THE MEASUREMENT OF LOSSES GENERATED IN KINEMATIC NODES OF ENGINE RUN AT TEST STAND

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Abstract

Reduction in losses accompanying the operation of IC engine requires previous identification of their sources and estimation of their value. Reduction in friction losses that is the most important component of the resistance to motion is particularly substantial. In following study the authors presented the best known methods of friction losses estimation pointing out their advantages and shortcomings. In authors' opinion the method of balance of torques generated during engine run is the most accurate one. The basic source of error in this method is an additional torque due to the torsional vibrations. Finally, torsional vibrations calculations concerning the VWR5 TDI diesel have been presented as an example.

Keywords: combustion engine, friction losses, mentioned methods, torsional vibrations

1. Introduction

Energy generated during fuel combustion in engine cylinder is higher than that transmitted to the receiver of a part indispensable for overcome the resistance in kinematic nodes and necessary for driving the auxiliaries. Difference between the indicated mean effective pressure (p_i) and the brake mean effective pressure (p_e) is so called resistance to motion mean pressure (p_m) and can be expressed in following formula:

$$p_m = p_i - p_e, \tag{1}$$

or expressed by the relation of relevant torques (or power):

$$M_m = M_i - M_e, \quad N_m = N_i - N_e, \tag{2}$$

where:

M_i, N_i – indicated torque and power,
M_e, N_e – effective torque and power,
M_m, N_m – torque and power due to the resistance to motion.

The concept of mechanical efficiency η_m is used in order to define a relative loss of energy necessary for overcoming engine natural resistance, at the same time proving the correct collaboration and quality of lubrication of engine moving parts. It can be expressed as follows:

$$\eta_m = \frac{N_e}{N_i} = \frac{N_e}{N_e + N_m}. \tag{3}$$

The total power of resistance to motion denoted as N_m contains losses that could be divided into five basic groups.

$$N_m = N_t + N_w + N_p + N_k + N_d. \tag{4}$$

Friction losses (N_t) are generated in engine kinematic nodes, above all in piston-cylinder assembly, main and crank bearings, and camshaft bearings. As tests show, they contribute for about 75% of all losses generated during engine run. Their increase proves worsening the conditions of collaboration in kinematic pairs due to wear of mating surfaces, insufficient lubrication, aging of lubricating oil and so on.

Charge exchange losses (N_w) caused by negative pressure acting on the piston during aspiration stroke as well as overpressure during exhaust stroke, what corresponds to the hydraulic resistance in inlet and outlet systems. The hydraulic resistance depends on construction of manifolds, condition of air filter and muffler.

Auxiliaries drive (N_p) requires power for driving such systems as fuel system, lubricating system or cooling system. Necessary energy is in most cases taken from the crankshaft.

Ventilation losses (N_k) include effect of surroundings on engine moving parts, i.e. all what is connected with pumping gases by pistons in crank case as well as air drag of connecting rod, flying wheel and attached elements.

Blower or compressor drive (N_d) present first of all in two stroke engines and engines mechanically supercharged (compressor driven by the crankshaft).

It is obvious that friction losses have the most considerable share in total resistance to motion. In order to determine these losses different methods are applied, but only few of them are suitable for determination of their variations within the entire cycle. A precise recognition of friction losses changes, in particular those relative to the run of individual engine subassemblies allows to point out necessary modifications in their construction and conditions of operation.

That is why the authors have analyzed the so far utilized methods of their determination, evaluating their suitability for planned tests.

2. Methods of friction losses measurement

There are computational and experimental methods of friction losses (expressed by forces, moments or power of friction) determination. In the case of computational ones the friction losses generated in engine individual subassemblies are determined using appropriate models of phenomena and processes, and eventually summarized. Accomplished results are as correct as precise are mathematical models and input data.

The experimental methods consist in the measurement of friction losses, carried out on running engine or a physical model of selected subassembly. The examples of application of both methods to particular engine kinematic nodes one can find in [3].

Methods utilizing the direct measurement of quantities needed for calculation of the resistance to motion (e.g. using formula 2) as well as those using indirect measurement of selected engine performance data and its dynamic properties can be classified as the experimental methods.

The method utilizing parallel engine indicating (p_i) and the measurement of break mean effective pressure (p_e) can be classified as the most accurate method of direct determination of

friction losses. Instantaneous value of cylinder pressure are measured during engine indication and eventually the value of indicated mean effective pressure and indicated torque are determined. For a measurement of effective power the test stand brake or torque meter fixed to the drive shaft can be utilized. This method can be applied to the one or multi cylinder engines but for the latter case differentiation of phenomena in individual cylinders might affect the accuracy of the method (higher accuracy could be attained indicating cylinders individually).

Method of:
– engine motoring,
– combustion switch off in consecutive cylinders,
– run off,
– load characteristics,
can be classified as the indirect methods.

The method of engine motoring consists in the forcing of engine shaft rotation with the external drive. An electrical motor which during other measurements can serve as combustion engine brake is a part of the test stand. When carrying out the measurements, the torque needed for crankshaft drive with a given rotational speed is being determined. The value of torque can be found through the measurement of energy consumed by the motor or measuring the moment of reaction acting on a swinging brake housing. Results obtained using this method are considered as inaccurate due to the lack of combustion. These changes affect to the considerable measure the collaboration between the mating elements and finally the value of resistance to motion.

However, use of other, earlier mentioned methods is restricted with various limitations and brings about considerable errors. For example, the method of switching off cylinders can be applied to multi cylinder engines and friction losses found concern fireless cylinder. The run off method requires in turn a precise knowledge on the dynamic properties of engine and its subassemblies.

3. Measurement of coupling torque between engine and power receiver

A number of measurements within a single engine cycle is necessary in order to define an instantaneous value of losses on a running engine using the method of indicated and load moment direct measurement. The measurement of pressure variations in engine cylinder require the use of typical electronic measurement system. On the other hand, an accurate measurement of load moment is quite a problem [1]. Most of the test stands equipped with electrical brakes allow a precise measurement of braking moment or power, but the result is the average one and the measurement of momentary value is impossible. However, the measurement of an instantaneous value of torque is possible using the modern torquemeters. The principle of their operation consists in a measurement of changes in a chosen physical quantity, relative to a selected section of coupling shaft (a part of torquemeter) as stress (using tensometer measuring system, see Fig. 1) or angle of torsion (Fig. 2). The angle of shaft torsion is proportional to the value of torque M (also called the coupling torque) as in Eq. (5):

$$M = \frac{G \cdot I \cdot \varphi}{l} \ ,$$

(5)

where:
l – distance between terminal cross sections,
φ – angle of torsion of shaft measuring section,
G – modulus of rigidity,
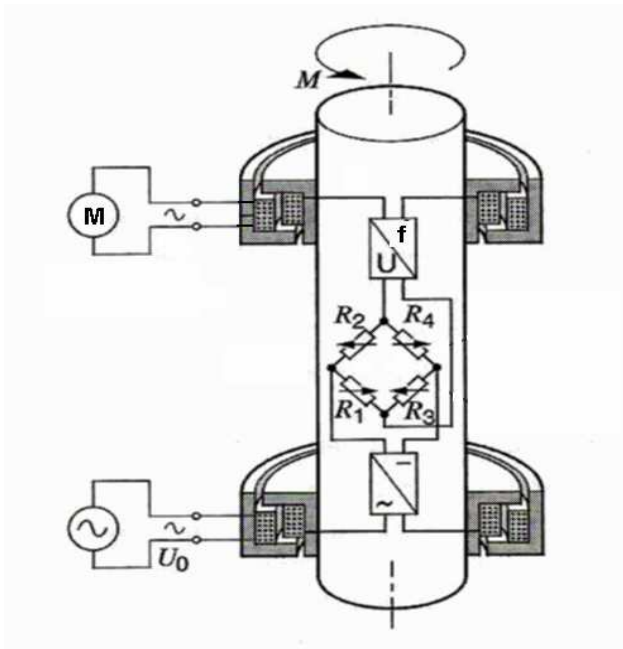I – moment of inertia.

Fig. 1. Tensometric torquemeter with contactless, transformer signal transmission: M – torque, U_0 – supply voltage, $R_1 \div R_4$ – tensometers [4]
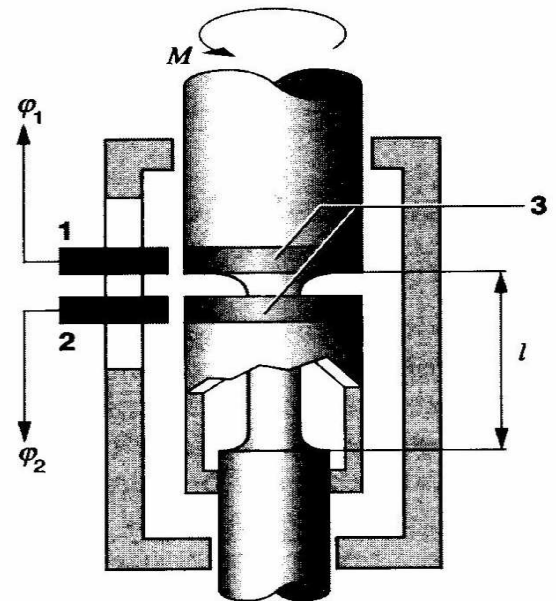
Fig. 2. Torquemeter; 1, 2 – angle (or angular velocity) transducers, 3 – twisted shaft section of the l length [4]

In both torquemeters presented in Fig. 1 and 2 the signal of measured value is transmitted contactless from rotating shaft to measuring system what improves measurement precision and limits distortions like sparkling characteristic for typical system of transmission.

Instead of torquemeter one can use measuring system containing sensors of angle or angular speed. Anyhow they work, the most important parameter is the number of impulses per one revolution because it decides about precision of measurement. In most cases there are inductive sensors which cooperate with toothed wheel (connected to the flying wheel for instance or separate). So called encoders, i.e. photoelectrical sensors of shaft angle producing great number of impulses per revolution are used.

Impulse from sensor is computer processed, which allows to obtain information on momentary value of displacement, angular velocity and acceleration of selected points of coupling shaft. Phase shift between impulses is a base for determination of instantaneous value of torque.

Application of high class torquemeter connected to the system of signal recording and analysis allows for precise analysis of torque momentary value. Besides of information on indicated and load torque, the registered signal contains also information on torques of other kind and on distortions as well (see Fig. 3). There could be different causes of these distortions and they could result from phenomena typical for combustion engine as well as distortions of electrical origin.

The latter can be easily eliminated or minimized using screening of measuring system. It is far more difficult to define sources and courses of distorting torques. One of the sources of significant distorting torque are the torsional vibrations produced by changing gas and inertia forces that cyclically act on the crankshaft. This additional torque adds to the torques substantial for the analysis and limits or even makes impossible their precise evaluation (it should be noted that the torque relative to the torsional vibrations can be used as valuable diagnostic parameters conveying information on engine technical condition). Because of that its elimination is necessary using, for instance, a filter in transmission and processing route.
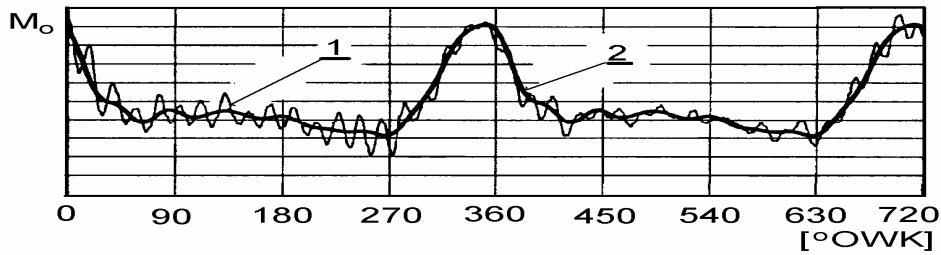
Fig. 3. Exemplary course of torque recorded for a single rotation of engine crankshaft;
1 – actual course of recorded torque, 2 – filtered course of torque signal [2]

However, application of filter requires previous definition of vibration parameters like frequency and amplitude. They can be estimated computationally but verification is possible only in a test stand.

4. Crankshaft torsional vibrations

Presented further results have been achieved in a course of analytical calculations of torsional vibrations using DELPHI computer program. A modern, five cylinder diesel of the R5 TDI type (AXD code) has been adopted as an exemplary test object (its basic technical data are shown in Table 1).

Tab. 1. Basic technical data of the R5 TDI engine [5]

Parameter	Data
Engine swept volume, cm^3	2460
Cylinder diameter, mm	81
Stroke, mm	95,5
Compression ratio	18,5
Cylinder number	5 (in line)
Cooling	water
Torque, Nm	320/2000
Power, kW	96/3500
Firing order	1 – 2 –4 –5 – 3

Taking into consideration the goal of investigation, i.e. determination of additional torque from torsional vibration, a computer program has been written which allows to find such quantities like vibration frequencies of different order, deflection amplitudes of equivalent masses (relative and actual) and courses of additional torques relative to cyclic forces resulting from engine operation. An earlier input of certain characteristic data was necessary such as those measurable (dimensions of piston-crankshaft assembly, for example) or those estimated (like damping coefficients of crankshaft sections).

Since the engine selected for exemplary calculations is a multi cylinder one its equivalent system has been assumed as the multi mass one. There are formulas obtainable in literature which can be used for calculation of system natural frequency of less than three masses. When larger number of masses is to be taken into consideration the universal formula of mechanics, known also as Lagrange formula should be applied (like in the presented program). This formula has following form:

$$\frac{d}{dt}\left(\frac{\partial E}{\partial \dot{q}_i}\right) - \frac{\partial}{\partial q_i}(E - V) + \frac{\partial D_t}{\partial \dot{q}_i} + \frac{\partial D_h}{\partial \dot{q}_i} = M_i(t),$$ (6)

where:

E – kinetic energy of vibrating system,
V – potential energy of the system,
q_i – general coordinate,
D_t – losses due to hysteresis,
D_h – losses due to friction.

Angular deflection β of individual equivalent mass has been assumed as general coordinate. Motion of each mass should satisfy the Lagrange formula.

At the initial stage of calculations the crankshaft (Fig. 4) and cooperating elements have been replaced with the equivalent torsional system.

Fig. 4 . A view of R5 TDI engine crankshaft

When defining the characteristic properties of equivalent shaft a principle has been assumed that shaft stiffness should be the same as that of real shaft and even along the whole structure. Masses of shaft parts (cranks, counterweights, damper) as well as masses of elements connected (piston, connecting rod) were replaced with rings of the same inertias mounted at relevant positions. The most important data of equivalent system have been summarized in Table 2.

Tab. 2. Classification of polar inertias and equivalent stiffness of the R5 TDI engine

Cyl. No	Symbol	Moment of inertia kg m^2	Equivalent length m	Remarks
1	Θ_1	0.010	0.150	Moment of inertia can vary when engine runs
2	Θ_2	0.010	0.150	
3	Θ_3	0.010	0.150	
4	Θ_4	0.010	0.150	
5	Θ_5	0.010	0.105	
6	Θ_6	0.36		Flywheel set

Input data used gave the following natural frequencies:
$\omega_1 = 2474$ rad/s,
$\omega_2 = 6850$ rad/s,
$\omega_3 = 10643$ rad/s,
$\omega_4 = 13449$ rad/s

of the first, second, third and fourth order, respectively. Relative angular deflections corresponding to these frequencies are summarized in Table 3 and presented in Fig. 5.

Tab. 3. Classification of relative amplitudes of equivalent system masses

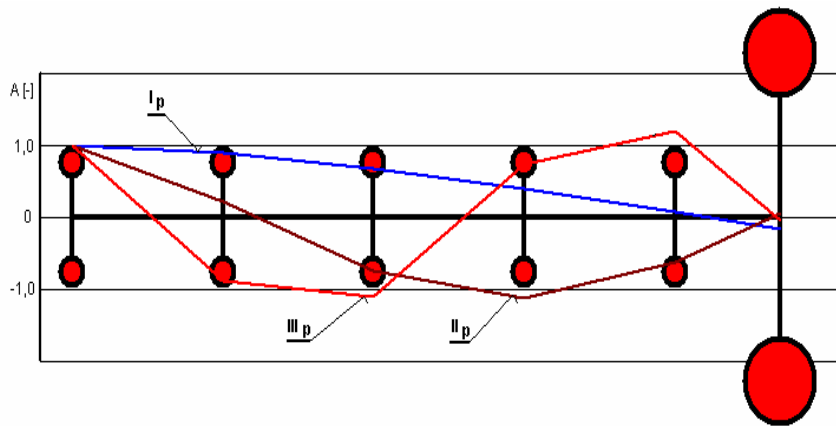| Mass No	1st order	2nd order	3rd order	4th order
1	1.000	1.000	1.000	1.000
2	0.899	0.226	-0.866	-1.981
3	0.707	-0.722	-1.115	0.943
4	0.444	-1.112	0.718	1.035
5	0.136	-0.642	1.211	-1.979
6	- 0.088	0.034	-0.026	0.026

Fig. 5. Relative angular deflection of equivalent system masses; I p, II p, III p – consecutive vibration order

Only the knowledge of equivalent system parameters facilitates achievement of the main goal, i.e. calculation of additional torques from torsional vibrations. The course of coupling torque M_s calculated for the section of crankshaft close to the flywheel has been presented in Fig. 6.
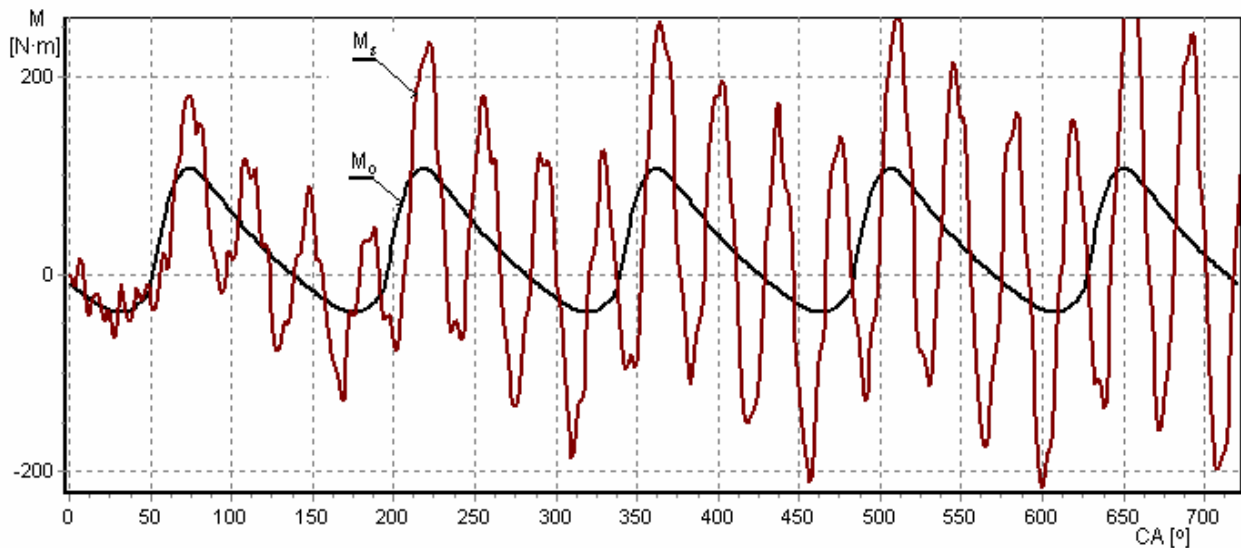

Fig. 6. Calculated courses of torque M_o and coupling torque M_s determined at the flywheel

Despite of cyclic change in value of forcing moment M_o the course of coupling torque M_s is of complex character, and its amplitude differs for each cylinder.The basic reason for such course of M_s is engine operation at speeds close to the resonanse frequencies of individual orders. Theoretic explanation of these phenomena is extremely difficult and will be the subject of eventual paper.

References

[1] Iskra, A., *Problems with determination of friction torque fluctuations at the cumbustion engin work cycle,* Journal of KONES Internal Combustion Engines 2004, vol.11, No. 1-2, pp.65-71, Warsaw 2004.

[2] Iskra, A., *Transformacja wyników pomiaru momentu sprzęgającego silnik z odbiornikiem mocy do momentu rzeczywistego*, Transformation of the Engine-to-receiver Coupling Torque Measurement Results to the Actual Value of Torque, Pomiary Automatyka Kontrola. 5/2005, pp. 41-45, 2006, Warszawa.

[3] Serdecki, W., *Badania współpracy elementów układu tłokowo-cylindrowego silnika spalinowego*, Wydawnictwo Politechniki Poznańskiej, Poznań 2002.

[4] Informator techniczny Bosch, *Czujniki w pojazdach samochodowych*, WKiŁ, Warszawa 2002.

[5] Materiały informacyjne firmy Volkswagen.