ASSESSMENT OF LARGE BORE MARINE ENGINE CYLINDER PRESSURES TO ESTIMATE NO_X EMISSION

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Abstract

High demand to improve exhaust gas emission characteristics while keeping fuel economy requires a new technology for engine control. One of the methods used to meet these requirements is a convenient and real-time cylinder pressure monitoring. The final aim is balanced combustion and reliable, emission-compliant operation. Excluding the direct exhaust emission measurement, easily accessible indicators come from the in-cylinder pressure combustion data. The aim of this paper is to establish combustion model variables for use in transient condition analysis. The model should be able to predict cylinder-averaged quantities such as NO_x emission. To overcome the exhaust analyzers problem, a set of engine operating conditions was tested at steady state to provide a wide range of combustion conditions with their associated emissions. Multi-zone models are the simplest and most suitable to observe the effects of empirical variations in the engine operating parameters. The obtained results are unlikely to be as detailed as those of a good CFD model, but the ease of changing characteristics makes such models useful for cheap and easy means of analyzing. Successful prediction of the NO_x emission in engines varies in operating conditions. The NO_x emission levels for different operating conditions are calculated for large bore, slow speed marine engine. The agreement between calculated and measured results is discussed. Cylinder pressure based model has the broad ability to predict NO_x emissions in steady state.

Keywords: *marine diesel engine, internal combustion engines, engine maintenance, air pollution*

1. Introduction

The increasing demands for environmental protection in shipping as well as other fields are being recognized by International Maritime Organization, through further development of the rules valid with respect to sulphur and nitrogen oxides. At present, an air pollution Annex to Marpol 73/78 convention is being compiled to regulate emission limits for the above mentioned gas components. Much effort has been spent developing combustion methods that will adequately predict engine performance and pollutant formation. Engine combustion pressure data, can be used to assess the condition of slow speed engines. Each cylinder combustion pressure component exhibits typical distinctive characteristics regarding timing (in crank angle), magnitude, and shape. Cylinder pressure data helps undertake the fuel injection timing adjustments - to improve power balance. These gives the technical bases for condition-based maintenance program on which engine maintenance is scheduled and exhaust emission levels, particularly NO_x have to be included. Thus, balancing the engine ensures that power of each cylinder is nearly equal and NO_x emission is similar. Marine engines are designed with adjustable components to allow the engine to be adjusted for maximum efficiency or NO_x emission, when used in particular application. Empirical methods for NO_x estimation utilizing constants derived from experimental data do not require detailed kinetics of emission formation knowledge. These models can be successfully applied to those systems and conditions from which they have been developed [1].

The commonly used simple two-part apparent heat release rate (AHHR) correlation is replaced by a detailed multizone combustion model. The attempts to simulate the overall thermal effect of the combustion, while the latter is able to predict the time dependent mixture formation and rate of heat release in each zone. This allows the distribution of burning rate, temperature and equivalence ratio to be determined both spatially and temporally. The adoption of a multizone combustion model, in this context, is to make the prediction of nitric oxide emissions possible [2]. Phenomenological multi-zone models take into account the spatial differences in temperature and chemical compositions by dividing a cylinder into two or more zones. Each zone is treated as a well-mixed open thermodynamic system. Their predictive ability and accuracy greatly depend on what methods and sub-models are used. Many aspects of multi-zone models, for example the accuracy of simultaneous prediction of exhaust emissions and cylinder pressure, still need to be improved. The aim of this paper is to adopt a phenomenological multi-zone combustion model for use in steady condition analysis. However, it is impossible for the phenomenological models to give the detailed in-cylinder data flow and temperature fields. Thus, substantial additional data about the engine combustion process can be obtained when the heat-release approach is coupled with a temporal analysis of geometric analogues the surface within the combustion chamber. Nevertheless, compared with the CFD model [3], [4], the phenomenological multi-zone model provides a simple way to obtain full cycle simulation.

Paper investigates the results of an experiment targeting prediction of NO_x emission. The experiment included a numerical simulation, where the calculated emission was compared to one investigated experimentally in marine large bore, slow speed engines. The predictive capabilities of the model have been tested by means of experimental data over a full spectrum of engine steady state conditions. The exhaust emissions were measured in accordance to IMO and ISO standards using well-proven experimental apparatus and techniques.

2. Model aspects description

Defined heat release in functional form was chosen to match experimentally observed heat-release profile. Basic phenomenological description of diesel combustion comprises three primary phases: the ignition delay, the rapid - premixed burning followed by a slower mixing-controlled fuel burning. The fraction of the injected fuel that burns in each of the phases is empirically linked to the duration of the ignition delay. One algebraic function is used to describe the premixed heat-release phase and second to describe the mixing-controlled phase [5]. These two functions are weighted with the phase proportionality factor β, which is largely a function of the ignition delay, thus:

$$\frac{m_{f,b}\left(t'\right)}{m_{f,o}} = \beta f_1 + \left(1-\beta\right) f_2 \tag{1}$$

where:

$m_{f,b}$ – mass of fuel burned,

$m_{f,o}$ - total fuel mass injected per cycle per cylinder,

t' – time from ignition (non-dimensionalized by total time allowed for combustion).

$$t' = \frac{t - t_{ign}}{\Delta t_{comb}}$$

The premixed-burning is:

$$f_1 = 1 - \left(1 - t'^{K_1}\right)^{K_2} \tag{2}$$

and mixing-controlled function is:

$$f_2 = 1 - exp\left(-K_3 t'^{K_4}\right) \tag{3}$$

where:

K_1, K_2, K_3, and K_4 are empirical coefficients.

The proportionality factor β is given by:

$$\beta = 1 - \frac{a\phi^2}{\tau_{id}^c} \tag{4}$$

where:

ϕ is the overall fuel/air equivalence ratio and a, b, and c are empirical constants.

Generally, multi-zone heat release models should be checked against experimentally derived heat-releases profiles and recalibrated if necessary, before being used for predictions. The concept of the model assumes that the liquid fuel injected into a combustion chamber as several jets is divided into many small zones. In large marine diesel engines, the air flow is essentially quiescent. All combustion events in each zone: droplet break-up, evaporation, air–fuel mixing, ignition, premixed heat release, mixing-controlled heat release, heat transfer and formation of exhaust emissions, are calculated in order to achieve zonal temperature and compositions. The zonal property is represented by the average state of temperature, air-fuel ratio and NO_x concentration. The cylinder pressure in the zonal calculation is solved iteratively to satisfy the volume constraint. The model contains: spray development, air entrainment and mixing, droplet evaporation, heat transfer, combustion and NO_x formation sub-models. Engine combustion model is based on the undisturbed turbulent gas jets, also referenced as the "Cummins model", where diesel spray is treated as quasi-steady gas jet penetrating into gaseous environment of combustion air [6]. The result is a continuous profile of fuel vapor concentration ranging from very rich at the core to very lean mixture at the periphery of the spray. Fresh air is continuously entrained into the spray and represents the main contribution for the estimated rates of combustion and pollutant formation. The tip penetration is calculated on the empirical correlation:

$$S = \frac{450 d_{noz}^{0.5} \left(\rho_f / \rho_{ref}\right)^{0.4}}{\left(1 + \rho_{cyl} / \rho_{atm}\right)^{0.85}} \left(\frac{\rho_{cyl}}{\rho_{atm}}\right)^{0.5} \Delta p_{inj}^{0.25} t^{0.6} \tag{5}$$

where:

S – tip penetration,

ρ_{ref}, ρ_f, ρ_{cyl}, ρ_{atm} – densities of: reference and actual fuel, cylinder gases, atmospheric air,

Δp_{inj} – pressure drop across the nozzle hole,

t – time increment since injection start.

The distribution of the fuel concentration across the equivalent circular spray slice is given by:

$$c = c_m \exp\left[-0.693\left(\frac{y}{R_{1/2}}\right)^{5/2}\right] \tag{6}$$

where:

c – fuel mass fraction,

c_m – fuel mass fraction (coordinate in radial direction of the spray cross-section),

$R_{1/2}$ – radial location in the spray (where a fuel concentration reaches a value $c_m/2$).

In order to estimate rates of combustion and pollutant formation, a set of evolving discrete combustion zones is superimposed on the continuous calculated fuel-air distribution. The contribution of each zone to the total wall heat transfer is based on the product of zone mass and temperature.

$$\frac{dQ_{w,z}}{d\varphi} = \frac{m_z t_z}{m_A T_A + m_C T_C + \sum_{i=1}^{j} m_{Bi} T_{Bi}} \cdot \frac{dQ_{w,tot}}{d\varphi} \quad (7)$$

where:

zonal index z applies to all zones within the cylinders.

The change of composition and internal energy of each zone caused by the mixing between the zones can be determined similarly to energy and mass balance equations for various zones. Thus, NO_x concentration caused by the mixing can be calculated from the following equations [7]

$$\frac{d[NO]_{Ci}}{dt} = \frac{d[NO]_{Ci,A}}{dt} + \sum_{i=1}^{j} \frac{d[NO]_{Ci,B}}{dt} \quad (14)$$

$$\frac{d[NO]_{Ci,A}}{dt} = \frac{dm_{Ci,A}}{dt} \frac{[NO]_A}{m_A} \quad (15)$$

$$\frac{d[NO]_{Ci,B}}{dt} = \frac{dm_{Ci,B}}{dt} \frac{[NO]_{Bj}}{m_{Bj}} \quad (16)$$

In order to enable prediction an emissions formation in terms of NO_x, the model has been equipped with the widely accepted extended Zeldovich model [6].

3. Experimental details

The experimental efforts described below were an attempt to quantify emissions associated with measured engine cylinder pressure. Using test bed engine, having the specification listed in Table 1, exhaust emission examinations were made.

Tab. 1. Test engine details

Engine type	Wärtsila **7S50MC**, reverse.
Number of cylinders	7
Bore [mm]	500
Stroke [mm	1910
Rated engine speed [rpm]	127
Output [kW]	10010

The combustion pressure was recorded by the fast data acquisition system. Subsequently, the trials were performed to assess the engine operation influences. Emission measurements were carried out on engine at steady-state operation. All engine performances were continuously recorded, together with exhaust gas components concentration. The performance measurement procedure of marine engines on test beds is performed in accordance to Annex VI of Marpol 73/78 convention - with the specification given in the IMO NO_x Technical Code and ISO standards.

To reduce emissions variability due to fuel variables, all tests are performed with the selected marine distillate fuel. Today there are no sensors that can be mounted directly into the combustion chamber in conventional, commercial diesel engines. Therefore, for this project a portable marine diesel engine electronic indicator (Premet-Lemag*) was chosen.

4. Results and discussion

In order to determine the NO_x emission rate using multizone phenomological model and some engine operating variables, experimental data have been engaged. The primary stage of the experiment involved measured cylinder pressures history analysis and the estimation area concerned: heat release, in-cylinder temperature (zonal and average) and finally NO formation. Heat release, temperature and NO formation calculation was conducted for each cylinder unit and test cycle load level. To evaluate the NO formation distribution over all engine cylinders and engine operating range, estimated NO concentrations (in ppm) cylinder were compared to IMEP and maximum combustion pressure, at each load and speed level, accordingly. Evaluation of data of the total measured NO_x engine emission and cylinder cycle computed concentration obtained from cylinder pressure is now possible. All estimated concentrations have been normalized by scavenge ratio which were dependent on engine load. The assumption was made that air distribution over all engine cylinders was equal under related load and speed condition. A comparison was made between an appropriate set of estimated NO_x cylinder rate profiles and measured average (after turbocharger) NO_x concentration.

The investigated engine test conditions have been carefully adjusted to match calculation procedure data. An example of cylinder pressure data – input matrix for specific condition calculation of the engine load and further data processing is shown in figure 1. It can be noticed from the heat release profiles that there is only one peak, which is in accordance with combustion proces for turbocharged diesel engines.

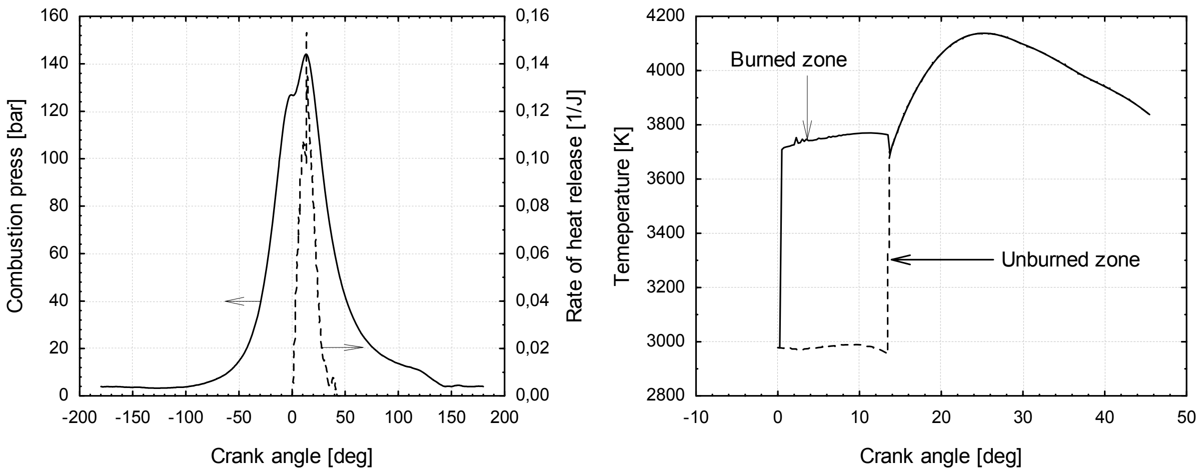

Fig. 1. Measured cylinder pressure, rate of heat release history and zonal temperature for tested engine

Cylinder pressure measurement has to be validated for further heat release and NO analysis. A zero-dimensional model has been tested for estimating cylinder pressure from the rate of heat release. The estimation is essentially based on a minimum approximation and arbitrary preferred equations. An example result of the computing process presented in figure 2 is synthesized to estimate cylinder unit performance.

* Lehmann & Michels GmbH & Co. KG, Germany

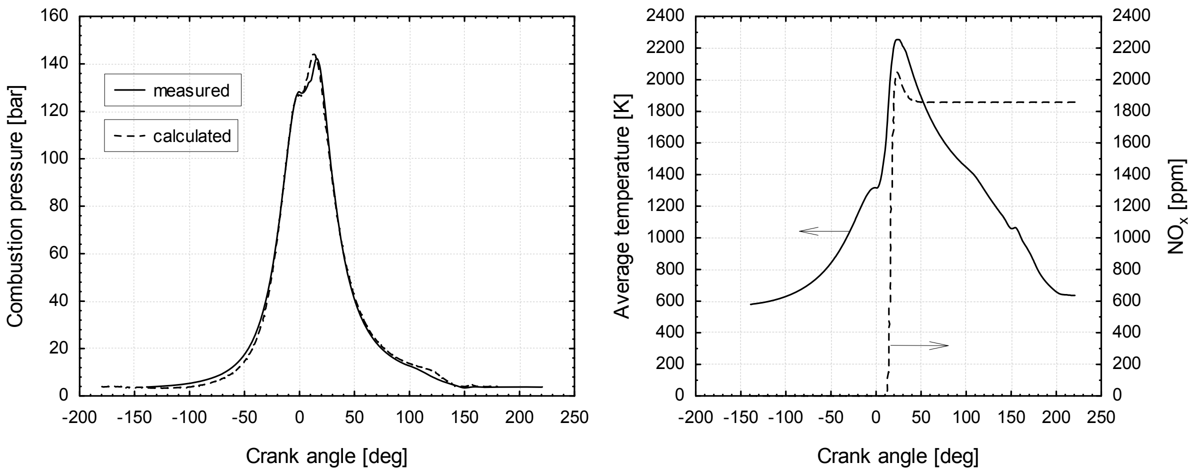

Fig. 2. Comaprison of measured and calculated cylinder pressure and average temperature with NO_x profile

It can be seen that NO_x emission profile starts to rise after the start of combustion and continues so to a maximum value. Later, the level remains steady or decreases slightly. There are a few degrees of crankshaft angle delays for NO_x process formation progress, and several degrees difference of the maximum pressure and maximum NO_x concentration position. Unlike high speed diesel engines experimental results, the cylinder maximum combustion pressure of slow speed engines does not coincide with highest NO_x formation rate. Thus, IMEP value of individual cylinder combustion event is a better choice for indicating NO_x formation rate. Figure 3 illustrates the set of cylinder pressures taken within the test cycle engine load range. Subsequent analysis of the presented in-cylinder pressure history produced the rate of heat release profile.

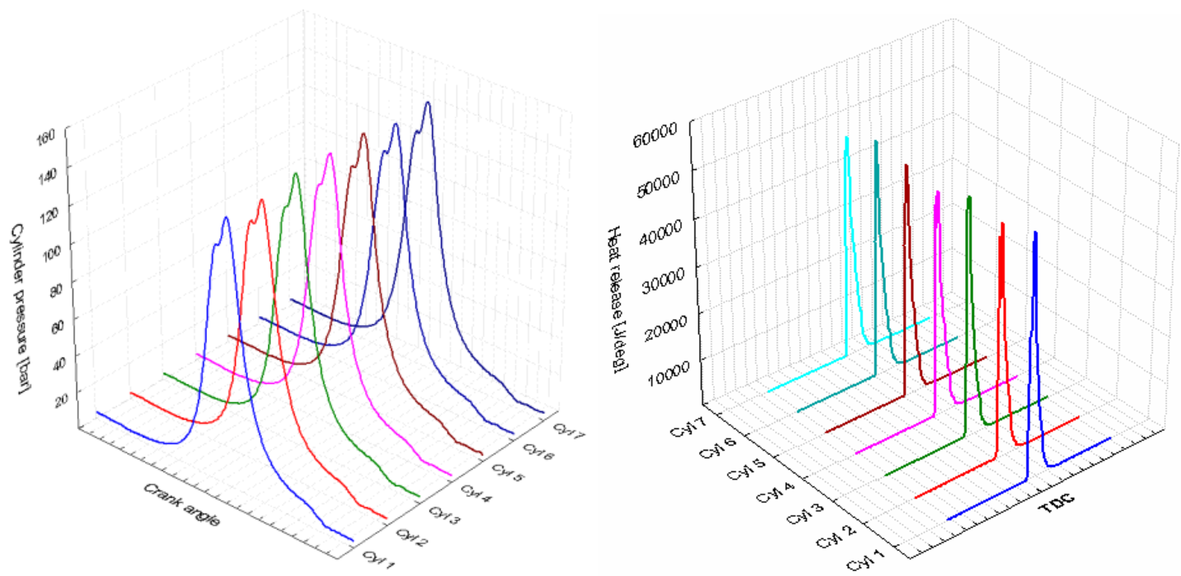

Fig. 3. Comparison of measured cylinder pressure and rate of heat release profile

The cylinders rate of heat release profile comparison illustrates the slight premixed spike during the first phase of burning and higher gradients during the diffusion part of burning. The rate of air-fuel mixing in a quiescent combustion chamber depends mainly on the fuel injection process. Therefore, the influence of individual engine or fuel parameters can not be investigated without considering secondary influences. In addition, the testing at low temperature in engines (low load) proved to be difficult.

The investigated operating conditions expressed as the histories of NO_x concentration are displayed in figure 4. It has been shown that there are several CA delays of the start NO_x formation process, with the combustion commencement. Later, while combustion proceeds NO_x concentration increases, up to maximum level at 10-20 CA with respect to highest cylinder pressure.

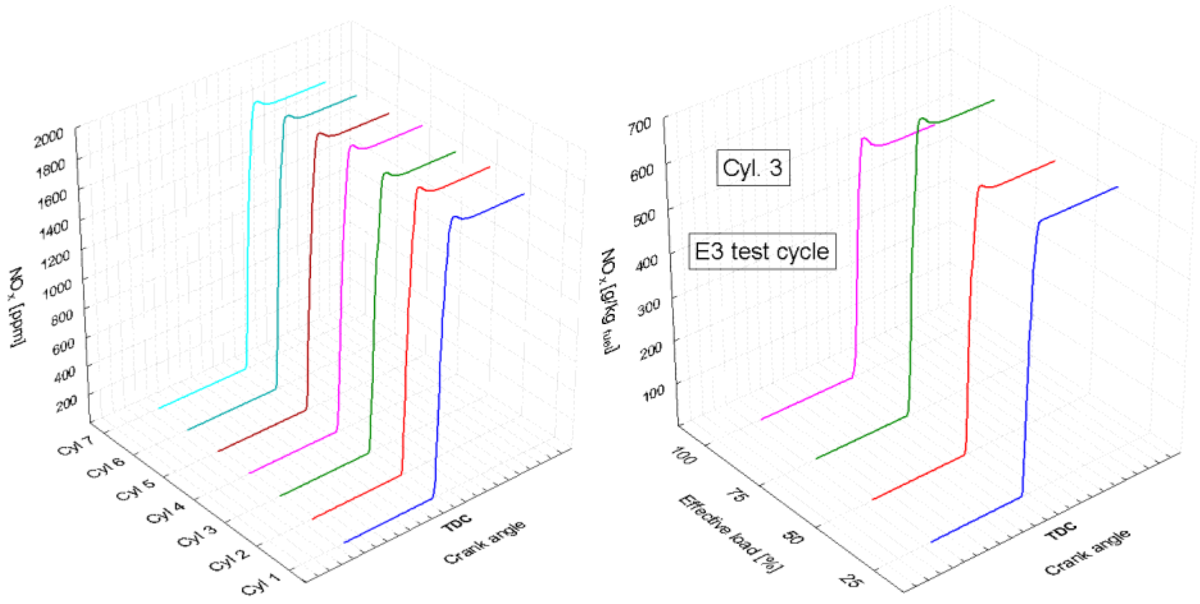

Fig. 4. Predicted average NO_x emission under nominal engine load for whole engine and one selected cylinder.

5. Conclusions

An experimental investigation of NO_x emission has been tested and confronted with phenomenological multi-zone combustion model calculation. Relevant and important values of comparison are presented in figure 5. The predictive ability of the used model has been partially proved by experimental results. There is a visible error concerning NO_x emission at low load, even when the maximum combustion pressure prediction and measurement show reasonable agreement. Thus, model calculation influenced by ignition delay, specifically at low engine load, needs to be improved as a further step. There is also a practical problem to overcome – inaccuracy of combustion pressure measurement by means of standard indicating valve-cock. The sub-models of spray development and air entrainment have to be carefully calibrated to increase model predictive ability. Nonetheless, the results are promising and show that the phenomenological multi-zone combustion model is capable of accurate predicting the NO_x emission. The development of marine engines diagnostics stimulates the implementation of a real-time cylinder pressure measurement. Furthermore, a simple combustion analysis method, including NO_x prediction is required. In the case of cylinder pressure measurement, where crank pulses are limited, pressure data have to be carefully treated in terms of P-V diagram, heat release and its cumulative value, due to individual (cylinder unit) crank position fluctuations. Thus, to omit this inconvenience, there is a possibility to introduce a system, where sampling pulse from flywheel or rotary encoder can be replaced by straight piston position measurement or another approach [7]. There is an error in the estimation of NO_x level at the higher load condition, slightly below measured trace.

In practice, however, averaging enigne exhaust emission concentration distribution at steady load condition produce a smoother profile, which do not reflect precisely results of calculation based on cylinder pressure – at selected crank angles of one cycle.

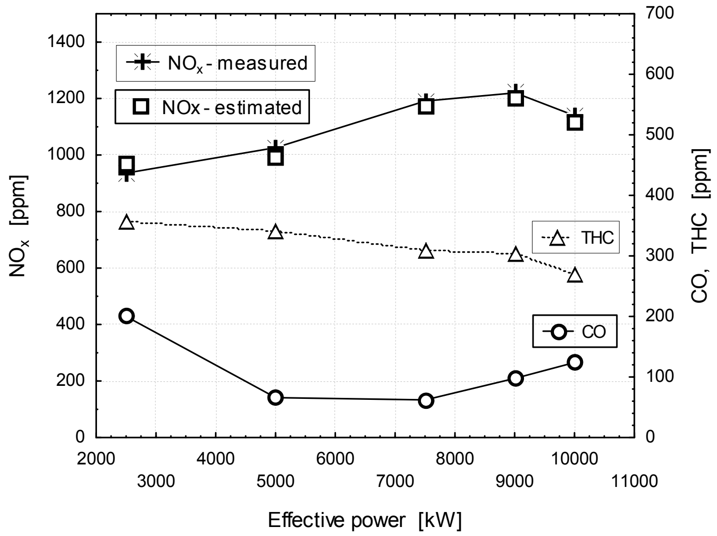

Fig. 5. Comparison of measured and predicted NO_x emission for 7S50MC engine

An additional possibility of discrepancy have to be taken into account as slow speed engines are characterized by cycle variation, particularly when used for ship propulsion. The present diagnostic systems need to be improved, especially in terms of crankshaft torsional vibration, because there is a strong dependence on measured cylinder pressures.

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