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ESTIMATION OF PARAMETERS FOR AN EXTREMELY LOW FUEL CONSUMPTION INTERNAL COMBUSTION ENGINE-BASED MEGAMETER-III VEHICLE

ISTVÁN PINTÉR[1]✉ AND MIHÁLY BAGÁNY[2]

[1] Department of Informatics, Kecskemét College, Izsáki str 10, Kecskemét, H-6000, HUNGARY
[2] Department of Natural Sciences and Engineering, Kecskemét College, Izsáki str 10, Kecskemét, H-6000, HUNGARY
✉E-mail: pinter.istvan@gamf.kefo.hu

Megameter-III is a special vehicle designed and constructed for international competitions by our team. A crucial part of the vehicle is the internal combustion engine, which ought to have extremely low fuel consumption. The experiences of previous competitions confirmed that it is necessary to develop a professional telemetry system, which allows tracking and logging online important parameters of the engine and the vehicle. In this paper, we discuss results for estimating of several parameters of the engine (angular velocity, angular acceleration, the Coulomb-Morin external friction torque and the damping constant). In our work, injector-pulses and inductive sensor's signals were monitored.

Keywords: internal combustion engine, extremely low fuel consumption, angular velocity, angular acceleration, external friction torque, damping constant

Introduction

Since 2010, the racing team at Kecskemét College participates in international competitions of fuel-efficient vehicles. The vehicles were named after 1 Megameter being 1000 km. The first result with Megameter-I in gasoline category was 1588 km L^{-1} (2010, Lausitz, Germany), and the best result of Megameter-II was 2661 km L^{-1} (2011, Nokia, Finland). Megameter-III achieved 2696 km L^{-1} (2012, Rotterdam, Nederland. The team-photo can be seen in *Fig.1*). The latest model, the Megameter-IV has exceeded the 3000 km/litre dream-limit (3082 km L^{-1}; 2013, Nokia, Finland) [1].

Figure 1: Student team photo with the Megameter-III vehicle (Rotterdam, 2010, second place)

During the development of the Megameter series, it became clear that in order to improve the technical parameters of the vehicle it is necessary to establish a telemetry system suitable for measuring, displaying and logging the engine's and vehicle's parameters during the race. This paper presents our recent results in this work, which have already been utilized in developing and construction of the Megameter-IV.

The measurements were taken using the Megameter-III engine, which has the following characteristics:

- one cylinder, four-stroke, air-cooled
- overhead controlled, overhead valve, intake manifold injection
- bore/stroke: 31.5/45 mm
- displacement volume: 45 cm^3
- compression ratio: 14
- power: 700 W
- torque: 2.2 N·m
- specific fuel consumption 235 g kWh^{-1}

There are different application areas and methods for estimation of engines' angular velocity. These estimation procedures are based mainly on crankshaft's rotation angle [2, 3, 4]. In the problem of model-based estimation of mechanical losses, the dependence of instantaneous angular velocity on angular position plays an important role [4]. A second-order spline-interpolation-based method was proposed in [2] for real-time measurement of angular velocity and angular acceleration. Our telemetry system is related to this latter work, and our results proved to be useful in further developing our special vehicle.

The physical model with its approximations and the parameter estimation

In order to achieve ever decreasing fuel consumption it is necessary to uncover the sources of energy losses. For this purpose a suitable physical model of internal combustion engine (ICE) is needed [5]. Moreover, with an appropriate physical model in hand, one can design the measurement for estimating the parameters of the physical model.

Physical model for estimating the external friction torque and damping constant

The instantaneous angular velocity of the crankshaft is changing even during only one crankshaft revolution. It is smaller on the compression stroke and greater in the midst of power stroke. The friction torque affecting the engine, averaged for two crankshaft revolutions depends on the instantaneous angular velocity. By expanding this function up to the second order we get *Eq.(1)*:

$$-M \approx M_0 + \frac{M_1}{\omega_1}\omega + \frac{M_2}{\omega_2^2}\omega^2 \tag{1}$$

where the indexed quantities are constant and positive, and they have physical meanings as follows: M_0 is the Coulomb-Morin external friction torque, which is independent of the relative speed of frictional surfaces (in this case the relative speed is the angular velocity); $M_1/\omega_1 \cdot \omega$ defines the damping constant (the second term corresponds to torque resulting from Newton's internal friction's force, it is proportional to the first power of angular velocity); $M_2/\omega_2^2 \cdot \omega^2$ stands for the drag torque characterizing the turbulent flows in pipes; it is in proportion to the square of angular velocity and it is resulting from the flow resistance in the intake and exhausting channels, in the valve-cage and in the air-pipe of the crankcase.

Our task is the estimation of the external torque from the measured data; therefore it is necessary to estimate the angular velocity. The basic equation of rotational motion is shown in *Eq.(2)*:

$$-M = -\Theta \frac{\mathrm{d}\omega}{\mathrm{d}t} \tag{2}$$

where after substitution of *Eq.(2)* into *Eq.(1)* we get

$$M_0 + \frac{M_1}{\omega_1}\omega + \frac{M_2}{\omega_2^2}\omega^2 = -\Theta \frac{\mathrm{d}\omega}{\mathrm{d}t} \tag{3}$$

Let's assume that the initial value of the angular velocity is ω_0. By separating the variables and integrating we get *Eq.(4)*:

$$\int_{\omega_0}^{\omega} \frac{\mathrm{d}\omega}{M_0 + \frac{M_1}{\omega_1}\omega + \frac{M_2}{\omega_2^2}\omega^2} = -\int_0^t \frac{\mathrm{d}t}{\Theta} \ . \tag{4}$$

The result of the integration is shown in *Eq.(5)*.

$$\begin{aligned} t = &-\frac{2\Theta}{\sqrt{4ac-b^2}}\operatorname{atan}\left(\frac{2c\omega+b}{\sqrt{4ac-b^2}}\right)+ \\ &+\frac{2\Theta}{\sqrt{4ac-b^2}}\operatorname{atan}\left(\frac{2c\omega_0+b}{\sqrt{4ac-b^2}}\right) \end{aligned} \ , \tag{5}$$

where three new variables were introduced for more readability as shown in *Eq.(6)*.

$$a = M_0\ ,\ b = \frac{M_1}{\omega_1}\ ,\ c = \frac{M_2}{\omega_2^2}^2 \ . \tag{6}$$

By solving *Eq.(5)* for ω one can get the final solution. However, instead of doing so, we analysed two approximate solutions of *Eq.(4)*.

As we will see below, these approximate solutions fit real measurements well. In order to estimate the engine's friction torque and damping constant long measurement duration (e.g. 20 s) is necessary. The relevant physical quantities can be estimated from the measured injector pulses.

Approximation 1: the friction torque is independent from angular velocity

By assuming that the other two terms are negligible compared to external friction M_0 the differential *Eq.(4)* simplifies to:

$$\int_{\omega_0}^{\omega} \frac{\mathrm{d}\omega}{M_0} \approx -\int_0^t \frac{\mathrm{d}t}{\Theta} \ , \tag{7}$$

and its solution is:

$$\omega = \omega_0 - \frac{M_0}{\Theta} t \ , \tag{8}$$

where ω_0 is the value of angular velocity at the beginning of deceleration.

The angular velocity of the engine's crankshaft has to be measured as the function of time for parameter estimation. Namely, after line-fitting to measured data the intercept and slope can be computed from relationship $\hat{\omega} = A - Bt$, so both the estimation of Coulomb-Morin external friction torque and the estimation of initial value of angular velocity can be determined: $\omega_0 = A$, $M_0 = B\Theta$.

Approximation 2: the friction torque is independent from the squared value of angular velocity

When the engine is running with wide open throttle, and the friction torque of pipe-flow losses is negligible comparing to external and internal losses, the differential equation to be solved is the following:

$$\int_{\omega_0}^{\omega} \frac{\mathrm{d}\omega}{M_0 + \frac{M_1}{\omega_1}\omega} \approx -\int_0^t \frac{\mathrm{d}t}{\Theta} . \quad (9)$$

The solution of *Eq.(9)* is:

$$\omega = \left(\omega_1 \frac{M_0}{M_1} + \omega_0\right) e^{-\frac{t}{\tau}} - \omega_1 \frac{M_0}{M_1} , \quad (10)$$

where $\tau = \omega_1\Theta/M_1$ is the time constant characterising the angular deceleration. By measuring the engine's crankshaft's angular velocity as the function of time, and after fitting a biased exponential-type curve we get the estimation $\hat{\omega} = Ae^{-Bt} - C$, from which the estimates of physical quantities are the following:

- the external friction torque is: $M_0 = \mathrm{B}\mathrm{C}\Theta$,
- the damping constant is: $M_1/\omega_1 = \mathrm{B}\Theta$,
- the initial value of angular velocity resulted from curve-fitting is: $\omega_0 = A\text{-}C$.

Estimation of the instantaneous torque and the running irregularity

The instantaneous angular acceleration as the derivative of the instantaneous angular velocity is needed for estimation of the instantaneous torque. Moreover, knowing the instantaneous angular velocity, another important parameter of the engine, the running irregularity can be determined. From the point of view of measurement, the estimation of instantaneous angular velocity requires the duration of two crankshaft revolutions (e.g. 100 ms). For measuring this, an inductive sensor was used.

When determining the running irregularity, the engine and the dynamometer were disconnected and the flywheel ring gear teeth together with an inductive sensor served as signal source. The induced voltage from the sensor can be approximated as:

$$\frac{U}{U_0} = \frac{(\omega + \beta t)}{\omega_0} \cos\left(z\omega t + z\frac{\beta}{2}t^2\right), \quad (11)$$

where U is the output signal of the sensor, U_0 is the amplitude of the sensor corresponding to the constant angular velocity of ω_0, ω is the angular velocity of the crankshaft, β is the angular acceleration of the latter, z is the number of teeth of ring gear.

In case of zero or small values of angular velocity the relationship *Eq.(11)* simplifies to

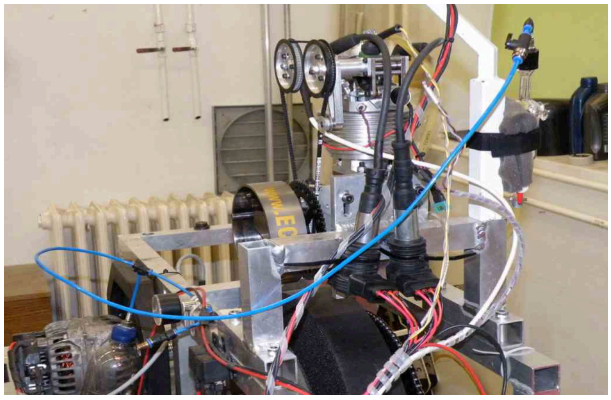

Figure 2: Arrangement for measuring the friction torque

$$\frac{U}{U_0} = \frac{\omega}{\omega_0} \cos(z\omega t) . \quad (12)$$

By assuming that during the pass-time of one tooth (tooth-time) the angular velocity is not changing significantly, the signal amplitude during the tooth-time can be well approximated by a constant. That is, the approximated sensor signal when passing the z^{th} tooth is given as $U/U_{0k} = \cos(z\,\omega_k\,t)$. The running irregularity is defined as a ratio in *Eq.(13)*:

$$\delta = \frac{\omega_{max} - \omega_{min}}{\omega_{average}} \quad (13)$$

Measurements, estimations and evaluations

The measurements were accomplished in the Student's Workshop of GAMF Faculty using an Agilent DSO-X 2002A digital storage oscilloscope. The oscilloscope is two-channel type with upper input cut-off frequency of 70 MHz, maximum sampling frequency of 2 GS/s and with sample storage of 50000 samples.

The algorithms used for evaluation of measured data were intentionally different. In one case the data series were not re-sampled and the differentiation was approximated by finite differences, and in the other case both re-sampling and smoothing derivative algorithm were applied. Moreover, the computation method of the trend of angular acceleration was also different.

Estimation of the external friction torque and damping constant

In this setup, an extra flywheel was driven by the engine fixed on the test stand. The measurement setup can be seen on *Fig.2*. The reduced resultant moment of inertia of synchronously rotating components with the crankshaft was $\Theta \approx 0.0534$ kg m^2.

The dynamometer's friction torque was approximately constant with a value of $M_{dynamometer} \approx$ 0.2 N m.

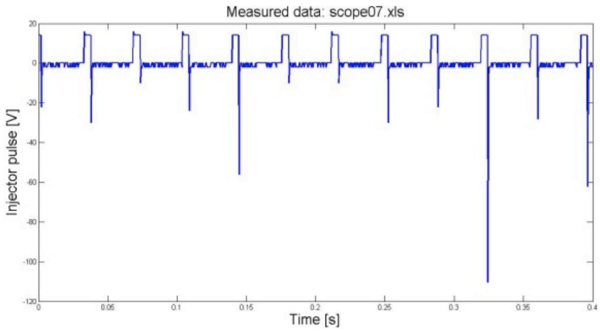

Figure 3: Measured injector pulses using a power MOSFET

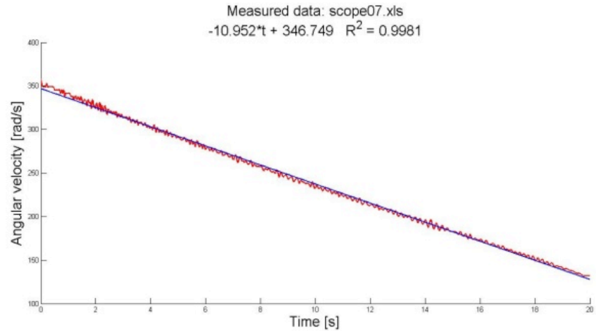

Figure 4: Linear model: the brake torque is independent of angular velocity

Table 1: Results of the fitting of data shown in *Fig.4*

Resampling	*A*, rad s^{-1}	*B*, rad s^{-2}	R^2
No	353.78	12.103	0.9954
Yes	346.75	10.952	0.9981

Table 2: Estimation of physical quantities

Resampling	n_0, rpm	M_0, N·m	M_{ICE}, N·m
No	3378	0.650	0.450
Yes	3311	0.585	0.385

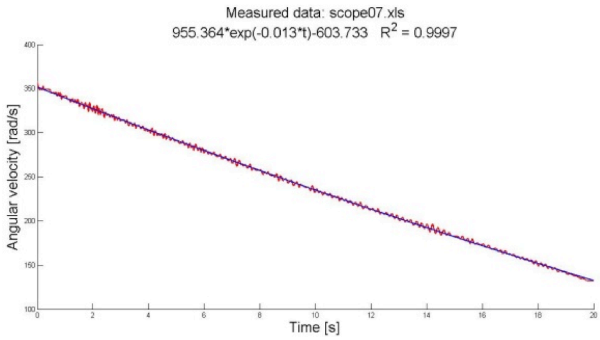

Figure 5: Biased exponential model: the braking torque is independent of the square of angular velocity

The engine's injector was triggered by a signal from the crankshaft, one injection occurred in every two crankshaft revolutions. The oscilloscope was connected to the injector's solenoid, and the solenoid's signal was sampled with a sampling frequency of 2.5 kHz, that is the sampling interval was T_s = 400 μs. The part of measured injector pulse-signal can be seen on *Fig.3*. The crankshaft's angular velocity ω = ω (t) as a function of time was determined from the stored samples, with duration of 20 seconds. At the beginning of the measurement, the engine was accelerated to 4000–4200 rpm, then the injector was disconnected from the intake manifold, and the engine was allowed to decelerate freely. By defining a suitable trigger level, the elapsed time *T* between two consecutive rising edges of the impulse can be determined. The instantaneous frequency is two times the reciprocal value of *T* ($f_{instantaneous} = 2 / T$), because every two revolution results in one impulse. The instantaneous angular velocity can be computed from this instantaneous frequency value. As the resulting angular velocity value sequence is non-equidistant, it has been re-sampled using cubic spline interpolation. First, a linear model was fitted *(Fig.4)*. The results are summarised in *Table 1*. The two methods gave similar parameters, but the resampling resulted in somewhat better goodness-of-fit. The estimation of the physical quantities from parameters shown in *Table 1* can be seen in *Table 2*. The estimations resulted by the two methods are similar in case of model of *Eq.(7)*.

In case of model of *Eq.(9)*, the fitting has been fulfilled on the resampled data series. The results are demonstrated on *Fig.5*.

From the above results the external friction torque is $M_0 = B{\cdot}C{\cdot}\Theta \approx 0.419$ N·m; the average braking torque of the engine is $M_{engine} = M_0 - M_{dynamometer} \approx 0.219$ N·m; the damping constant is $M_1/\omega_1 = B\Theta = 6.94{\cdot}10^{-4}$ N·m·s. The initial angular velocity and revolution are $\omega_0 = A - C \approx$ 352 rad s^{-1}, n_0 = 3358 rpm.

We can conclude that the two types of models gave similar results, but the biased exponential model fitted somewhat better the measured data than the linear model, thus the former are preferred. Furthermore, the presented data indicate that both the average breaking torque of the engine and the damping constant are acceptable small, therefore the engine is acceptable from this point of view.

Estimation of the instantaneous torque and the running irregularity from measurements

When the running irregularity was measured, the engine and the dynamometer were disconnected. The moment of inertia of the parts running synchronously with the crankshaft was Θ = 4000 kg·mm^2. A ring gear fixed to the crankshaft was used as signal source (number of teeth is z = 66) together with an inductive sensor fitted from the teeth in a distance of 4 mm. An oscilloscope was connected to the sensor and the voltage signal was sampled with 500 kHz sampling frequency *(Fig.6)*. It is necessary to use such a great sampling rate because of the following. In order to acceptably estimate the instantaneous angular velocity, it is necessary to measure *N* samples during the tooth-time. In case of nominal revolution of n_0, the sampling frequency is f_S =

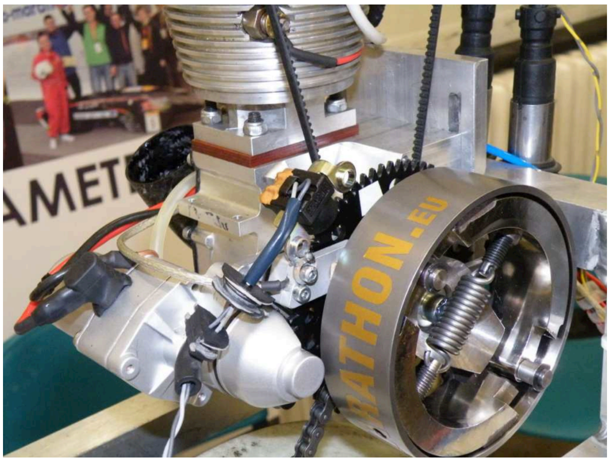

Figure 6: Setup for measuring the instantaneous torque

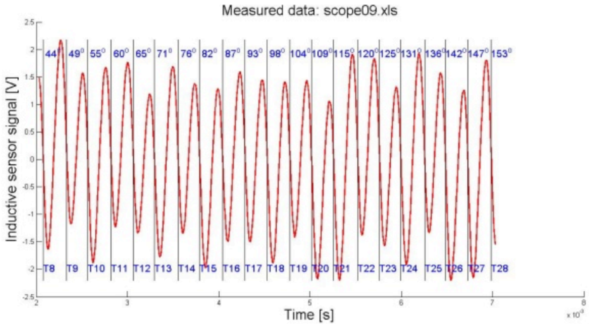

Figure 8: Inductive sensor's signal labelled with the crankshaft's angle and tooth number (angles are in degrees)

$n_0 \cdot N \cdot z$ / 60, that is in case of n_0 = 4000 rpm and N = 100 samples, f_S = 440000 Hz. The nearest possible setting of the DSO is 500 kHz. The capacity of the sample storage is 50000 samples, so the duration of the measurable signal is 0.1 s. As it is greater than the duration T_2 of two crankshaft revolution, the measurement of at least one full engine cycle is feasible using this DSO ($T_2 = 2 \cdot (60/n_0) = 0.03$ s). The instantaneous angular velocity is the reciprocal of the time between two falling edge of the sensor's signal, so the raw signal had to be smoothed before the zero-crossing detection.

Two different smoothing methods were implemented in this case as well: a 10-point moving average smoothing and SAVITZKY-GOLAY smoothing with tenth order polynomial and 31-point length window (for the latter see *Fig.7*). Because the number of teeth is known, the exact value of the crankshaft angle also with the tooth number can be tracked (see *Fig.8*). As the angular velocity values are non-equidistant, cubic spline interpolation-based re-sampling was applied as in the previous section.

Due to the fast sampling rate, we can track very short-time changes in angular velocity values during 100 ms. However, our task is to estimate the crankshaft's instantaneous angular velocity. For this purpose the trend of the angular velocity curve estimated at teeth is necessary. The trend was determined on the one hand by using a 10-point moving average filter, on the other with FFT-based filtering (for

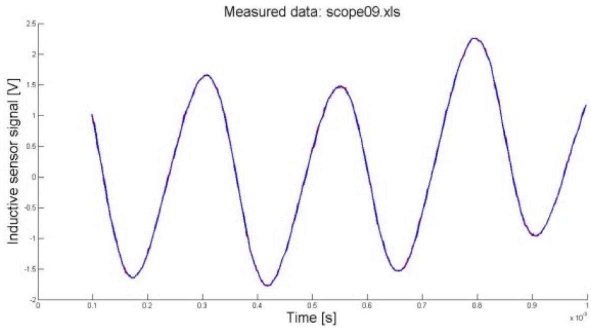

Figure 7: The raw and smoothed signal of inductive sensor in case of SAVITZKY-GOLAY smoothing

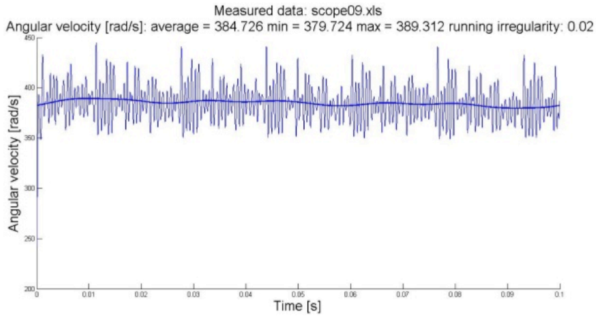

Figure 9: Instantaneous angular velocity values and the FFT-based trend

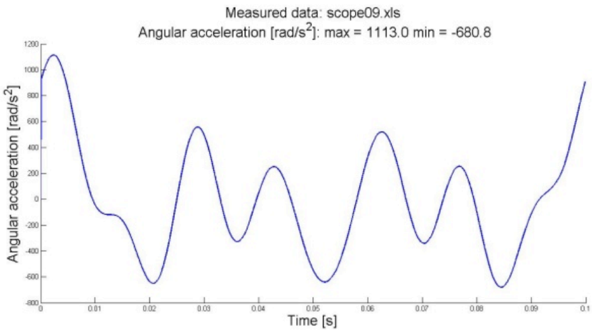

Figure 10: Estimation of instantaneous acceleration as smoothing derivative of the trend of instantaneous angular velocity

the latter see *Fig.9*). In the first case the time of averaging was around 2 ms. Using this value for the time-constant τ of a low-pass filter for the FFT-based computation, for the cut-off frequency of the low-pass filter we get $f_H = 1 / (2 \cdot \pi \cdot t) \approx 80$ Hz. Following its definition, the running irregularity was computed using the maximum, minimum and average values of the angular velocity trend shown in *Eq.(14)*.

$$\delta = \frac{\omega_{max} - \omega_{min}}{\omega_{average}} = \frac{389.3 - 379.7}{384.7} \approx 0.02 \,. \qquad (14)$$

The estimation of instantaneous acceleration was computed with differences in case of moving average-based trend. In case of FFT-based trend, the SAVITZKY-GOLAY smoothing derivative algorithm was applied (tenth order polynomial, 31-point length window).

Fig.10 illustrates the instantaneous acceleration as a function of time. In this measurement the maximum value of angular acceleration was 1113 rad s^{-2}, so for the corresponding instantaneous torque we get:

$$M_{\max} = \Theta \beta_{\max} \approx 0.004\ \mathrm{kg} \cdot \mathrm{m}^2 \cdot 1113\ \frac{\mathrm{rad}}{\mathrm{s}^2} \approx 4.5\,\mathrm{N} \cdot \mathrm{m}. \quad (15)$$

Nearby the power stroke instant higher value of angular acceleration can be estimated. In that measurement the corresponding instantaneous torque was of 16 N·m (for details see Ref. [6]). The crankshaft's torque gets to wheel by a drive train with a reduction ratio of 7, so the maximal torque at the wheel is 112 N·m. The wheel's maximal slipless torque transmission is 40–50 N·m that is during the maximal instantaneous acceleration the wheel will slip on the road, which results in energy loss. This effect ought to be concerned in designing the drive chain of Megameter-IV, e.g. torque-damping torsion clutch should be built in.

Conclusions

Two methods were elaborated for instantaneous speed and acceleration of an internal combustion engine via on-line measurement. Both the injector-pulse-based and the inductive sensor-based methods gave useful data for developing the next engine, namely the Megameter-IV. The results presented in this paper are the numerical estimation of the engine's internal friction, the damping constant, the instantaneous angular velocity, the instantaneous angular acceleration and the estimation of the instantaneous torque of the engine during the power stroke. These results provided the foundation for the design and construction of the on-board electronic control unit, which can be seen on *Fig.11*.

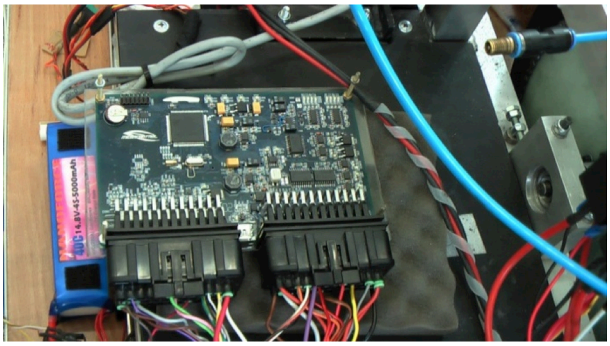

Figure 11: ECU is under development for motor control and telemetry

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