DETERMINATION OF THE MECHANICAL POWER LOSSES OF ENGINE-COMPRESSOR GMVH FOR DIAGNOSTIC PURPOSES

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Abstract

The engine-compressors GMVH types are used in the national gas transmission system in Poland. There is a danger of scuffing of the working components of the engine-compressor particular pistons and cylinders during the process of their exploitation. The growth of resistance in motion parts of the machine is the main symptom of the scuffing. In the cinematic system of the engine-compressor where the engine and compressor are fully mechanically integrated it is impossible to assess the power of mechanical friction without stopping the machine. But if the engine-compressor is equipped with a cylinder pressure measuring system it is possible to determine the indicated power of all compressor and engine cylinders and assess the friction losses as a difference between the sum of the engine's cylinders indicated power and the sum of the compressor's cylinders indicated power. In the differential measurements of absolute values measuring errors are accumulated so considerable uncertainty of the measured parameter is expected. The accuracy of the friction power losses assessment method is not sufficient for research purposes. However, despite the expected significant errors, assessed parameters trends observation should allow to detect of rapid changes of machine technical condition caused by piton scuffing. To ensure the sense of this method a precise measurements and in particular the error consistency of indicated pressure assessment of all cylinders in long time period are necessary. The experience from engine-compressors monitoring and diagnostic system services, the main sources of errors and methods for their minimization are described in the paper.

Keywords: *combustion engines, engine-compressor, indicating, MIP, pressure sensors.*

1. Introduction

In the national natural gas transmission and processing system a large number of engine-compressors of GMVH type are used.

Fig. 1. An engine-compressor installed in one of the gas transmission stations

The engine-compressor is an integrated multi-cylinder system in which the engine and the compressor pistons cooperate with one common crankshaft. Type GMVH of engine-compressor is made in the configuration of incomplete star. It is characterized by the fact that the double-acting crosshead compressor works with the main connecting-rod, and two engine pistons cooperate with the articulated connecting-rods. Axes of the engine cylinders are shifted by 60˚ and 120˚ respectively to the axis of the compressor cylinder. Diagram of a single crank of crankshaft is shown in the Fig. 2.

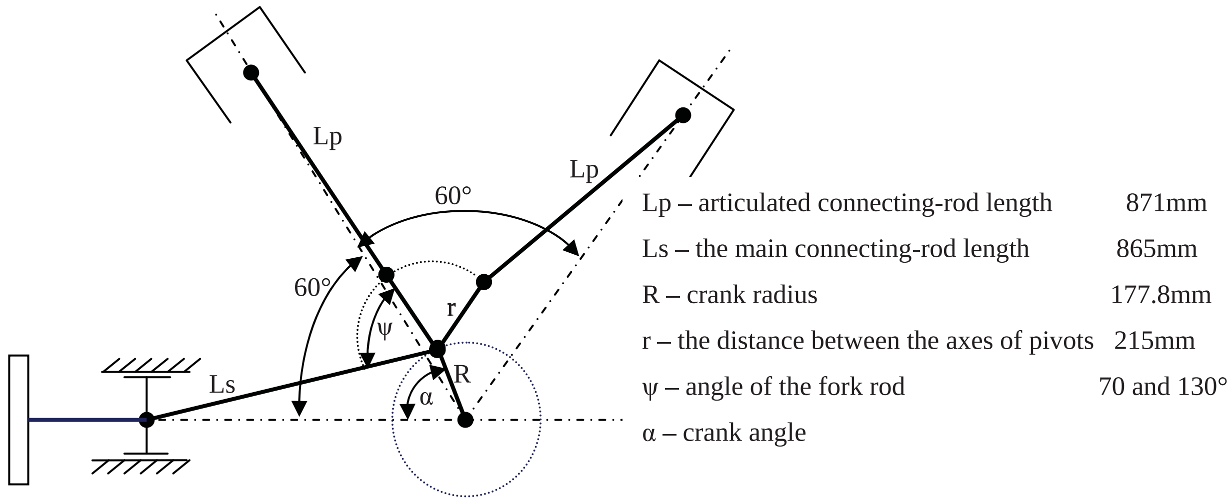

Fig. 2. Diagram of a single crank of crankshaft of engine-compressor type GMVH

In the neighbouring crank the identical crank modules are mounted but with the free crosshead, i.e. without a compressor module. This system may be freely multiplied, but in Poland there are engine-compressors only with doubled and tripled such systems. Because the originally set up periods of operation for pumping stations have been exceeded many times, further operation of that system can take place only under special technical supervision. For this purpose at all pumping stations monitoring systems of engine-compressors performance are developed.

2. Engine-compressors indicating system

Currently parameters such as cooling water temperature and cooling water flow, bearings temperature, lubricating system parameters such as oil pressure and oil temperature, exhaust gases temperature, transferred gases parameters, and many other parameters, mainly of low frequency changes are monitored. In the case of certain malfunctions changes in the measured parameters may be too small and too slow to make the correct intervention. Hence, already in 1998, emerged in the Naval Academy the idea of building a system for continuous monitoring of cylinder pressure in all engines and compressors cylinders. Developed conceptual system project after modifications, arising mainly from the new components availability on the market, was implemented by the plant Zamtech KRIO in Odolanów in 2008. The implemented monitoring and diagnostics system makes possible the simultaneous measurements of pressure in 12 engine cylinders and 6 compressor cylinders in each of the five engine-compressors. Simultaneous measurements are necessary because of the uniqueness of ignition and significant spread of indicated power in the individual cycles of engine operation. This system is currently working on a continuous basis and is used mainly to control and regulate load balancing in all of the engine cylinders.

In the cylinder pressure measuring and monitoring system the specific requirements to the pressure sensors, which are typically mounted on the engines indicator valves are applied. Under conditions of continuous measurements sensors are subjected to high temperatures, variable loads, impacts of aggressive gases and are polluted. Sometimes sensors are subjected to mechanical and thermal overload, especially in the case of gas engines, where knocking combustion can occur. In the case of pumping stations of natural gas additional significant intrinsic safety standards have to be fulfilled. Taking into account these requirements the possibility of sensors choice is very limited. Usually for the professional engines indicating systems piezoelectric sensors of AVL and Kistler firms are used. These manufacturers cannot guarantee safety for the use of their sensors in the explosive zones. So, it was decided to choose fibre-optic sensors from firm Optrand type C82283-Q. The sensors used in the sensing element devoid of any electrical circuit are connected to a fiber optic cable with a length of 2 meters with converter and amplifier. Amplifiers were placed in housing in which the air from outside the explosive zone is blown into. The method of sensors and amplifiers montage is shown in the Fig. 3.

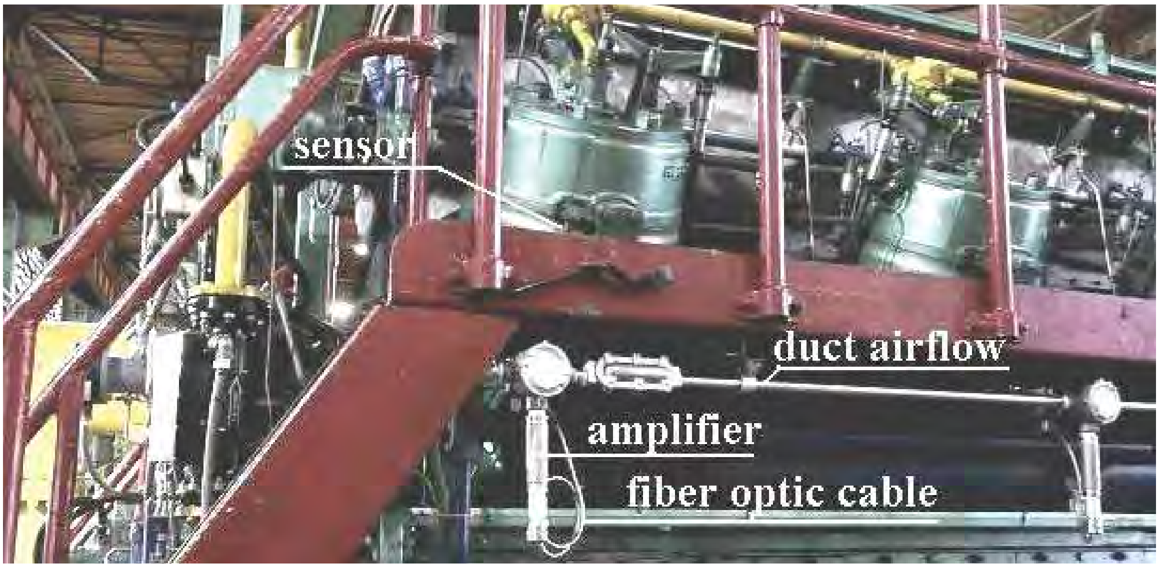

Fig. 3. The method of sensors and amplifiers montage

Fiber optic cable provides optic-isolation eliminating from the slotted line isolating amplifiers. Isolating amplifiers in the slotted lines with sensors of other types have to be used despite the fact that they would be a source of serious measurement errors. Durability of pressure sensors for gas engines has been determined by the Optrand firm as $\geq 2 * 10^8$ loading cycles at a temperature not exceeding 350˚C at sensor membrane. The measurement range of the sensor in a brief time could be four times exceeded according to the technical data. It follows from this that these sensors can work without faults. The Optrand sensors are dynamic type so the constant component of the measured value is not indicated. In terms of assessing of: mean indicated pressure and indicated power of engine-compressor unit lack of constant component of the values does not matter.

To determine the value of mean indicated pressure it would be preferably to perform pressure measurements in the domain of the piston path. In the operating conditions such measurements are not possible. Measurements are usually performed in the time or in the crank angle domain of rotating crankshaft. In the developed measuring system sampling of cylinder pressure is controlled by 0.5˚ of crank angle pulses generated by a software implementation of the PLL system based on the pulses from the ignition system.

3. The mechanical power losses calculation

The growth of resistance in motion parts of the machine is the main symptom of the scuffing. In the cinematic system of the engine-compressor where the engine and compressor are fully mechanically integrated it is impossible to assess the power of mechanical friction without stopping the machine. But if the engine-compressor is equipped with a cylinder pressure measuring system it is possible to determine the indicated power of all compressor and engine cylinders and assess the friction losses as a difference between the sum of the engine's cylinders indicated power and the sum of the compressor's cylinders indicated power. Assessed observations of parameters trends should allow to detect rapid changes of machine technical condition caused by scuffing. In the differential measurements of absolute values measuring errors are accumulated so considerable uncertainty of the measured parameter is expected. To ensure the sense of this method a precise measurements and in particular the error consistency of indicating pressure assessment of all cylinders in long time period are necessary.

3.1. Mean indicated pressure calculating method

To determine the mean indicated pressure and as a consequence to determine indicated power the trapezoids method was used. During each complete crankshaft revolution dimensionless (relative) path of the piston is multiplied by the mean pressure values between adjacent pressure samples and calculated as a sum of the sub products. Trapezoidal integration method with a resolution of 0.5° of CA gives negligibly small errors of integration. It is important to calculate the results according to the formula (1)

$$p_i = \sum_{k=1}^{N/2} \left(\frac{p_{(k)} + p_{(k+1)}}{2} \cdot \left(S_{(k)} - S_{(k+1)} \right) + \frac{p_{(N-k)} + p_{(N-k+1)}}{2} \cdot \left(S_{(N-k)} - S_{(N-k+1)} \right) \right), \tag{1}$$

where:
N - number of samples per revolution of the crankshaft
S - relative piston path calculated from the piston BDC
k - number of pressure sample

From a mathematical point of view, the same effect can be obtain from the simpler formula (2).

$$p_i = \sum_{k=1}^{N} \left(\frac{p_{(k)} + p_{(k+1)}}{2} \cdot \left(S_{(k)} - S_{(k+1)} \right) \right). \tag{2}$$

Due to the interpretation of the number of floating-point by the computer and the way of the number rounding this method would have had bigger error. In the first case due to the symmetry of the calculation alternating subtraction and addition occurs so that the partial sum in the calculation never significantly exceeds the final calculated value.

A key problem in the integration is the samples number conversion on the path of the piston. Compressor kinematic system is a simple central type system and formulas to convert the angle of rotation of the crankshaft into piston path are simple and well known [4]. However engine connecting-rods are the articulate type and the conversion the angle of rotation of the crankshaft on the piston path is much more complex and required separate calculations for the left and right cylinder banks. To determine the angle of rotation of the crankshaft a reference (marker) pulse from the ignition system was used. This pulse occurs once for each rotation of the crankshaft. The piston path calculations should only take into account the angular distance of each cylinder TDC from the reference marker.

It is important to determine the distance with the accuracy better than the angular resolution of measurements timing. In the completed system obtained accuracy was not worse than 0.1° of CA. For example, in typical marine diesel engine the maintenance of such accuracy is virtually impossible because of the torsion in the crankshaft which depends on the load and torsional vibrations [1]. The connecting rods in the engine-compressor are located on the same crank of the crankshaft so crankshaft is loaded only by a small torque value between adjacent cylinders, mainly due to uneven regulation of individual cylinders of the engine-compressor. It was therefore assumed that the shaft during operation is only slightly twisted and setting (during the calibration) angular distance from the reference point to the TDC does not change.

3.2. A TDC determination

Determination of the TDC on the pressure curve of the engine cylinder is relatively simple. The easiest way is to cut-off gas access into the cylinder to find the point of maximum cylinder pressure. Using the polynomial approximation of the pressure trace the apex location obtained accuracy can be greater than the angular resolution of the recording equipment. A similar result can be obtained by examining the derivative of pressure curves. The derivative of pressure curve goes through the 0 value at TDC even if the fuel is delivered into a cylinder if only for the time of calibration ignition will be maximally delayed [2]. In such a way designated TDC differs from the actual value by angle of the losses, which in the new engine is small and based solely on the heat transfer during compression. During engine-compressor operation the angle of losses increased due to increased leakage from the combustion chamber and sometimes becomes unacceptable source of error [3]. During the installation of the measuring system on the worn out engine calibration of the TDC can be determined mathematically. For this purpose a theoretical model of the compression process was prepared which can be used to determine the theoretical pressure curves during the compression stroke. The model gives possibility to simulate the conditions of loading and wear of engine components that affect the pressure value at the end of the compression stroke and the shape of the pressure curve throughout the compression process. If the curve (result of theoretical simulation) is aligned with the part of measured pressure curves (real curves) limited to the period from the outlet windows covering until the time of fuel ignitions it can be assumed that the loading conditions and the simulated malfunctions have the same effect as the actual working conditions. Then the extrapolation of such compression theoretical curves designated in the combustion area enables to assess the pressure value and the location on the graph the point in which the piston reaches the TDC without cutting off the fuel supply. Determination of the TDC on the average compression process curve by extrapolation is shown in the Fig. 4.

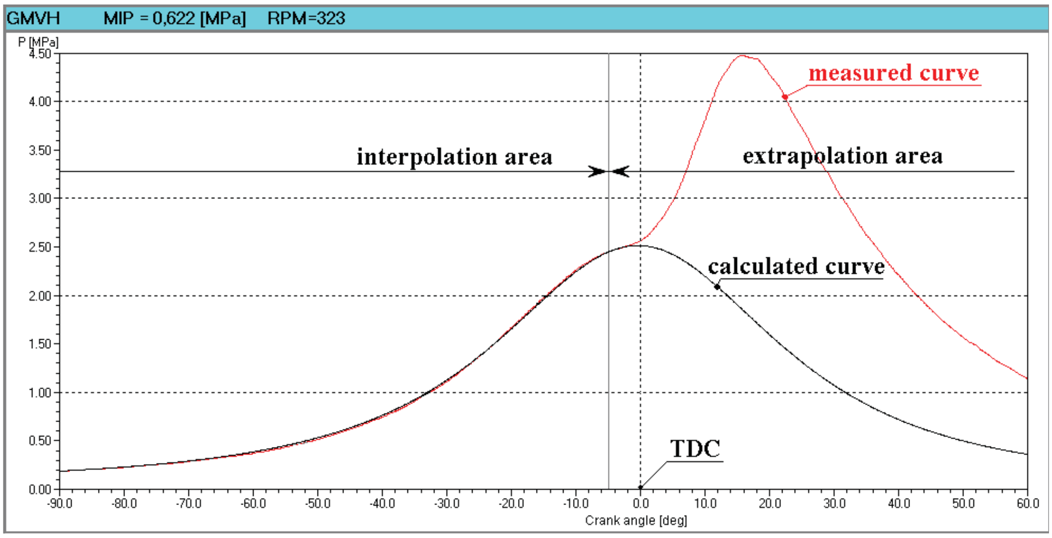

Fig. 4. Determination of the TDC by extrapolation of an average compression pressure curves

Definitely the more difficult is to determine the TDC from pressure curve in the compressor. During the compression the discharge valve is opened before the piston reaches TDC which result in pressure drop. The TDC can be determined by the compression curve extrapolation in the area from BDC to the discharge valve opens but the long distance from the opening of the discharge valve to the TDC requires precise extrapolation. An attempt to extrapolate the compression curve using the polytrophic curve is shown in the Fig. 5.

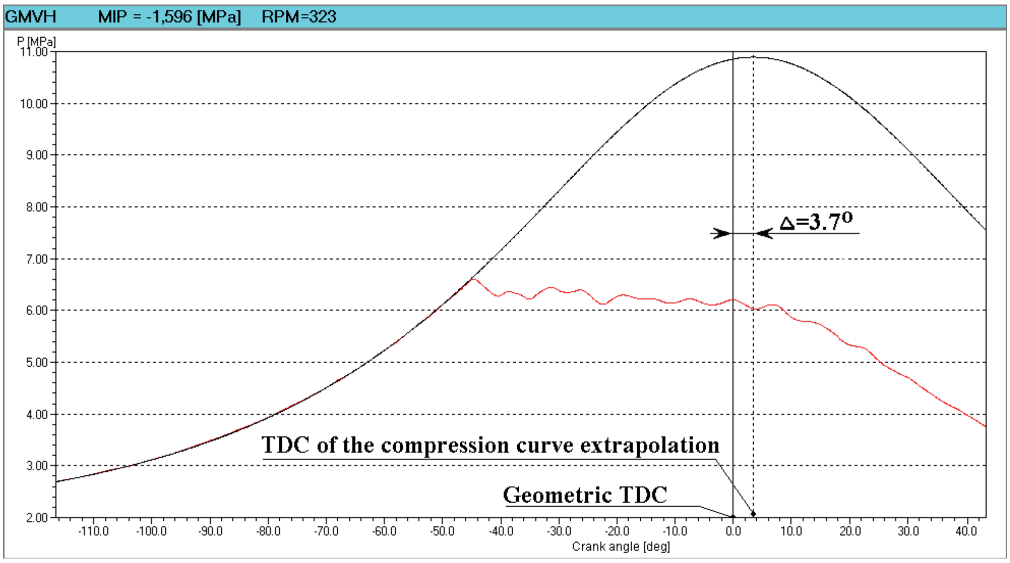

Fig. 5. An attempt to determine of the TDC by polytrophic compression curve extrapolation

To ensure the required accuracy the extrapolation should be perform on the theoretical compression curve. The polytrophic curve does not provide the required accuracy. The compressed gas occurring in the gas system should be regarded as a solution of higher hydrocarbons vapors in a mixture of nitrogen and methane. In the absence of information about the current gas composition and the actual compression ratio this method of exploration has been abandoned as not prospective. As more realistic the expansion curve at the return piston movement was used. Expansion begins just after the TDC and discharge valve close so the required range of the extrapolation would be not too far. Unfortunately, the expansion curve shown in the Fig. 6 is too much distorted by the wave effects in the indicator channel so required accuracy of extrapolation is not easy to obtain.

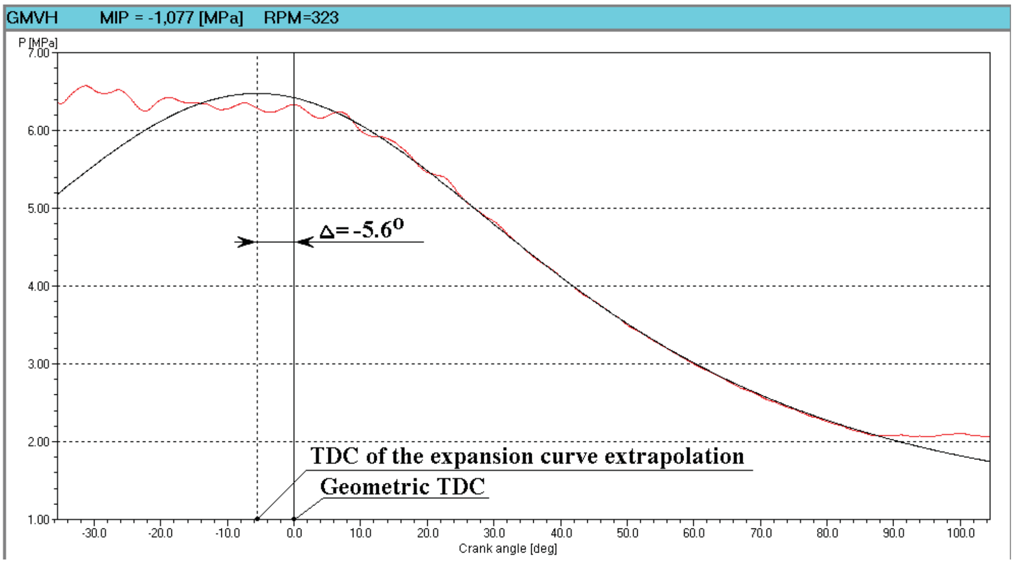

Fig. 6. The expansion curve distortions

From the analysis of the assembly of the crankshaft which shown in the Fig. 2 it is seen that despite the asymmetry of the crankshaft assembly the main connecting rod length ratio λ, the angle of the fork cylinder axis φ and the angle of crank webs ψ are selected in such way that each cylinder TDC is shifted by the angle of the cylinder axis, i.e. the 60˚ and 120˚

If the TDC is determined for the engine cylinder placed on the common crankshaft web with the compressor it is assumed that the TDC of an external compressor is located at 60˚ after engine cylinder TDC of right bank and 120˚ after TDC for left engine cylinder bank. For the external compressor TDC mean arithmetic values were adopted. Due to the symmetry of the compressor crankshaft it was assumed that the internal compressor TDC is shifted by 180˚ in comparison with the external compressor TDC. This approach seems to be reasonable if only the clearances in the crank system are not increased beyond permissible values. At the time of the measurements a signal delay occurs because of the indication valves channels length. Since in the engine-compressor cylinders are used the same indication valves it can be assumed that the errors generated by the channels are partially compensated.

4. Exploitation experiences

Sensors fixed on the engines indicating valves have had high failure rate in the operation. Most of failures were the result of the errors in sensor membrane assembling process (Fig. 7) made by the manufacturer. In addition to mechanical damage of sensors a thermal damage also occurred. Measurements of the temperature, even under the gas blow (repealed valve) did not confirm excess of the declared temperature boundary value of 350˚C. Only 230˚C was reached. The manufacturer claims that the conditions for the sensor destruction may arise in the event of knock combustion. The same position in term of the sensors has also Kistler Company. In this situation in the front of each sensor membrane on the engines special flame suppressors offered by Optrand were used (Fig. 8).

Fig. 7. The Optrand pressure sensor membrane damage

Fig. 8. The flame suppressors to the Optrand pressure sensors

Flame suppressors prevent sensors damage but after some time pollute which is manifested by a decrease in amplitude and phase delay of pressure signal. In this situation indicating diagrams of engine cylinders lean to the right resulting in an overstatement of the determined engine indicated power at the same (not changed) indicated power of compressors. This apparent increase in the efficiency of the whole machine even above 100% made the correct diagnostic process impossible. There are plans to introduce software detection and signalling of this phenomenon and its automatic correction. This will require regular simultaneous measurements with Optrand sensor permanently mounted and equipped with a flame suppressor and Kistler sensor mounted temporarily on the indicating valve. Analysis of the occurred apparent shifting of the TDC as a result of the suppressor pollution and false increase in the indicated power enables the development of the amendment algorithm. After reaching a certain level of pollution replacement of the contaminated suppressor will be required as it is easy to carry but if too frequent exchanges can be cumbersome and laborious. Perhaps the improvement could be achieved by changing the fiber-optic Optrand sensors by the piezoelectric Kistler sensors, which, despite various manufacturers declarations appear to be more durable than Optrand sensors. High durability of Kistler sensors type 6613CA was confirmed by 20 years of reliably use them through the diagnostic team at the Polish Naval Academy for the indication of many engines on the Navy ships. There is no assurance, however, that without the flame suppressor sensor will work properly under continuous operation conditions. This would required a separate studies.

5. Conclusions

1. Monitoring of the friction losses power in the engine-compressor unit is possible but difficult because of the required accuracy of indicated power determination for separate cylinders.
2. Due to the uniqueness of the combustion pressure curves the measurements have to be carried out simultaneously across whole engine and averaged over several revolutions of the crankshaft.
3. For high assessment accuracy of the friction losses power angular and amplitude resolution of used equipment is not critical. System measurements timing (trigger) resolution of 0.5˚ in CA domain and 12-bit amplitude processing are sufficient.
4. The biggest impact on the measurement accuracy has sensor linearity and a determination of the TDC.
5. The basic difficulty in ensuring the repeatability of measurements is to ensure sufficient stability of measurement channels from indicator channel patency up to the long term stability of the sensors and amplifiers.

References

[1] Łutowicz, M., *Evaluation of the Condition of Cylinder Systems of the Engine Based on an Compression Process Analisis*, Polish Journal of Environmental Studies, Vol. 18, No. 2A, 2009.
[2] Polanowski, S., *Determination of location of Top Dead Centre and compression ratio value on the basis of ship engine indicator diagram*, Polish Maritime Research, No. 2(56), Vol. 15, 2008.
[3] Tazerout, M., Le Corre, O., Rousseau, S., *TDC Determination in IC Engines Based on the Thermodynamic*. SAE Paper No. 1999-01-1489.
[4] Wimmer, A., Glaser, J., *The indicated engine,* AVL, Warsaw 2004.