_{P'min} + \frac{\varphi_{P'max1} - \varphi_{P'min1}}{\theta_1 - \theta_2} \cdot \left(\frac{P'_{max1}}{P'_{max2}} - \theta_2\right) \quad \textbf{(15)}$$

where θ_1=[3–5],θ_2=[0.3–0.9] are empirical coefficients; $\varphi_{'min}$ is the minimum phase of the rate of pressure change between the maximum phases of compression and combustion; $\varphi_{'max1}$ is the phase of the minimum rate of pressure change during fuel combustion; and P'_{max1} and P'_{max2} are the maximum rates of change in compression and combustion pressure.

From Table 2, it can be seen that in most cases, the initial determination of TDC has a relatively low absolute error, and that this approach can be applied for real-time monitoring. One advantage of calculating the initial approximate value of TDC is the high calculation speed, due to the need for preliminary determination of the extrema of the first derivative from the pressure curve. A further advantage of this algorithm is its low sensitivity to noise in the first derivative of pressure experimental data. Although one disadvantage of this approach is the need for tuning of the coefficients θ_1, θ_2, we note that once tuned, these coefficients can be subsequently used for the main operating modes of the engine, therefore allowing for real-time analysis of the pressure diagram in the cylinder.

The calculated results for the initial approximation of TDC for several engines in which different methods are used to organise the work processes are shown in Table 2 and Figs. 4 and 5.

Tab. 2. Determination of the initial approximation of TDC

Engine: Weichai CCFJ40Y-W, θ_1=1, θ_2=0											
φ bTDC, °CA	P'_{max2}	°CA	P'_{max1}	°CA	$\frac{P'_{max1}}{P'_{max2}}$	P_{min1}	°CA	*Δ'TDC**	$\Delta\varphi_{TDC}$	*φTDC*	*ΔTDC*
6 *bTDC*	172.49	348	634.2	359	3.68	49.03	358	2	3.68	361.68	1.7
8 *bTDC*	172.15	348	580.2	358	3.37	70.4	357	3	3.37	360.37	0.4
10.4 *bTDC*	171.95	348	588	356	3.42	110	355	5	3.42	358.42	−1.6
12 *bTDC*	171.3	348	1110	354	6.48	144.9	353	7	6.48	359.48	−0.5
										359.99	**0.16**
Engine: WUXI 277 rpm (5, 7, 10, 12 bTDC), θ_1=5, θ_2=0.3											
φ bTDC, °CA		°CA		°CA	$\frac{P'_{max1}}{P'_{max2}}$	P_{min1}	°CA	*Δ'TDC*	$\Delta\varphi_{TDC}$	*φTDC*	*ΔTDC*
5 *bTDC*	80.85	344	229.8	365	2.84	15.68	358	2	3.79	361.78	1.8
7 *bTDC*	80.85	344	261.8	364	3.24	32.81	357	3	5.00	361.0	1.0
10 *bTDC*	80.85	344	317.64	361	3.93	54.67	355	5	6.18	359.18	−0.8
12 *bTDC*	80.85	344	358.88	359	4.44	65.78	353	7	7.04	358.04	−1.9
										360.0	**0.0**
Engine: Wartsila 16V32, θ_1 = 3, θ_2=0.3											
% iPOWER [φ_{inj}, °CA]		°CA		°CA	$\frac{P'_{max1}}{P'_{max2}}$	P_{min1}	°CA	*Δ'TDC*	$\Delta\varphi_{TDC}$	*φTDC*	*ΔTDC*
0 [2, 14.5]	299.3	346	308.13	366	1.03	0.364	359.8	0.2	1.68	361.48	1.5
+3% [4, 15]	299.99	346	414.13	364	1.38	70.18	358	2	2.40	360.40	0.4
+5% [7, 16]	300.47	346	518.62	361	1.73	171.79	355	5	3.17	358.17	−1.8
										360.02	**0.0**

$* \ \Delta'TDC = \varphi_{Pmin1} - 360°$

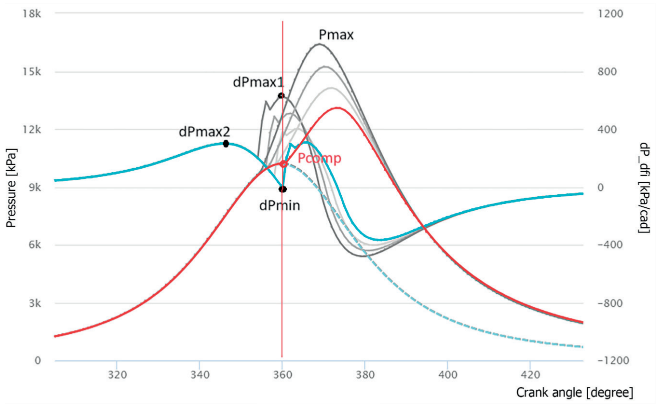


Fig. 5. Working process of the Wartsila 16V32 engine with different fuel injection advance angles

DETERMINATION OF CYCLE-TO-CYCLE IRREGULARITY

One of the tasks of a real-time diagnostic system for marine diesel engines is to assess the unevenness in the operating cycles in real time. When analysing a series of indicator diagrams of one cylinder, changes in the peak pressure values (P_{max}) are primarily seen, and with a more detailed analysis, changes in compression end pressures (P_{comp}) and mean effective pressures (*IMEP*) can be observed. In fact, changes occur from one cycle to the next throughout the indicator diagram curve *P(φ,* °CA*)* or *P(t,* ms*)*. Unevenness in the operating cycles is mainly caused by the operation of the systems that are responsible for the quality of fuel combustion in the cylinder, i.e., the high-pressure fuel system and the valve timing control system.

Unevenness in the working cycles is a highly undesirable feature of marine diesel engines, since it leads to the following negative consequences:

– Increased overall levels of vibration of the engine;
– Uneven distribution of mechanical loads on the cylinders;
– Overloading of individual cylinders.

As a general consequence of these factors, there is a decrease in the effective efficiency and limitations on the power of the engine [3,33,34].

The uneven distribution of loads across cylinders leads to increased concentrations of toxic emissions in exhaust gases, which is currently strictly regulated by IMO requirements [3,25,29-34].

To reduce engine vibration and minimise the emission of harmful substances, it is necessary to control the irregularity in the operating cycles during engine operation. In the following, we identify some criteria of irregularity that can be obtained and analysed during engine operation in real time. Unlike other engine parameters (IMEP, IPOWER, etc.), an analysis of irregularity criteria can be performed via a 'quick scan' of the engine in real time, as the temporal diagrams *P*(*t*, ms) can be used for this purpose. It can be said that evaluation of the irregularity in engine operating cycles is the main task of a real-time diagnostic system for marine diesel engines.

One of the main reasons for cycle-to-cycle variations is irregularity in fuel delivery, which can be caused by cycle-to-cycle variations in the injected fuel dose, injection timing angle $\Delta\varphi$, and injection duration $\Delta\varphi_{inj}$, as shown in Fig. 6. In this case, changes occur in the main parameters of the working process, including IMEP and IPOWER (Table 1) for each cycle, thus reducing the operational efficiency of the engine. As it is approximately proportional to the percentage change in cycle fuel delivery, the indicated power of the cylinder changes, thus increasing the level of vibration and decreasing the effective efficiency of the engine.

By recording and analysing the pressure in the cylinder over several consecutive working cycles, a real-time system can compare the differences in the parameters during the compression and combustion stages, and hence assess these differences numerically. Numerical analysis of vibrograms during fuel injection and valve closing stages can also be performed in real-time mode. A comparative analysis of the vibrograms during the compression stages enables an evaluation of the unevenness in the operation of the valve mechanism, whereas a comparative analysis during the combustion stages enables an assessment of the irregularities in the operation of the high-pressure fuel equipment.

Assessment of the unevenness in the working cycles as a diagnostic parameter should be carried out under a constant engine load. Since the load varies constantly due to external conditions during the operation of marine diesel engines, it is necessary to select stable periods of operation when assessing unevenness and to take into account the engine runtime [3,4].

Let us consider a certain number of indicator diagrams *m* of working processes (j = 1,…,*m*), recorded under a constant load of a main S60MC-C engine, as shown in Fig. 6. The maximum pressure at each *j*-th cycle P_{max} and the maximum average pressure over *m* cycles $\overline{P_{max}}$ can be written as:

$$P_{max\,i} = \max(P_{ij});\ \overline{P_{max}} = \frac{1}{m}\sum_{j=1}^{m} P_{max\,ij} \quad \mathbf{(16)}$$

where *i*=1,...,*N* is the number of points in one cycle when recording an indicator diagram; *m* is the number of recorded cycles; and *j* is the cycle index.

In most modern marine diesel diagnostic systems, step recording is used for pressure diagrams, with values of Δ*P* = 0.1°CA for four-stroke engines and Δ*P* = 0.05°CA for low-speed two-stroke engines. Such a small step allows for an analysis of the working process and its irregularity with the maximum degree of detail, and for an evaluation of the detonation coefficient during fuel combustion using spectral analysis methods [12,13,16,21].

To assess the unevenness in the working cycles, we use the popular coefficient of variation (COV) criterion [33,34], which can be applied to the entire pressure curve, thus obtaining a graph of COV_{pi} values as an estimate of the unevenness in the engine operation. We can express the coefficient of variation of working cycles as a percentage that is equal to the ratio of the standard deviation to the current mean value of the selected quantity (in this case, pressure), as follows:

$$COV_{Pi} = \frac{\sigma_{pj}}{\bar{P}_j} * 100\ \% \qquad (16)$$

where $\bar{P}_i = \frac{1}{m}\sum_{j=1}^{m} P_{ij}$ and the standard deviation σ_{pj} is calculated as:

$$\sigma_{pj} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(P_i - \bar{P}_i)^2} \qquad (17)$$

$$COV_{P.max} = max(COV_{Pi})\ \%;\ \overline{COV_P} = \frac{1}{N}\sum_{j=1}^{N} COV_{Pi}\ \% \qquad (18)$$

It is possible to formulate a complex criterion for the irregularity of the operating cycles based on the pressure, referred to here as CII. This generally characterises the cyclic irregularity of the engine, and is defined as follows:

$$CII = \sqrt{\overline{COV_P} * COV_{P.max}}\ ,\ \% \qquad (19)$$

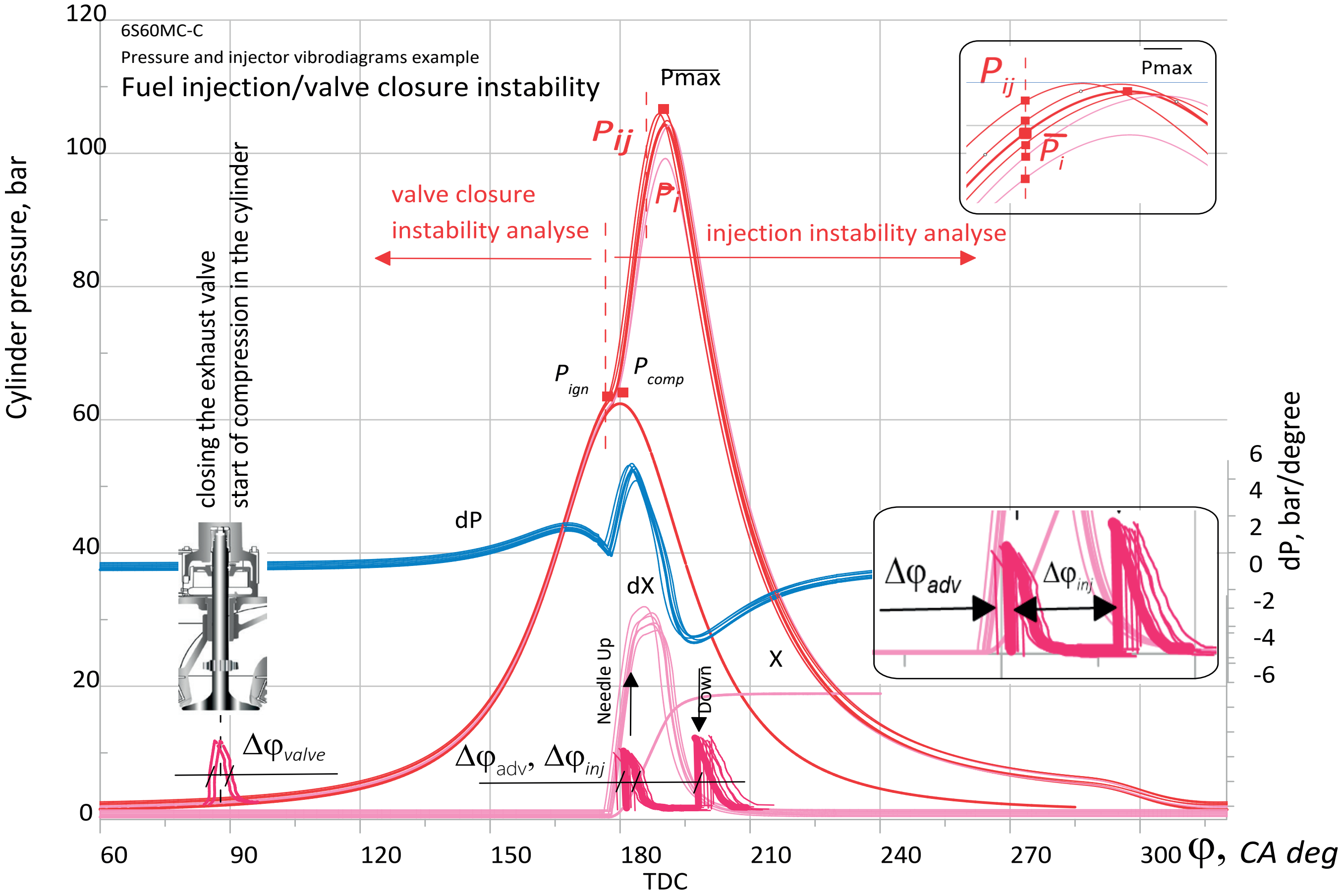


Fig. 6. Irregularity in the working processes of an S60MC-C low-speed engine based on pressure diagrams and vibrodiagrams

We propose to calculate and analyse the maximum value $COV_{P.max}$ [33]. In our opinion, in addition to $COV_{P.max}$, it is also possible to analyse the current (diagram COV_{Pj}) and average $\overline{COV_P}$. From a chart of the current values of COV_{Pj}, the part of the diagram representing the greatest deviation of the operating cycles can be observed, and depending on the results, a conclusion can be drawn about the irregularity of fuel injection or the irregularity of the gas distribution valves (Fig. 7).

CII is expressed as a percentage. As shown by a numerical analysis of computational and experimental data (Fig. 8), it is a convenient means of analysis, and allows for an overall characterisation of cycle-to-cycle variability.

Fig. 6 and Table 1 show the results of a numerical analysis of three specified criteria for several types of engines: a low-speed two-stroke MAN S60MC-C, a four-stroke WUXI ship main engine, a four-stroke ship diesel generator Wartsila 16V32, and two laboratory four-stroke engines: a Deutz 226 (manufactured in 2021), and an NVD24 (manufactured in 1970).

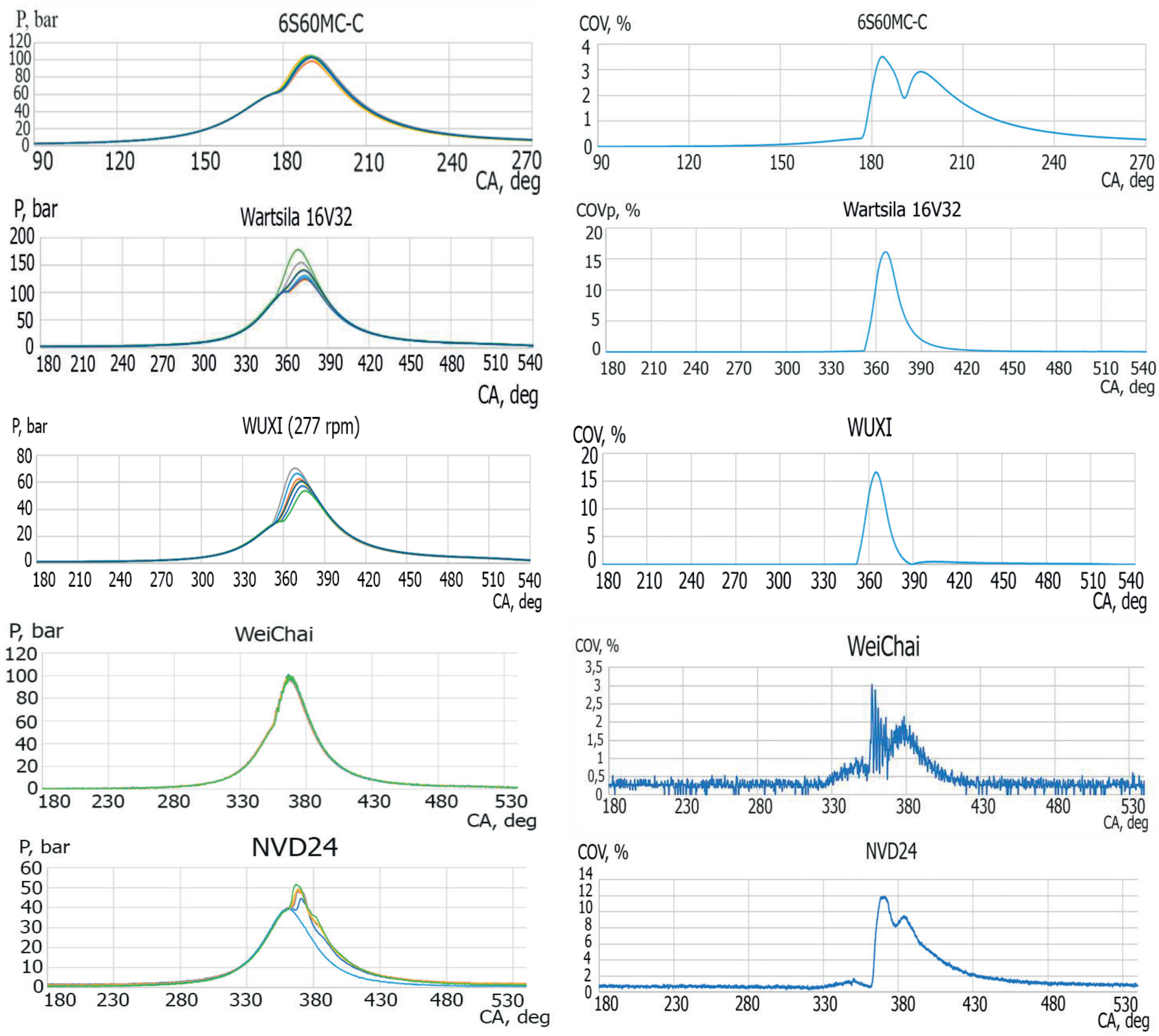


Fig. 7. Indicator and COV diagrams for five different types of engine

This analysis revealed significant cyclic irregularity in the old NVD24 laboratory engine and, as expected, low cyclic irregularity in the new WeiChai engine (Deutz 226). When analysing the operating cycles of the WeiChai, the values of $COV_{P.max}$, $\overline{COV_P}$ and CII_p were found to be significantly lower than the values of the same metrics for the other engines. A joint analysis of these three metrics showed that the generalised criterion CII_p is a good overall qualitative assessment of cycle-to-cycle variability, and can be applied in real time to evaluate the irregularity in the engine operation under load (Fig. 8).

An assessment of the fuel injection irregularity in the engine cylinder can be carried out by analysing the phase fronts of the vibration signal of the injector, as shown in Fig. 6. The principle used to obtain and analyse vibration diagrams for the injectors and gas distribution valves has been published in several sources [4,12,13,16]. Since the pressure and vibration signals are measured simultaneously, after aligning the working cycles, it is possible to estimate the dispersion of the phase spread of the front fronts of the injection vibration diagram. φ_{adv} and $\Delta\varphi_{inj}$ (Fig. 6).

Tab. 3. Comparison of COV and CII values for five types of engine

	6S60MC-C	Wärtsilä 16V32	WUXI	NVD	WeiChai
$\overline{COV_P}$ %	0.943	0.980	0.852	1.906	0.451
CII, %	1.822	3.977	3.759	4.778	1.168
$COV_{P.max}$ %	3.524	16.142	16.594	11.976	3.025

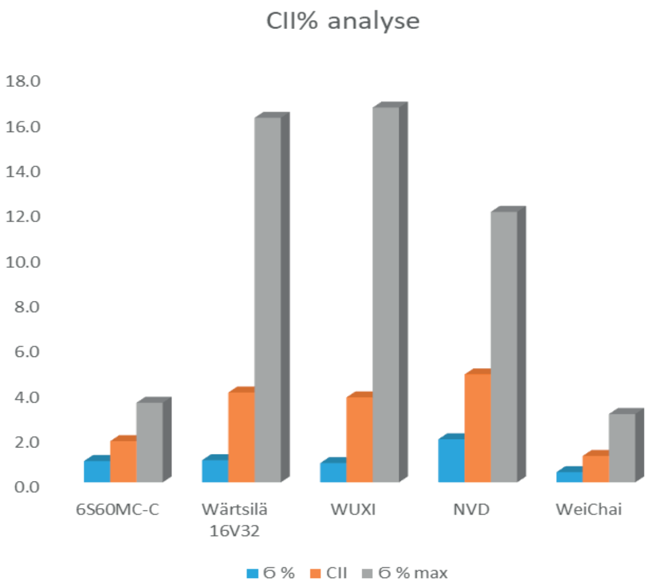


Fig. 8. Comparison of the CII % metric for five types of engine

If the metrics expressed in percentages are to be used to assess the cycle-to-cycle variability of pressure in the cylinder, then to evaluate the cycle-to-cycle variability of fuel injection and intake valve operation, an absolute assessment of the unevenness expressed in crank angle degrees (°CA) is necessary. In this case, a comparison of the dispersion of fuel injection phases in different cylinders, for example $\Delta\varphi_{inj}$ = 3.5°CA, indicating significant irregularity, and $\Delta\varphi_{inj}$ = 0.3°CA, indicating low irregularity, is straightforward and logical.

To assess the cycle-to-cycle variation in the fuel injection timing angle in the cylinder, we can use the dispersion in the deviations of the leading edges of the first pulses in the vibrograms of injection.

$$\Delta\varphi_{adv}, {}^{\circ}CA \cong \sigma_{\varphi adv} = \sqrt{\frac{1}{m}\sum_{j=1}^{m}\left(\varphi_{adv\ j} - \overline{\varphi_{adv\ j}}\right)^2} \quad (20)$$

To assess the cycle-to-cycle variability in the duration of fuel injection in the cylinder, we consider the variance in the deviations in the difference between the leading edges of the impulse vibrations from the injection vibrograms.

$$\Delta\varphi_{inj}, {}^{\circ}CA \cong \sigma_{\varphi in} = \sqrt{\frac{1}{m}\sum_{j=1}^{m}\left(\varphi_{in\ j} - \overline{\varphi_{in\ j}}\right)^2} \quad (21)$$

To assess the cycle-to-cycle variability in the valve closing angle [4,13,16], we consider the dispersion in the deviations in the phases of maximum vibration impulses of the valves from the of injection vibration diagrams.

$$\Delta\varphi_{valve}, {}^{\circ}CA \cong \sigma_{\varphi\ valve} = \sqrt{\frac{1}{m}\sum_{j=1}^{m}\left(\varphi_{valve\ j} - \overline{\varphi_{valve\ j}}\right)^2} \quad (22)$$

Green/yellow/red boundaries showing the normal and increased values of cycle-to-cycle variability can be established for the different engine models. For example, to assess the irregularity of the fuel injection timing advance angle for the S60MC-C engine in operation, the limits can be conditionally set as shown in Fig. 9.

$\Delta\varphi_{adv} \cong \sigma_{\varphi adv}$	Normal < 0.5 °CA	Warning 0.6 – 2.5 °CA	Alarm > 2.6 °CA

Fig. 9. Values of cycle-to-cycle variability

Visual monitoring of the cycle-to-cycle variability metrics offers convenience, and can be carried out in real time through an analysis of the temporal diagrams prior to solving the data synchronisation task [4,13,16].

CONCLUSION

Real-time diagnostics of marine engines offers significant advantages, as it allows us to analyse the current condition of the cylinder-piston group, the high-pressure fuel equipment, and the valve train mechanism directly, during operation. A real-time diagnostic system enables better adjustment of the corresponding mechanisms and facilitates monitoring of the results of this adjustment during operation.

Enabling the real-time operation of portable diagnostic systems represents a significant development in terms of expanding their capabilities. This advance required a number of problems to be solved, including the development of an efficient algorithm for piston TDC estimation, based exclusively on the cylinder pressure diagram, and the development of special methods for assessing the cycle irregularity. Two sensors are proposed here as a standard configuration of the portable system: a gas pressure sensor, to enable the creation of the pressure diagram for the engine cylinder, and a vibration sensor, which provides data on the fuel injection and gas distribution timing. Real-time assessment of cycle-to-cycle irregularity is suggested, based on the calculation of the following coefficients and indexes: the maximum and average values of coefficients of variation for in-cylinder gas pressure $COV_{P.max}, \overline{COV_P}$, the standard deviation for fuel injection and gas distribution timing σ_{φ}, and the cycle irregularity index *CII*. These factors are calculated from sensor data before real-time cycle synchronisation.

The proposed method for determining the TDC of the piston is based on an analysis of the compression curve segment in the cylinder. Compared to other methods, it possesses greater accuracy, as it does not rely on derivatives of the pressure curve, which may contain a high level of noise when drawn from experimental data. The proposed method for determining TDC can be applied without information on the actual compression ratio in the cylinder, a typical scenario for modern engines with variable valve timing.

An algorithm is proposed for preliminary synchronisation in which the initial position of TDC is determined with sufficient accuracy for most practical cases; this algorithm is used as the first approximation for the method developed here.

To enable a comprehensive assessment of engine irregularity in real time, we propose a factor called CII, which is expressed as a percentage. Coefficients for assessing the irregularity in the timing of the fuel injection and the gas distribution valves are also proposed, using dispersion analysis of the vibrogram fronts. These coefficients are expressed in degrees of rotation of the crankshaft.

ABBREVIATIONS

IMO	- International Maritime Organization
HFO	- Heavy fuel oil
ICE	- Internal combustion engine
ADC	- Analogue-to-digital converter
TDC	- Top dead centre
BDC	- Bottom dead centre
MIP, IMEP	- Mean indicated pressure
IPOWER	- Indicated engine power
COV	- Coefficient of variation
CII	- Cycles irregularity index

REFERENCES

1. IMO. International convention for the safety of life at sea, SOLAS consolidated edition. London, International Maritime Organization, 2020.

2. Varbanets R, Minchev D, Savelieva I, Rodionov A, Mazur T, Psariuk S, Bondarenko V. Advanced marine diesel engines diagnostics for IMO decarbonization compliance. AIP Conf. Proc. 2024, 3104(1), 020004. https://doi.org/10.1063/5.0198828

3. Heywood J B. Internal combustion engine fundamentals, 2nd ed. New York, McGraw-Hill Education; 2018.

4. Varbanets R. Diagnostic control of the working process of marine diesel engines in operation. Dissertation for Doctor of Technical Sciences, Odessa National Maritime University, 2010.

5. TDC Sensor System. 2024. Retrieved from https://www.kistler.com/INT/en/cp/top-dead-center-sensor-systems-2629d/P0001160.

6. Polanowski S. Determination of location of top dead centre and compression ratio value on the basis of ship engine indicator diagram. Polish Maritime Research 2008, 2(56). https://doi.org/10.2478/v10012-007-0065-2

7. Tunestal P. Model based TDC offset estimation from motored cylinder pressure data. Proceedings of the 2009 IFAC Workshop on Engine and Powertrain Control, Simulation and Modeling IFP, RueilMalmaison, France, Nov. 30–Dec. 2, 2009. https://doi.org/10.3182/20091130-3-FR-4008.00032

8. Pipitone E, Beccari A. Determination of TDC in internal combustion engines by a newly developed thermodynamic approach. Applied Thermal Engineering, 2009.

9. Staś M. An universally applicable thermodynamic method for TDC determination. SAE Technical Paper 2000-01-0561. 2000. Retrieved from http://papers.sae.org/2000-01-0561/. doi, 10.4271/2000-01-0561

10. Tazerout M, Le Corre O, Rousseau S. TDC determination in IC engines based on the thermodynamic analysis of the temperature-entropy diagram. SAE Technical Paper 1999-01-1489. 1999. Retrieved from http://papers.sae.org/1999-01-1489/. doi, 10.4271/1999-01-1489.

11. Varbanets R A, Zalozh V I, Shakhov A V, Savelieva I V, Piterska V M. Determination of top dead centre location based on the marine diesel engine indicator diagram analysis. Diagnostyka 2020, 21(1), 51–60. https://doi.org/10.29354/diag/116585

12. Neumann S, Varbanets R, Kyrylash O, Yeryganov O V, Maulevych V O. Marine diesels working cycle monitoring on the base of IMES GmbH pressure sensors data. Diagnostyka 2019, 20(2), 19–26. https://doi.org/10.29354/diag/104516

13. Varbanets R, Karianskyi S, Rudenko S, Gritsuk I V, Yeryganov A, Kyrylash O, Aleksandrovskaya N. Improvement of diagnosing methods of the diesel engine functioning under operating conditions (No. 2017-01-2218). SAE Technical Paper, 2017.

14. Doctor Analysis Software V6.4. 2024. Retrieved from https://iconresearch.co.uk/wp-content/uploads/2017/10/doctor-v6-4-reference-guide-rev-1-4.pdf.

15. Minchev D, Varbanets R, Shumylo O, Zalozh V, Aleksandrovska N, Bratchenko P Truong T H. Digital twin test-bench performance for marine diesel engine applications. Polish Maritime Research 2023, 30(4), 81–91. https://doi.org/10.2478/pomr-2023-0061

16. Neumann S, Varbanets R, Minchev D, Malchevsky V, Zalozh V. Vibrodiagnostics of marine diesel engines in IMES GmbH systems. Ships and Offshore Structures 2023, 18(11), 1535-1546. https://doi.org/10.1080/17445302.2022.2128558

17. Neumann S. High temperature pressure sensor based on thin film strain gauges on stainless steel for continuous cylinder pressure control. CIMAC Congress Digest, Hamburg. 2001, pp. 1–12.

18. Lehmann & Michels GmbH. Premet type L, LS, and XL electronic indicators. 2006. Retrieved from http://www.lemag.de/fileadmin/user_upload/PREMET_liste_100_04_2006.pdf

19. Maridis GmbH. MarPrime technical data. Maridis GmbH. Rostock, Germany; 2015.

20. Varbanets R, Fomin O, Píštěk V, Klymenko V, Minchev D, Khrulev A, Zalozh V, Kučera P. Acoustic method for estimation of marine low-speed engine turbocharger parameters. Journal of Marine Science and Engineering 2021, 9(3), 321. Retrieved from http://dx.doi.org/10.3390/jmse9030321

21. Minchev D, Varbanets R, Aleksandrovskaya N, Pisintsaly L. Marine diesel engines operating cycle simulation for diagnostics issues. Acta Polytechnica 2021, 3(61), 428–440. http://dx.doi.org/10.14311/AP.2021.61.0435

22. Shi J, Wang T, Zhao Z, Wu Z, Zhang Z. Cycle-to-cycle variation of a diesel engine fueled with Fischer–Tropsch fuel synthesized from coal. Appl. Sci. 2019, 9, 2032. https://doi.org/10.3390/app9102032

23. Raspberry Pi Pico W and Pico WH. 2024. Retrieved from https://www.raspberrypi.com/documentation/microcontrollers/raspberry-pi-pico.html.

24. Blitz-PRO by D. S. Minchev. User's manual. Retrieved from, http://blitzpro.zeddmalam.com/ extra/Tutorial/Help.pdf.

25. Minchev D S, Gogorenko O A, Varbanets R A, Moshentsev Y L, Píštěk V, Kučera P, et al. Prediction of centrifugal compressor instabilities for internal combustion engines operating cycle simulation. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2022. https://doi.org/10.1177/09544070221075419

26. Neumann S. High temperature pressure sensor based on thin film strain gauges on stainless steel for continuous cylinder pressure control. CIMAC Congress Digest. Hamburg. 2001. pp. 1–12.

27. Himmelblau D M. Applied nonlinear programming. 1972.

28. Powell M J D. An efficient method for finding the minimum of a function of several variables without calculating derivatives. Computer J. 1964, 7, 155.

29. Melnyk O, Onyshchenko S, Onishchenko O, Lohinov O, Ocheretna V. Integral approach to vulnerability assessment of ship's critical equipment and systems. Transactions on Maritime Science 2023, 12(1). doi, 10.7225/toms.v12.n01.002

30. Melnyk O, Onyshchenko S, Onishchenko O, Shumylo O, Voloshyn A, Koskina Y, Volianska Y. Review of ship information security risks and safety of maritime transportation issues. TransNav 2022, 16(4), 717-722. doi, 10.12716/1001.16.04.13

31. Orobey V, Nemchuk O, Lymarenko O, Piterska V, Lohinova L. Taking account of the shift and inertia of rotation in problems of diagnostics of the spectra of critical forces mechanical systems. Diagnostyka 2021, 22(1), 39–44. https://doi.org/10.29354/diag/132555

32. IMO. International convention for the safety of life at sea, part B. Prevention of fire and explosion, paragraph 2.2.5.2, SOLAS consolidated edition. London, International Maritime Organization, 2020.

33. Shi J, Wang T, Zhao Z, Wu Z, Zhang Z. Cycle-to-cycle variation of a diesel engine fueled with Fischer–Tropsch fuel synthesized from coal. Appl. Sci. 2019; 9: 2032. https://doi.org/10.3390/app9102032

34. Schmillen K, Wolschendorf J. Cycle-to-cycle variations of combustion noise in diesel engines. SAE Transactions 1989, 98, 60-70. http://www.jstor.org/stable/44580924