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THE DIAGNOSTIC SYSTEM OF PISTON ENGINES BASED ON UNUNIFORMITY IN SPEED

1. INTRODUCTION

Reciprocating internal combustion engines are increasingly becoming integrated systems of mechanical engineering, hydraulic and electronic components. The assessment of their mechanical condition as well as the localisation of malfunctions are therefore becoming all the more complex and require personnel of greater skills and knowledge.

One of the possible ways to solve the problem is the development of the so-called automatic diagnostic systems (ADS). The ADS concept means that the diagnostic parameters are obtained by measurements done by sensors placed on the objects, and the diagnosis is obtained on the basis of processing the data fed into the computer [1,2].

The development of sensors, electronics, i.e. microprocessors, and their increasing application, justify the efforts in developing diagnostic methods and techniques, based on the fast changing values of operating processes in the engine. These processes carry usually most information, but have been less used in practice due to the expensive and inadequate equipment as well as because of the need for a more extensive processing of measurement results. Today, because of the widespread application of microprocessors, this does not present a problem for the practical implementation.

Within the development of methods and techniques applicable to automatic diagnostics, this paper analyses the possibility of using the lack of uniformity of the crankshaft speed for the assessment of the overall mechanical condition of the reciprocating internal combustion engine. The diagnostics model development requires accurate knowledge of the output characteristics of the operating process, in order to define the diagnostic parameters in the best possible way. Therefore, a mathematical model for describing the lack of uniformity of the crankshaft speed has been developed as a simulator for diagnostics development. The model can simulate certain malfunctions, thus enabling a simpler and more efficient analysis of the influence of the system structure on the predefined parameters describing the lack of uniformity in speed.

2. THE MODEL FOR LACK OF UNIFORMITY IN CRANKSHAFT SPEED

The mathematical model for calculating the crankshaft speed for a one-cylinder internal combustion engine is being described. This can be later very simply extended to multi-cylinder engines.

2.1. Dynamics of engine mechanism

In order to grasp the causes and the essence of the lack of uniformity of crankshaft speed in an internal combustion engine, one should start from the general dynamic equation of a body with a changing moment of inertia revolving around a fixed axis, which is obtained from the formula for kinetic energy of an engine cycle as follows:

$$E_k(\alpha) = \int_{\alpha_1}^{\alpha_2} M(\alpha) \cdot d\alpha$$

By derivation of the left and the right side, we get

$$\frac{d}{d\alpha}\left[\frac{1}{2} \cdot J(\alpha) \cdot \omega^2\right] = M(\alpha)$$

where

α – is angle of crankshaft revolution

$J(\alpha)$ – is the moment of inertia of the moving parts of the engine reduced to the crankshaft axis

ω – is the angular speed of the crankshaft

$M(\alpha)$ – is the resulting torque of the forces acting on the engine elements, reduced to the crankshaft axis.

This analysis shows that the angular speed of the crankshaft changes during a single operating process. Its change is caused on one hand by the variable reduced moment of inertia of the reciprocating mechanism, and on the other hand, by the variability of the resulting torque reduced to the crankshaft axis.

In determining the character of the angular speed change, a hypothesis has been introduced that cycles repeat constantly in a succession in one cylinder, so that the variations in engine cycle are not taken into consideration.

$$\omega_i = \sqrt{\frac{2}{J_i} \cdot \left(\int_{\alpha_{i-1}}^{\alpha_i} M \cdot d\alpha + \frac{\omega_{i-1}^2}{2} \cdot J_{i-1} \right)}$$

Index *i-1* refers to the value of the corresponding magnitude at the angle of α_{i-1}, and index *i* to values at the angle α. The integral under the square root can be solved by one of the methods of numeric integration, i.e. by the trapezium rule:

$$\int_{\alpha_{i-1}}^{\alpha_i} M \cdot d\alpha = \frac{\Delta\alpha}{2} \cdot (M_{i-1} + M_i),$$

where

$M = M_\alpha - M_{otp}$ – is the resulting moment,

$\Delta\alpha = \alpha_i - \alpha_{i-1}$ – is the integration step.

In order to solve the above differential equation, it is necessary to have good knowledge of the resulting moment $M(\alpha)$ and the moment of inertia $J(\alpha)$, which is obtainable from the kinematic and dynamic analysis of the engine mechanism.

2.2. Kinematic and dynamic magnitudes of the engine

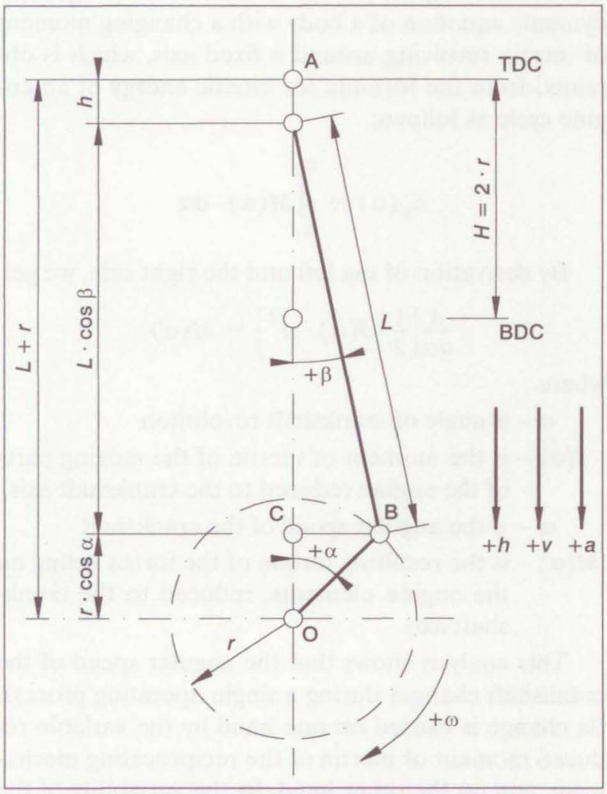

Figure 1 - Scheme of the piston mechanism

The reduced moment of inertia of the one-cylinder engine piston mechanism is obtained from the condition of equality of the kinetic energies of the actual mechanism and its reduced model. For the practical application, a simplified expression is used for the calculation of the reduced moment of inertia:

$$J(\alpha) = J_{kv} + r^2 \cdot m_r + r^2 \cdot m_t \cdot \sin^2\alpha$$

where:

J_{kv} – is the moment of inertia of the crankshaft with all the related elements,

m_r – is the part of the crankshaft mass that performs the rotation,

m_t – is the mass performing translation, consisting of the mass of the piston group and a part of the crankshaft mass that performs translation.

For calculating the resultant moment, it is necessary to analyse all the forces and moments which act in the piston mechanism. The analysis of the idling engine is accepted. This engine operating regime is the most suitable for the needs of diagnostics in cases when the engine has not been dismounted. In idling regime the total indicated work produced during a single cycle is spent on overcoming the inner losses in the engine, and the resultant moment $M(\alpha)$ at any arbitrary moment equals:

$$M(\alpha) = M_{pl}(\alpha) - M_{otp}(\alpha),$$

where:

$M_{pl}(\alpha)$ – is the moment of gas pressure forces,

$M_{otp}(\alpha)$ – is the moment of inner resistance.

Since the whole indicated work from the gas forces is spent on inner losses, the moment of inner resistance equals:

$$M_{otp} = \frac{1}{4\pi} \cdot \int_0^{4\pi} M_{pl} \cdot d\alpha$$

Moment M_{pl} which is caused by the gas force F_{pl}, can be obtained by reduction of this force to the moment which acts on the crankshaft. Taking into consideration also the kinematic relations in the piston mechanism shown in Figure 1, the following is obtained:

$$M_{pl} = F_{pl} \cdot r \cdot K$$

where:

$$K = \sin\alpha + \frac{\sin 2\alpha}{2 \cdot \sqrt{(1/\lambda)^2 - \sin^2\alpha}}$$

$$\lambda = \frac{r}{l}$$

The gas force equals:

$$F_{pl} = p_{pl} \cdot \frac{D^2 \cdot \pi}{4}$$

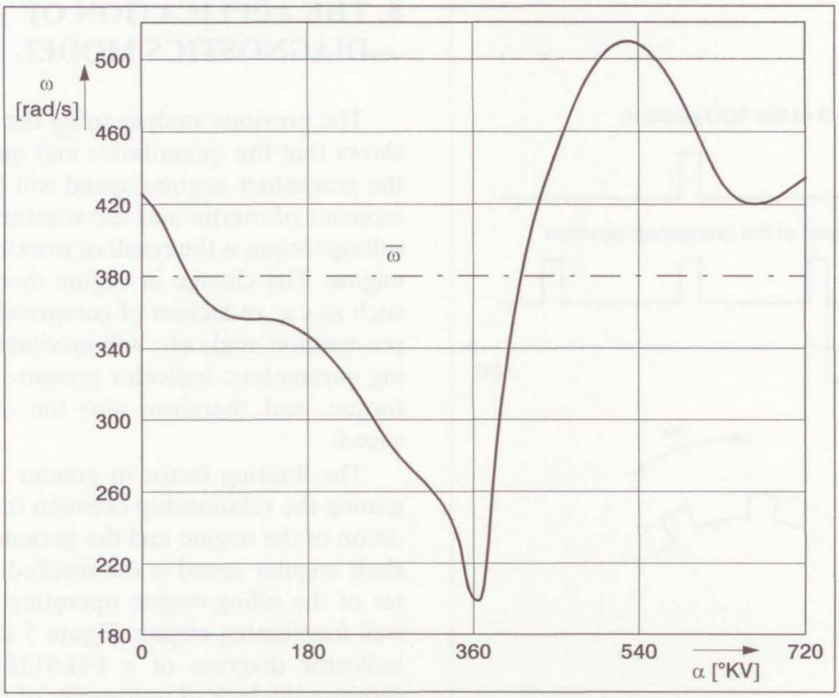

Figure 2 - The change of one-cylinder engine angular speed

Gas pressure acting on the piston p_{pl} equals:

$$p_{pl} = p_i - p_k,$$

where:

p_i – is the indicator pressure in the cylinder of the engine,

p_k – is the gas pressure in the oil-sump.

For the needs of the model, the indicator pressure can be obtained by recording the pressure in the engine cylinder or by modelling. In this work the indicator pressure has been modelled, using the Vibe model for describing the combustion process.

Thus it was possible to solve the initial differential equation and to obtain the angular speed of the crankshaft depending on the change of the crankshaft torque angle.

3. THE POSSIBILITY TO DETERMINE THE ANGULAR SPEED IN PRACTICE

The angular speed can be determined in practice in a very simple way shown in Figure 3.

In order to determine the position of the top dead centre (TDC) and the crankshaft angular speed, inductive sensors were used. The angular speed for a certain angle α is determined numerically on the basis of the record analysis shown in Figure 4, according to the following:

$$\omega = \frac{\Delta\alpha}{\Delta t}$$

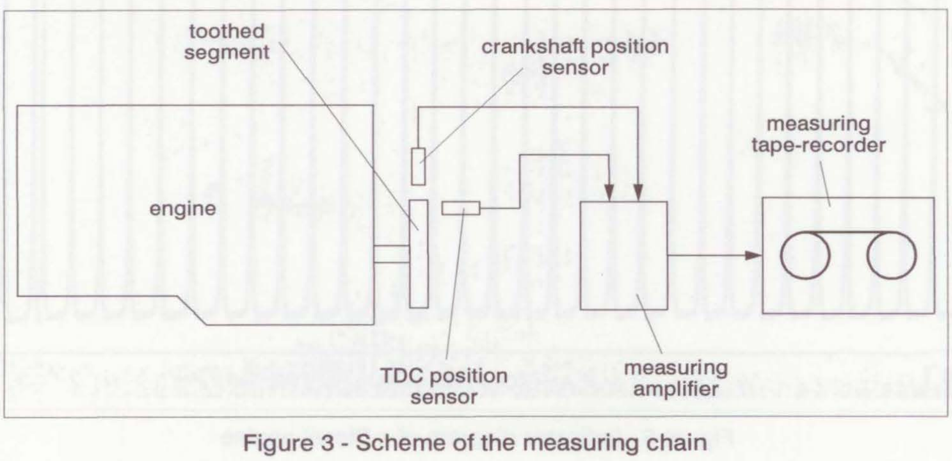

Figure 3 - Scheme of the measuring chain

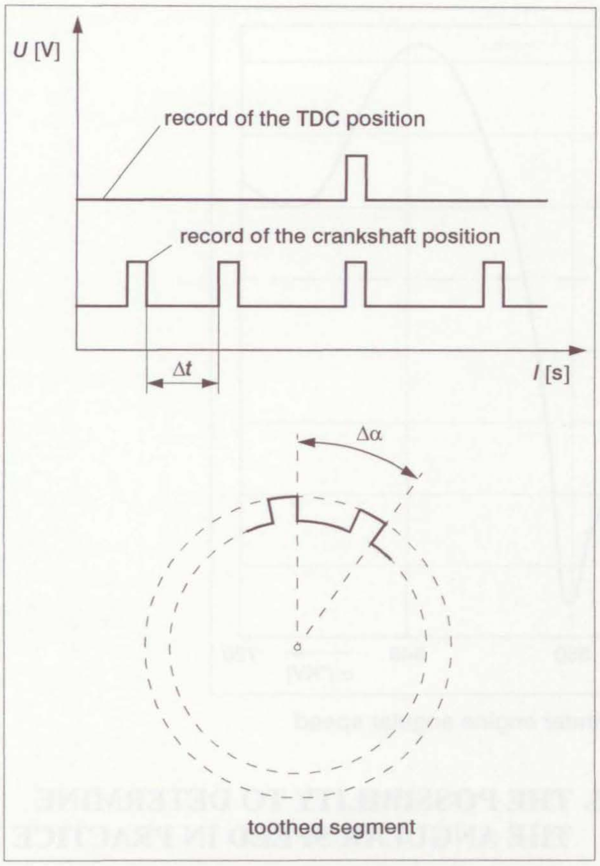

Figure 4 - Determining of the angular speed

4. THE APPLICATION OF THE DIAGNOSTICS MODEL

The previous analysis using the theoretical model shows that the quantitative and qualitative image of the crankshaft angular speed will be affected by the moment of inertia and the resulting torque. The resulting torque is the result of working processes in the engine. The change in engine mechanical operation such as e.g. reduction of compression, change of the pre-ignition angle etc. will inevitably affect the working parameters: indicator pressure and the resultant torque, and therefore also the crankshaft angular speed.

The limiting factor in greater accuracy of determining the relationship between the mechanical condition of the engine and the parameters of the crankshaft angular speed is the marked stochastic character of the idling engine operating process even in a well functioning engine. Figure 5 shows the recorded indicator diagram of a F4L413FR Diesel engine, showing the lack of uniformity of the indicated pressure in the cylinder.

The malfunction of the first cylinder in a 4-cylinder engine has been simulated. A malfunction was supposed regarding compression reduced to 60% from the nominal pressure. The angular speed diagram has been shown for one single revolution of the

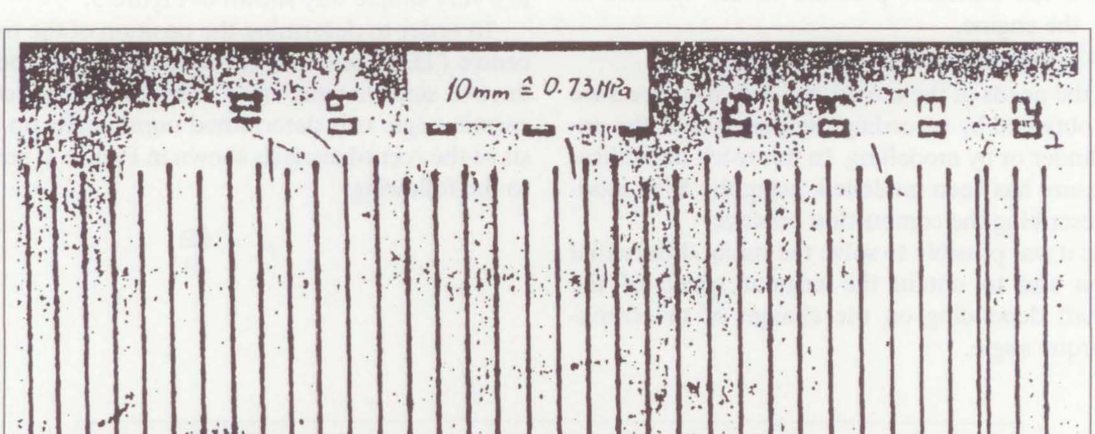

Figure 5 - Indicator diagram of a Diesel engine

crankshaft without the influence of lacking uniformity of the indicator pressure.

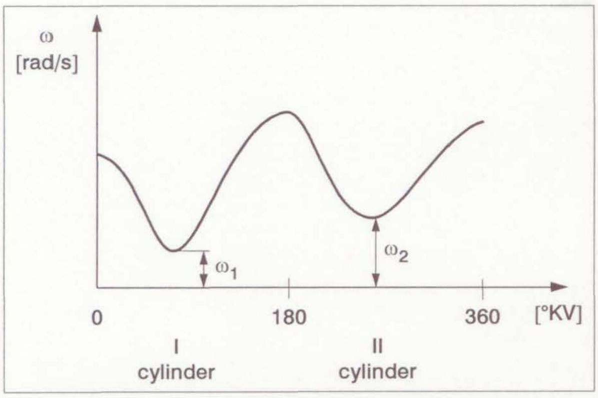

Figure 6 - Crankshaft angular speed diagram of a malfunctioning engine

Figure 6 shows that on the basis of the minimum angular speed value ($\omega_1 \neq \omega_2$) during a single operating process, malfunctioning of certain piston engine cylinders may be assessed.

5. CONCLUSION

The mentioned theoretical analysis has justified the hypothesis that the piston engine crankshaft angular speed has its diagnostic value in assessing the overall mechanical operating condition. The angular speed values can be recorded by simple inductive sensors and displayed on the screen of a diagnostic device. As the inductive sensor trigger (Figure 4), also a toothed crankshaft flywheel row may be used.

Following the theoretical analysis performed on a mathematical model - simulator, it has been proven that it is justified to continue with the experiments regarding application in practice, where problems can be expected with the influence of the stochastic character of the indicator pressure and therefore also the stochastic character of the speed.

SUMMARY

The work analyses the possibility to diagnose the overall operation of the piston engine based on the crankshaft angular speed. The testing was carried out on the basis of the mathematical model-simulator of malfunctioning. The possibility for diagnostics based on the lack of uniformity of the crankshaft speed has been confirmed and the continuation of experimenting in practice with the proposed method justified.

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