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CONCEPT OF VIBROACOUSTIC DIAGNOSTICS OF THE FUEL INJECTION AND ELECTRONIC CYLINDER LUBRICATION SYSTEMS OF MARINE DIESEL ENGINES

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ABSTRACT

Although direct measurements of the fuel injection pressure and the travel of the injector needle in conjunction with measurements of the valve train mechanism timing can provide complete diagnostic information about the technical conditions of the fuel injection and valve train systems, this requires the installation of sensors and other equipment directly into the systems, which is possible within research laboratories but is generally forbidden during operation of the ship. Malfunctions in the fuel injection and valve train systems can also be identified from the indicator diagrams of an engine operating cycle, expressed as P(V) and P(deg) diagrams. The basic parameters of the engine operating cycle, such as the maximum combustion pressure Pmax, compression pressure Pcompr, and indicated mean effective pressure IMEP, can also be used to indicate deviations from proper engine operation. Using a combination of a vibration sensor with an in-cylinder gas pressure sensor widens the capabilities of diagnostics for marine diesel engines under operational conditions. A vibration sensor with a magnetic base can help in determining the timings of the lifting and landing of the injector needle, fuel delivery by the fuel injection pump, opening and closing of the circulation of heated heavy fuel oil, and opening and closing of the gas distribution valves. This also offers a promising solution for diagnostics of the cylinder lubrication oil injectors. The proposed approach allows valuable information to be received during engine operation in accordance with the principle of non-destructive control, and can help in early detection of possible engine malfunctions.

Keywords: low-speed two-stroke engines; fuel injection system; electronic cylinder lubrication system

INTRODUCTION

The reliable operation of marine diesel engines can be ensured by periodic control and monitoring. Determining the parameters of the operating cycle for each engine cylinder can help in making the necessary corrections and adjustments to ensure the efficiency of operation of the engine, to distribute the load on the engine cylinders evenly, and to enable early detection of dangerous trends in the evolution of technical conditions. One important task is to control the level of toxic emissions at ship operation, which involves adjusting the engine in order to obtain a minimal value of nitrogen oxides (NOx) emissions in compliance with IMO regulations [1-3].

An analysis of indicator diagrams such as P(V) and P(deg), based on an examination of their shape and consideration of the main parameters (such as maximum combustion pressure Pmax, pressure at the end of compression Pcomp, indicated mean effective pressure IMEP, ignition point, etc.), could enable the detection of a number of possible malfunctions, as shown by Varbanets et al. [4-9]. These malfunctions include incorrect fuel injection timing, ore fuel distribution, wrong gas distribution valves timing, valve leakage, etc.

It should be mentioned that different malfunctions may give rise to the same behaviour in the indicator diagram, as reported by Neumann et al. [10]. For example, late fuel injection and wear to the fuel injection equipment both result in the same changes in the shape of the indicator diagram, and a reduction in the compression pressure Pcomp could be caused by wear to the cylinders, damage to the piston rings, a leaking exhaust valve, or incorrect exhaust valve timing. Another issue is that in their early stages, many malfunctions can result in only minor changes to the indicator diagram, despite the importance of detecting them.

In view of this, identifying the type of malfunction using only indicator diagrams for analysis can be a complicated task. An advanced analysis of the fuel injection pressure diagrams and valve train diagrams could solve the problem, but this would require interruption of engine systems and the installation of special equipment. Under the operating conditions of the ship, taking such additional measurements is generally forbidden, as it could create fire hazards from possible fuel leakages, according to regulations from the IMO and classification societies [1, 11].

One alternative to direct measurements is the application of non-destructive methods. Vibration sensors with a magnetic base can be effective tools, as Varbanets et al. have shown [9, 11-14]. A magnetic vibration sensor can be easily mounted in the desired place, and provides fast measurements, meaning that measurements under pseudo-stationary conditions of engine operation are obtained. It is important that the vibration diagram and indicator diagram are recorded synchronously and can be analysed in parallel, as this allows for more accurate detection of possible malfunctions.

The operation of a high-pressure fuel injection system and valve train mechanism could be controlled based on an analysis of the signal from a vibration sensor, thus allowing for the detection of fuel injection and valve timings.

Another powerful approach for further experimental data processing is the application of mathematical simulation tools, which enable prediction of the engine operation under normal conditions and under some types of malfunction [15, 16].

The aim of this paper is to present an approach for the non-destructive diagnostics of a fuel injection system, a valve train mechanism and cylinder lubrication for marine diesel engines, based on the use of vibroacoustic sensors, analysis of in-cylinder pressure diagrams and a mathematical simulation of the engine operating cycle.

MEASURING TECHNIQUE AND DATA PROCESSING

Fig. 1 shows an example of diagnostic data recorded from a MAN 6S60MC-C marine diesel engine operating at maximum continuous rating (MCR). A special algorithm is applied to accurately estimate the top dead centre [17].

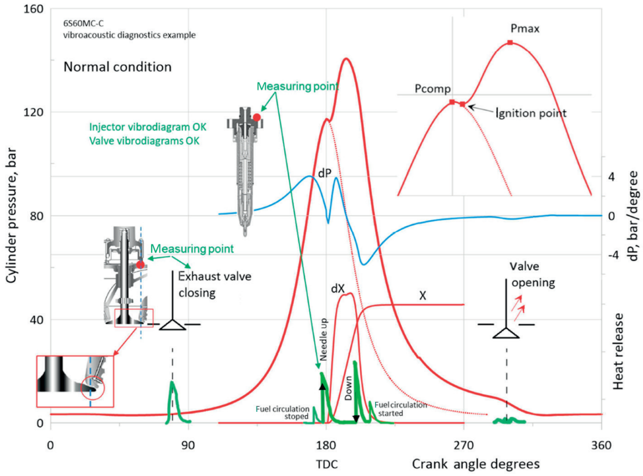

Fig. 1. Example of diagnostics data, including superimposed diagrams of the in-cylinder gas pressure (red line) and the processed vibration sensor signals from the fuel injector and exhaust valve [18]

Table 1 shows the basic parameters of the engine operating cycle, which were obtained from the diagnostics procedure. The in-cylinder pressure sensor was attached to the indicator valve with a Thompson adapter [11]. The vibration sensor was installed sequentially at the following points: the back end of the fuel injector, the exhaust valve housing, and the high-pressure fuel pump outlet. For each installation point, the measurements included synchronous recording of the in-cylinder pressure sensor signals and vibration sensor signals. A combined analysis of the indicator diagram, $P(deg)$, constructed on the basis of recorded in-cylinder pressure data, and a vibration diagram, obtained from the processed data from the vibration sensor, enabled us to determine the fuel ignition delay and the difference between the geometrical and actual fuel injection timing (*DeltaG*), which is an indicator of wear of the injection equipment and the quality of fuel atomisation.

Tab. 1. Operating cycle parameters determined by the diagnostics procedure [5]

Pmax (Pz)	Maximum combustion pressure at the corresponding crank angle, bar/°CA
Pcomp	Compression pressure, bar
IMEP (MIP)	Indicated mean effective pressure, bar
IPOWER	Cylinder indicated power, kW
Pignition	Firing pressure and corresponding crank angle, bar/°CA
Pexp (P36)	Pressure at 36°CA after TDC
Fuel injection timing	Actual and geometrical fuel injection timing, °CA
Valve timing	Timing of valves opening and closing, °CA
Fuel ignition delay	Ignition delay between fuel injection and the start of combustion, °CA/ms
DeltaG	Difference between the actual and geometrical fuel injection timings, °CA

The positioning of the vibration sensor at the back end of the fuel injector and the exhaust valve housing is also shown in Fig. 1. It should be mentioned that in most cases, in the authors' experience, installing the vibration sensor at the back end of the injector enables signals to be recorded from both the fuel injector and the exhaust valve [11].

Following these measurements, the signals from both sensors were processed to generate the superimposed diagram in Fig. 1, where a red line shows the in-cylinder pressure *P*(*deg*) and a green line shows the vibration signal. Processing of the vibration signal included frequency filtration and amplitude demodulation methods.

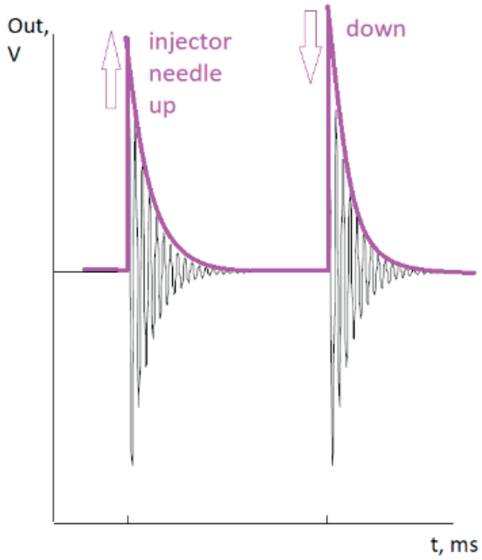

Fig. 2. Amplitude demodulation of high-frequency injector signals [18]

The principle used to process the signal from the vibration sensors is shown in Fig. 2, and has been described by Neumann et al. [18]. It involved amplitude demodulation of the high-frequency shock pulse signals that occur when the nozzle needle rises and sets, when the exhaust and intake valves close, and when the fuel supply starts and is cut off in the high pressure fuel pump (HPFP). For synchronous demodulation, the recovered signal $D(t)$ can be expressed as:

$$D(t)=\frac{1}{2}\cos(-\phi)+\frac{m}{4}[\cos(\omega_m t+\phi)+\cos(-\omega_m t+\phi)],$$

where φ is the phase of the reinserted carrier, m is the modulation index, and ω_m is the modulating frequency.

Some of the details of vibration sensor diagram should be mentioned as follows. As the high pressure of the fuel from the injection pump exceeds the tightness of the fuel injector spring, the injector needle rises to hit the top limiter. The impact of the needle on the limiter generates specific vibrations with fading oscillations. Processing this signal with the amplitude demodulation application gives the moment at which the fuel injector lifts, marked as "Needle up" in Fig. 1. When the fuel is cut off on the injection pump side, the fuel pressure at the injector inlet drops, so the spring causes the needle to fall and hit the seat, generating another vibration impulse marked as "Needle down". It should be mentioned that leading edges of the processed vibration signals represent full opening and full closing of the injector valve. Hence, fuel injection starts slightly earlier than the first pulse, although this difference is quite small (about 0.1–0.3 °CA), and the amount of fuel injected within this period is usually less than 3% of the total injected amount [4, 11]. Thus, in most cases, the instants marked "Needle up" and "Needle down" represent the actual fuel injection timing.

Preheated heavy fuel circulates in the high-pressure fuel injection pipe through the fuel injector. The designers of the MAN applied this solution to avoid the formation of deposits and cocking. Fuel circulation takes place between the fuel injection periods. Immediately before fuel injection starts, when the fuel pressure in the high-pressure pump reaches approximately 10 bar, the fuel circulation valve closes; it opens again after fuel injection ends and the high-pressure pipe fuel pressure drops below 10 bars. The vibration sensor allows for detection of the timing of the opening and closing of the circulation valve, which can provide valuable diagnostic information for each fuel injector (two or three injectors are typically installed per cylinder in a marine two-stroke engine, and the probability of some kind of injection malfunction is relatively high).

In summary, the following instants in the operation of the fuel injector are detected with the vibration sensor, allowing us to calculate the timings:

- "Fuel circulation stops" – closing of the fuel circulation valve;
- "Needleup" – full lifting of the injector needle;
- "Needle down" – closing of the injector needle;
- "Fuel circulation starts" – opening of the circulation valve.

The timings of the opening and closing of the exhaust valve are also detected by the vibration sensor. In most cases, the signal can be recorded from the back end of the fuel injector, meaning that there is no need to change the position of the sensor during measurements. An analysis of the exhaust valve vibration signal helps in detecting the technical state of the valve train system, and particularly the damper valves, and in confirming the correct adjustment of valve timing. Full closing of the exhaust valve occurs at the end of its backward movement, and is influenced by the hydraulic damper, which is installed to soften the impact of the valve when it hits the valve seat. The hydraulic damper delays the closing of the exhaust valve in its very late stages, at about 10–12 °CA [4, 11]. As the result, the typical timing for the closing of the recorded valve is about 90 °CA or even later, while in terms of the gas exchange processes, the valve closes earlier. Examples of recorded timings for the exhaust valve opening and closing are given in Fig. 1.

It is worth mentioning that without the use of the proposed technology, the diagnostics of the valve train system requires complex equipment, which is typically only available in a laboratory setting rather than for an operating engine.

MATHEMATICAL SIMULATION FOR PREDICTION OF MALFUNCTIONS

Mathematical simulation of the engine operating cycle is an effective approach allowing for an advanced analysis of the diagnostics measurements data, as demonstrated by Minchev et al. [19]. Identification and normalisation of experimental data, prediction of the effects of a malfunction, and decision making can be facilitated with this method.

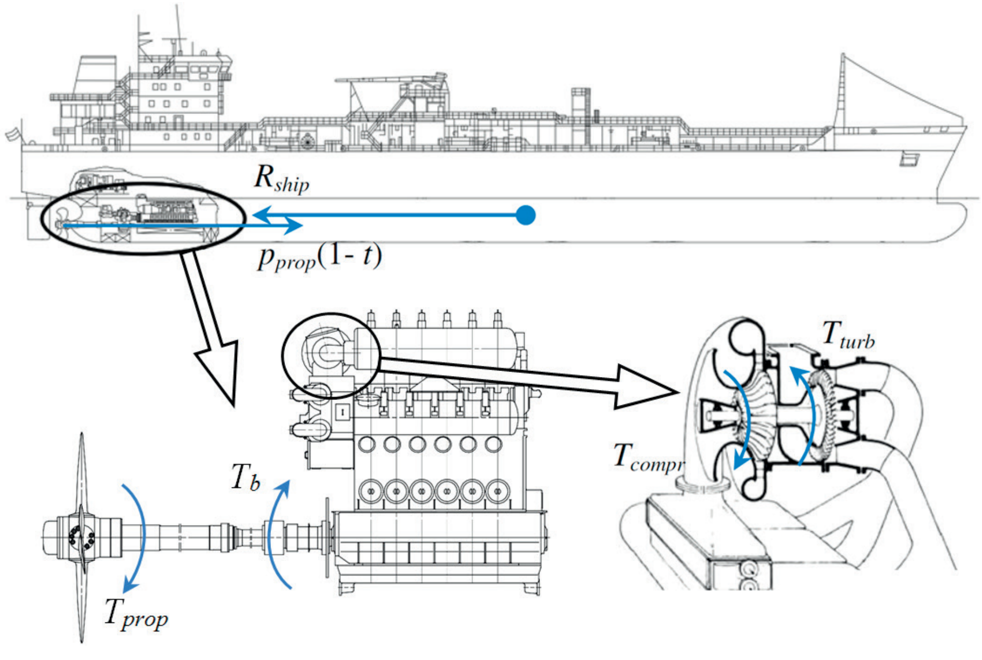

Fig. 3. Decomposition of the ship's propulsion system dynamics system

The Blitz-PRO online simulation service provides the ability to model both the steady and unsteady modes of engine operation [20]. Our mathematical model is based on the quasi-steady and one-dimensional unsteady sets of equations, which describe the processes in the open thermodynamic systems making up the general thermodynamic system represented by the engine.

The quasi-steady equations set include the first law of thermodynamics, mass balance and gas state differential equations. These can be applied to single-zone or double-zone open thermodynamic systems. Paying special attention to simulating the performance of the advanced turbocharger, as implemented in the computational software, helps in accurately predicting the behaviour of the engine under operational conditions, as shown by Minchev et al. [21].

Transient simulation of the engine's operation is based on consecutive (cycle-by-cycle) synthesis of the engine's working processes. The mechanical dynamics of the ship, propulsion system and turbocharging system are described using the following set of equations:

$$p_{prop}(1-t) - R_{ship} = m_{ship}\frac{dv_{ship}}{d\tau};$$

$$T_b\eta_{train} - T_{prop} = J_{rot}\frac{\pi}{30}\frac{dn}{d\tau};$$

$$P_{turb.pulse} - P_{compr} = J_{TC}\left(\frac{\pi}{30}\right)^2 n_{TC}\frac{dn_{TC}}{d\tau},$$

where p_{prop}, T_{prop} are the propeller thrust and torque, respectively; R_{ship} is the resistance of the ship's hull, t is the thrust deduction coefficient, m_{ship} is the displacement of the ship and added water, v_{ship} is the ship's velocity, T_b is the torque of the engine brake, J_{rot} is the moment of inertia of the propulsion system (engine, gear, shaft and propeller) with reference to the crankshaft speed, n is the crankshaft speed, $P_{turb.pulse}$ is the power of the turbocharger turbine at the pulse flow, P_{compr} is the power consumption by the turbocharger compressor, J_{TC} is the moment of inertia of the turbocharger, and n_{TC} is the speed of the turbocharger.

The decomposition of the ship's propulsion system is given in Fig. 3, and the interrelations between its constituents are shown. The characteristics of the propeller and the ship resistance law were set to make the calculations, as shown in Blitz-PRO user's manual [20].

The behaviour and control of the engine are determined by the engine control maps, which include the fuel injection parameters (injection advance, injection duration and amount of injected fuel), and the control parameters of the turbocharger. Engine control maps may be expressed as a function of the engine speed, load, time or cycle number, depending on the malfunction to be studied.

The following example presents an analysis of the possible effects of an irregularity in the fuel injection on the operation of the engine. This irregularity can be detected with the proposed vibroacoustic technology from a set of consecutive injections. Fig. 4 shows an example of a set of fuel injection control maps with irregular fuel injection for the MAN 5G70ME-C diesel engine. This type of malfunction is typical for part-load operation of the engine, but is a cause for more concern if it happens at higher engine loads. A value of 79% for the MCR is considered to determine the effect of periodic missed injections (33% and 85% of the full injection, once in every 12 operating cycles). The results of the calculations are shown in Fig. 5. The engine output power and speed drop from 12,740 kW at 73.6 rpm to average of 10,309 kW at 70.9 rpm, and the scavenging pressure also falls from 281 kPa to an average of 256 kPa. It should be mentioned that the engine speed and output torque create oscillations under irregular fuel conditions, which could cause a high level of vibration of the engine.

A fall in the scavenging pressure causes a corresponding decrease in the air excess ratio for the full injection cycle and a rise in the temperature of the exhaust gases, which could lead to overheating of the exhaust valve and the piston of the cylinder piston with normal fuel injection.

The cyclic variations in the engine operating process are also very clear on the *P*(*V*) diagram indicator in Fig. 6. Mathematical simulation can help to clarify the experimental data and to make predictions of the effects of different malfunctions, including those that occur periodically, as shown in the example.

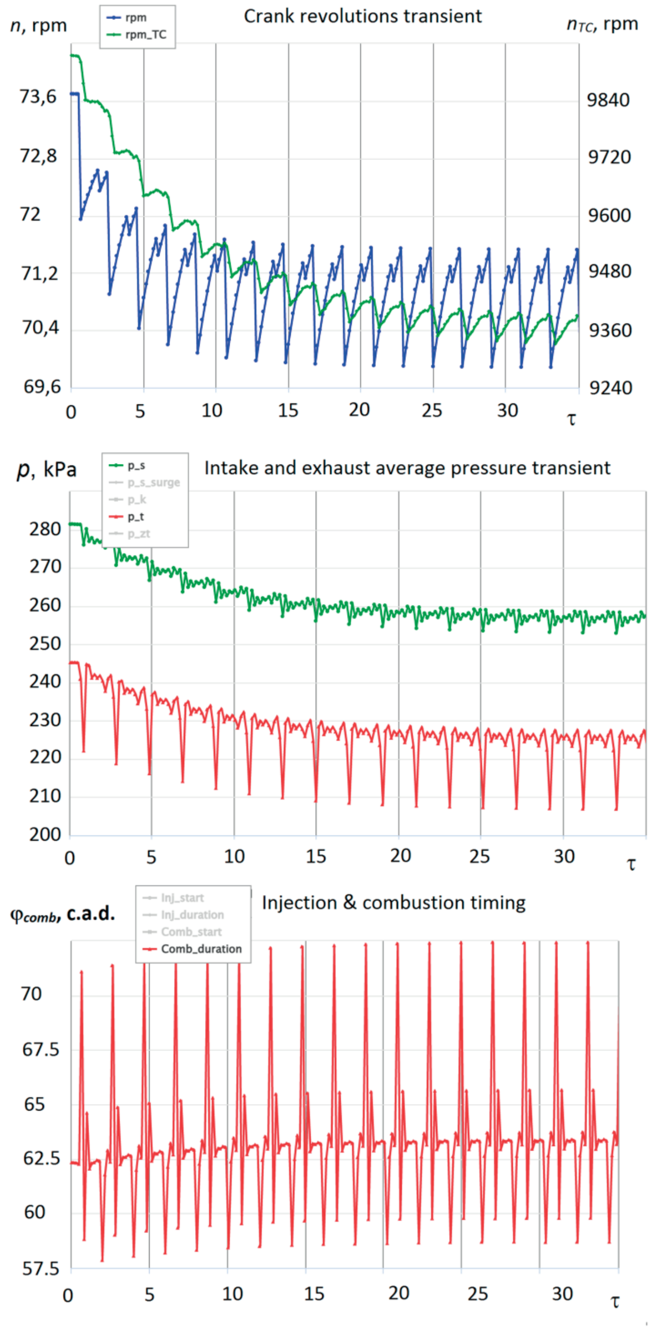

Fig. 5. Results for transient engine operation under a fuel injection irregularity

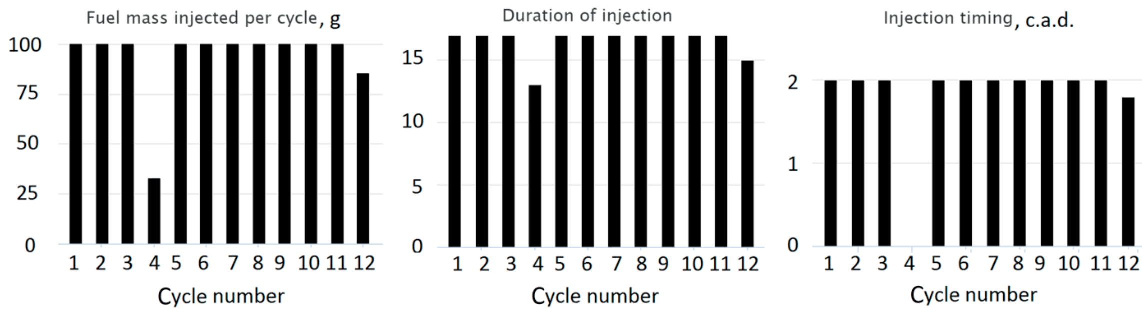

Fig. 4. Fuel injection control maps, which are used to examine the effects of an irregularity in fuel injection

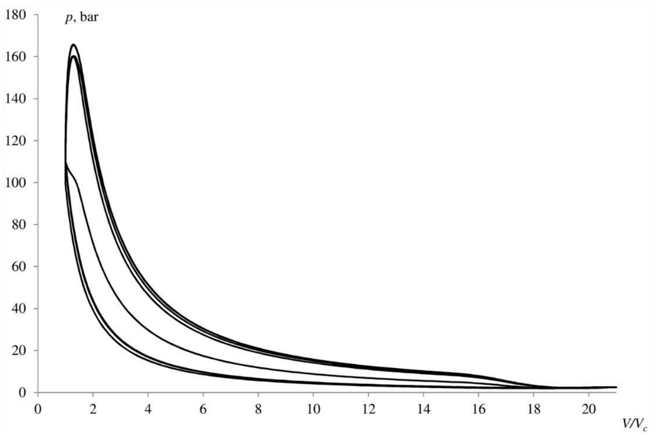

Fig. 6. Cycle variations under irregular fuel injection conditions

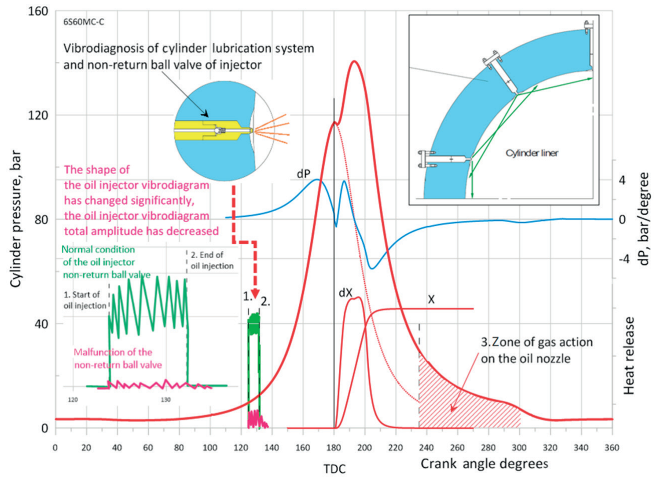

Fig. 7. Vibration diagram for the cylinder lubrication injector, combined with a P(deg) diagram

DIAGNOSTICS OF THE CYLINDER LUBRICATION SYSTEM

In a similar way to the operational diagnostics of the fuel injector, the electronic cylinder lubrication system can be examined with the proposed technology.

According to MAN Diesel [22, 23], the purposes of cylinder lubrication include the following:

- Reduction of the friction between the piston rings and the cylinder walls;
- Neutralisation of the acid properties of the fuel oil;
- Sealing of the piston rings, and elimination of the gaps between the rings and the cylinder walls;
- Cleaning deposits from the cylinder surface;
- Heat transfer from the piston to the cylinder walls and from there to the cooling liquid.

As leading engine manufacturers, WinGD and MAN see friction reduction and the avoidance of acid corrosion as priorities. To solve these problems, the lubricating oil must be injected with the proper pressure and timing by the lubricating oil injectors.

Modern cylinder lubricating systems with electronic control generally operate at an oil injection pressure of up to 40 bars over very short periods of time, when the piston rises on the compression stroke. The oil is injected just at the moment when the piston passes the position of the oil injectors, which are arranged in a circle in the liner of the cylinder. Injection starts when the first compression ring reaches the level of oil injector nozzles, and stops at the moment when the last compression ring passes by [23]. Since the oil injection period is very short (less than 10 *°CA)* and the piston speed is high at this point (14–20 m/s [4]), the importance of precise timing becomes obvious, and this imposes strict demands on the control system.

The discussion above explains why the mechanical cylinder lubrication system is ineffective for the engines of the S-MC(ME) and G-ME series manufactured by MAN, as even slight wear of the components of the mechanical lubricating system could cause shifting of the oil injection timing, meaning that the oil misses the pack of piston rings and takes place in an undesirable region. Incorrect timing could cause early oil injection above the piston, which could result in burning of the oil and the formation of deposits on the piston surface. Late oil injection would mean the oil is wasted, as it would enter under the piston space and could cause a fire hazard in the engine intake plenum. In both cases, the solid friction mode could occur, leading to a dramatic rise in the wear rate of the cylinder.

The Alpha Lubricator cylinder lubrication system for MAN ME and MC series engines includes an array of oil injectors arranged uniformly at the same height around the cylinder liner. It contains six injectors per cylinder for a cylinder bore of 50 or 60 cm, eight injectors for a cylinder bore of 65, 70 or 80 cm, and 10 injectors for a cylinder bore of 90 or 98 cm [23]. In other words, a typical six-cylinder engine may have between 36 and 60 oil injectors, and for the MAN 14K98ME-C engine, the total number is 140. With the application of the ball-type non-return valve in the oil injectors as shown in Fig. 7, the probability of several injectors malfunctioning in a given engine can be predicted as relatively high. An example of the installation of oil injection equipment is shown in Fig. 8.

Malfunctions in the ball non-return valves caused by long-term cyclic operation could result in loose closing, meaning that the high-pressure gases from the cylinder on the power stroke could enter the oil high-pressure pipe, and the next oil injection pattern could be distorted. The part of the power stroke in which the gases could enter oil injection pipe is shown in Fig. 7 by a dashed region.

Visual control of the operation of the injector according to the regular procedure is possible only during technical maintenance, through the transfer ports. In the testing mode, an inspector confirms the delivery of oil by each oil injector. Visual control requires a experienced mechanical engineer, is inconvenient, and can result in mistakes. Moreover, it can be performed only on a stopped engine, and the operating conditions of the oil injectors in the test mode are very different from those when the engine is running.

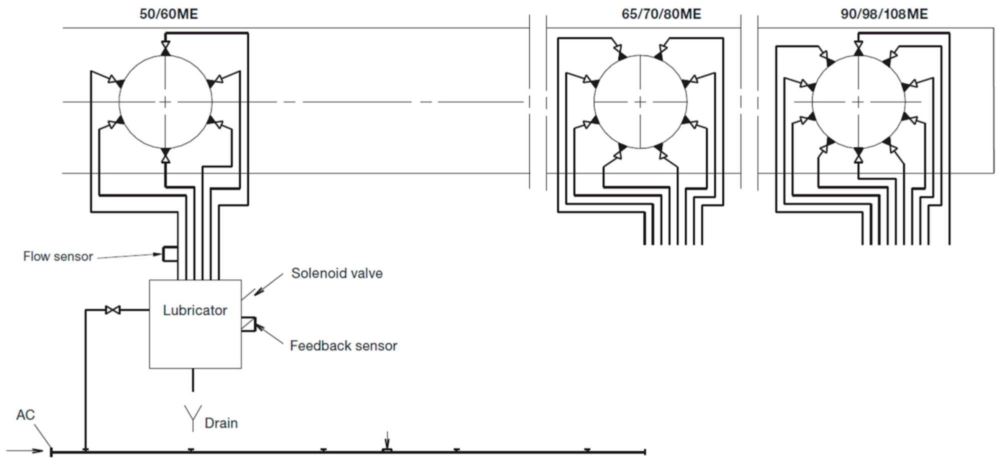

Fig. 8. Cylinder lubricator system for ME engines [23]

The oil injection control sensors, feedback sensor and level sensor shown in Fig. 8 may fail to detect exhaust gases in the high-pressure oil pipe. The signal from the vibration sensor on the oil injector offers a very promising solution in this case. The basic technique is similar to that described above for the diagnostics of fuel injectors. This type of control could be performed periodically during operation of the engine, and can help in the early detection of malfunctions of the oil injector, thus avoiding cylinder wear and engine damage.

An alternative approach to the problem of cylinder lubrication has been presented by Hans Jensen Lubricators [24]. The lubricating oil is injected into the engine's scavenging air swirl at the beginning of the piston compression stroke; in this way, the lubrication oil is diffused and optimum coverage of the cylinder wall is obtained, where the need for lubrication is greatest. The oil injection valves are designed similarly to the fuel injectors and are under electronic control. The manufacturer recommends that the injection valves should be checked at appropriate intervals, for example when renovating a cylinder or after 12,000–15,000 hours of operation. It is recommended that the shape of the oil spray, the closing pressure and the quantity of leaking oil should be checked about every two years. A valve control interval of 12,000–15,000 hours represents a long period of operation, and in our opinion, periodic vibroacoustic examinations of the injection valves will be very useful for this system in between the recommended checks.

An example of problems with cylinder lubrication is provided by a case involving the Mitsubishi 6UEC50LSH main diesel engine [25], with the critical wear of cylinder liners over one year of operation. The engine oil lubrication system was adjusted for minimal cylinder oil consumption based on the manufacturer's regulations, but the timing of the oil injection was set incorrectly by a couple of °CA. The unplanned cost for the replacement of all of the cylinder liners exceeded $120,000.

CONCLUSIONS

When in-cylinder pressure diagrams are recorded synchronously with the vibration sensor signals from fuel injection equipment and the valve train mechanism, they can provide valuable diagnostic information, as indicated by practical experience. The use of a vibration sensor with a magnetic base allows for the estimation of various timings, including:

- lifting and landing of the injector needle;
- circulation of heated heavy fuel oil ;
- fuel injection delivery;
- the opening and closing of exhaust valves;
- the injection of cylinder lubricating oil.

Mathematical simulation of the engine operation under steady and transient conditions is helpful in identifying malfunctions, analysing measurement data and predicting the effects of a malfunction on the performance of the engine. Moreover, engine transient simulations, which can be implemented using the Blitz-PRO online computation service, make it possible to predict malfunctions caused by irregular operation of engine systems, and particularly the fuel injection system.

The primary innovation of this study is a technique for vibroacoustic control of the technical state of each oil lubrication injector. Malfunctions of the ball-type non-return valve of an oil injector could cause penetration of the exhaust gas into the high-pressure pump of the oil injector. Oil injection malfunctions may result in extensive wear and damage to the engine, which can be critical considering the complex conditions of operation of the ship.

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