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Adaptive Estimation of the Engine Friction Torque

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Abstract— **Errors in an estimate of friction torque in modern spark ignition automotive engines have a direct impact on driveability performance of a vehicle and necessitate a development of real-time algorithms for adaptation of the friction torque. The friction torque in the engine control unit is presented as a look-up table with two input variables (engine speed and indicated engine torque). Algorithms proposed in this paper estimate the friction torque during engine start and idle. Newton's second law for rotational dynamics is used as a reference model during engine start. The friction torque is estimated via a deviation from the reference model. The values of the friction torque at the nodes of the look-up table are updated, if new measured data of the friction torque is available. New recursive and computationally efficient algorithms are developed for adaptation of the nodes of the look-up tables. The algorithms are tested on a Volvo vehicle equipped with a six cylinder prototype engine.**

Keywords: Adaptive Estimation, Model Reference Adaptive System, Spark Ignition Engines, Engine Friction.

I. INTRODUCTION

The performance of the engine control system depends on the accuracy of the engine torque model. One of the important parts of the torque model is the friction losses. The values of the engine friction torque, which is precalibrated, are memorized in a look-up table (static map) residing in the memory of an engine controller. Friction torque is mainly a function of engine speed, load (engine indicated torque) and engine oil temperature. Variability in engine components may result in variations in the engine friction torque. Friction torque losses, moreover, change with time due to aging of the engine components. These variations cause errors in the estimate of the friction torque, and thus lead to deterioration of driveability performance. Despite the fact that engine friction modeling is a well studied field (see for example [3] and references therein), very little attention has been paid to real-time friction estimation. The facts presented above necessitate the development of the real-time adaptation algorithms to improve the accuracy of the engine friction model. The most promising opportunity for estimating friction is during engine idle when the engine is decoupled from the driveline [1]. The idle state however, could give an estimate of the friction torque only at idle speed and low indicated torque. All the nodes of the look-up table could be adapted by using new value of the friction torque at idle. However, even small errors in the friction estimation at idle due to the errors in accessory loads, for example, could lead to significant errors in the friction estimation at high rotational speeds. Moreover, the friction losses due to the aging of the components could also change as a function of the engine speed (not only the offset, but also the gradient of the map should be adapted). Therefore, more points for different engine speeds and loads are required for successful adaptation of a friction look-up table. An opportunity for obtaining an accurate estimate of a friction torque is the period following engine start. At engine start, the engine speed increases to a relatively high (compared with the idle speed) level, and then slowly decreases, converging to the desired idle speed. This period when the engine speed decreases gives an opportunity to estimate engine friction torque. Newton's law for rotational dynamics can be used as a reference model. The difference between the derivative of the engine speed multiplied by the inertia moment and the engine brake torque can be seen as a deviation from the reference model. If the friction losses are correctly estimated, the deviation from the reference model is close to zero at the interval of interest. This reference model should be valid during long term engine operation. Any deviation from the reference model at the interval of interest is assumed to be related to the friction losses, since the aging of the engine components first of all affects the friction losses. If a deviation from the reference model is detected, then the friction look-up table is updated so that the deviation is minimized. The closed loop system can be seen as a model reference adaptive system driven by the engine start events. The algorithm proposed in the present paper can be divided in two parts. The first part is the estimation of the friction losses at start and idle and the second part is the adaptation of the friction look-up table. Adaptation of look-up tables (static maps) is widely used in the engine control functionality for robustness improvement [5]. However, adaptation algorithms described in [5] do not allow a prediction of the values of the operating parameter in the regions with meager new data representation.

In this paper the look-up table defines a manifold for engine friction torque in three dimensional space with engine speed and indicated torque as independent variables, whereby the shape of the manifold reflects physical dependencies of the friction torque as a function of speed and indicated torque. If new data is available only in a certain operating region, then the part of the manifold parameters are adapted (for example the offset and the gradient in the engine speed direction). Adaptation of the look-up table is associated with a motion of the manifold in three dimensional space. The position and the orientation of the manifold in three dimensional space change only after adaptation, which in turn, allows for a prediction of the friction torque for a wide range of

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speeds and indicated torques even with few new measured points by taking into account physical dependencies, which are present in the shape of the manifold. An adaptation algorithm is constructed so that only the nodes (sites) of the look-up table are adapted, thereby the values of engine friction torque between the nodes are computed using linear interpolation. In order to reduce the computational burden of the processor, recursive and computationally efficient algorithms for adaptation of the nodes in the look-up table are developed.

A Volvo vehicle equipped with six cylinder prototype engine was used in the experiments.

The contributions of this paper are the following: 1) new recursive algorithms for adaptation of the nodes of the look-up tables, 2) new algorithms for estimation of the engine friction torque at start.

II. IMPACT ON DRIVEABILITY

Errors in the estimate of engine friction torque have a direct impact on the behaviour of the engine speed during negative transients (where the driver releases the accelerator pedal and switches to a neutral gear). The engine speed during negative transients is governed by the torque model. Requested indicated engine torque is calculated from the requested engine brake torque by adding the torque losses (friction and pump losses). The requested engine brake torque is calculated as a function of accelerator pedal position and engine speed. The requested indicated engine torque in the negative transient of the engine speed with overestimated friction losses (real losses are less than estimated), is higher than it would be if friction losses were to be correctly estimated. The desired engine load is calculated from the desired indicated torque. The feedback load control system regulates the engine load to the desired load, which implies that the actual indicated torque converges to the desired indicated torque. The actual indicated engine torque (which is negative during a negative transient) is higher than it would be if the losses were estimated correctly. Therefore, the engine speed decays slowly. Moreover, overestimation of the friction torque leads not only to slow negative transients of the engine speed, but also to a constant offset in the steady-state engine speed with respect to a target idle speed. Errors in the estimation of the friction losses thus can lead directly to deterioration of driveability performance.

III. PROBLEM STATEMENT

The errors in the estimated friction losses have an effect on the behaviour of the engine torque at start and idle. Newton's law

$$J\dot{\omega} = T_{brake} - T_{acs} \tag{1}$$

can be seen as a reference model at the interval $[t_i \quad t_f]$, where t_i is the time when the engine speed nears a maximum value at start, t_f is the time when the engine speed reaches the desired idle speed, ω is the engine speed, J is the inertia moment of the engine, T_{brake} is the engine brake torque, T_{acs} is the torque corresponding to accessory loads. The engine brake torque is the difference between the engine indicated torque and the torque corresponding to the losses, i.e., $T_{brake} = T_{ind} - T_{loss}$, where T_{ind} is the indicated engine torque, $T_{loss} = T_f + T_p$, T_{loss} is the torque corresponding to the losses, which in turn is the sum of the friction T_f and the pump losses T_p.

Let us introduce the following error

$$e(t) = J\dot{\omega} - (T_{brake} - T_{acs}) \tag{2}$$

If the torque model is well calibrated, then the error $|e(t)|$ is close to zero at the interval of interest. Any deviation from the reference model is assumed to be related to the friction losses, since aging of the engine components first of all affects the friction losses. The friction torque is a function of engine speed and indicated engine torque, $T_f = f(\omega, T_{ind})$. The friction torque is presented as a look-up table with two inputs ω and T_{ind}. The nodes of the look-up table should be updated so that the absolute values of the error $e(t)$ are reduced after each start event. Warm starts only are considered in the present paper. The control aim can be formulated as follows. It is necessary to find an adaptation mechanism for adaptation of the nodes of the engine friction look-up table such that the following control aim is reached:

$$\overline{\lim_{k\to\infty}} |e(t)| \leq \Delta \tag{3}$$

where k is the number of the start events, $\Delta > 0$ is a small positive constant, $t \in [t_i \quad t_f]$. The system as described can be seen as a model reference adaptive system driven by the engine start events.

IV. ESTIMATION OF THE FRICTION TORQUE DURING THE ENGINE AFTER START PERIOD

The problem stated above can be solved in two steps. In the first step, the deviation from the engine friction torque, which is pre-calibrated in the rig, is calculated for each start event by a comparison of $J\dot{\omega}$ and $T_{brake} - T_{acs}$ at a certain interval. If $J\dot{\omega}$ significantly deviates from $T_{brake} - T_{acs}$, then the number of the actual values of the engine friction torque is computed. The number of the actual values of the engine friction torque as a function of speed and indicated torque is the input to the second step. In the second step, the nodes of the friction torque look-up table are adapted so that the deviation between $J\dot{\omega}$ and $T_{brake} - T_{acs}$ is reduced for the next start event.

Assume that the engine friction torque can be presented as a sum of two components $T_{fc} + \Delta T_f$, where T_{fc} is the engine torque pre-calibrated in the rig, and ΔT_f is the deviation from the pre-calibrated torque. The deviation ΔT_f is calculated by using an error $e(t)$ which is evaluated at certain discrete points t_p, $(p = 1, 2, ...)$, on a time scale, i.e.,

$$\begin{aligned} &\Delta T_f(w(t_p), T(t_p)_{ind}) = e(t_p) - J\dot{\omega}(t_p) + T(t_p)_{ind} \\ &-T_{fc}(t_p) - T_p(t_p) - T_{acs}(t_p) \end{aligned} \tag{4}$$

where $t_p \in [t_i \quad t_f]$. The points t_p should be well separated from each other on the time scale, providing information about ΔT_f for different values of engine speed and indicated torque. From two to four measured points can be obtained during a negative transient. One point is obtained at idle. The deviation from the calibrated engine friction torque at idle $\Delta T_f(w_{id}, T_{ind_{id}})$, where w_{id} is the engine idle speed, $T_{ind_{id}}$ is the indicated torque at idle, is calculated as follows:

$$\Delta T_f(w_{id}, T_{ind_{id}}) = T_{ind_{id}} - T_{fc_{id}} - T_{p_{id}} - T_{acs_{id}} \quad (5)$$

where $T_{fc_{id}}$, $T_{p_{id}}$, $T_{acs_{id}}$ are the values of the friction torque, pump torque and the torque corresponding to the accessory loads, respectively. If the engine is idling for a relatively long period, the deviation ΔT_f is averaged over certain number of steps, providing a consistent estimate for the deviation $\Delta T_f(w_{id}, T_{ind_{id}})$.

For calculation of $\Delta T_f(w(t_p), T(t_p)_{ind})$ according to (4) during a start, the estimate of the derivative of the engine speed is required. The backward difference method, which is widely used for calculation of the derivative of the signal, often gives noisy estimates. For the improvement of the quality of the estimate of the derivative of the engine speed signal, a spline interpolation method is used [2], [4]. A spline interpolation method is based on on-line least-squares polynomial fitting over a moving-in-time window of a certain size. The advantage of this method over the backward difference method is its good transient behavior. The idea for the spline interpolation method is to fit a polynomial of a certain order as a function of time to the measured signal in the least-squares sense, and to take the derivatives analytically. Since the nodes of the friction look-up table are adapted after the start events, a post-processing of the signals is allowed, i.e., the signals are memorized and processed 'offline'. The spline interpolation method gives an accurate estimate of the derivative of the engine speed in the post-processing, since the derivative of the engine speed is computed in the middle of a moving window. This technique improves essentially the quality of the engine speed derivative signal. Other signals in (4) should also be delayed. Fig. 1 shows the behaviour of the engine speed together with its derivative and engine brake torque during a start. The derivative of the engine speed is computed by using the spline interpolation method with a window size of 250 steps (each step is $4ms$). The derivative was computed in the middle of a moving window. The friction losses are correctly estimated, and the difference $e(t) = J\dot{\omega} - T_{brake}$, which is plotted with dotted line is close to zero in the interval where engine speed decreases. Since the second step of the algorithm has a discrete input, the values of $e(t)$ are evaluated at two points indicated with plus signs. Figures 2 and 3 show the behaviour of the engine speed and brake torque during a start where the friction losses are overestimated, i.e., $\Delta T_f = 10[Nm]$. Fig. 3 shows the difference between $J\dot{\omega}$ (dashed line) and engine brake torque (dashdot line). The difference is plotted with dotted line. The points where ΔT_f is calculated are shown with plus signs. The deviations from the pre-calibrated friction torque ΔT_f as a function of engine speed and indicated torque are the inputs for adaptation algorithms, to be described in the next Section. As can be seen from Fig. 3, the deviations ΔT_f are estimated with some errors. For each deviation ΔT_f, a weighting factor, which indicates the consistency of the point, is assigned. As can be seen from the Figures 2 and 3, two points are available for adaptation of the friction losses. The third point for calculation ΔT_f is available when engine is idling. The deviation ΔT_f at idle is averaged over a certain number of steps, providing a consistent estimate. Therefore, the weighting factor for the deviation ΔT_f at idle is chosen higher, since the idle conditions provide more consistent estimate of ΔT_f than engine start conditions. Adaptation algorithms for look-up tables are presented in the next Section.

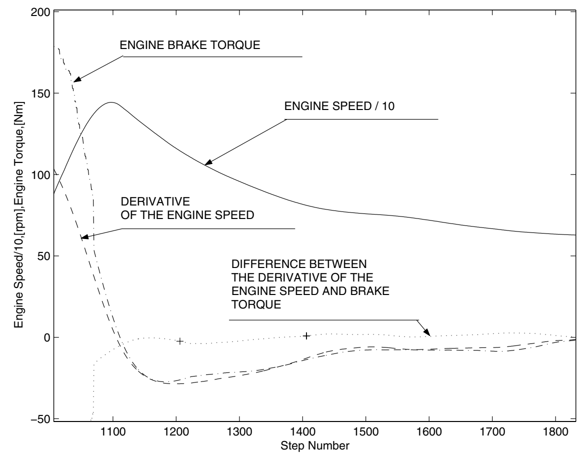

Figure 1

Measurements with the step of 4 ms on the prototype engine. The friction losses are correctly estimated. Engine speed during a start is plotted with solid line. The values of the engine speed are divided by ten. Engine brake torque is plotted with dashdot line. The derivative of the engine speed multiplied by the inertia moment $J\dot{\omega}$ is plotted with dashed line. The difference $e(t) = J\dot{\omega} - T_{brake}$ is plotted with dotted line. The points where $e(t_p)$ is evaluated are indicated with plus signs.

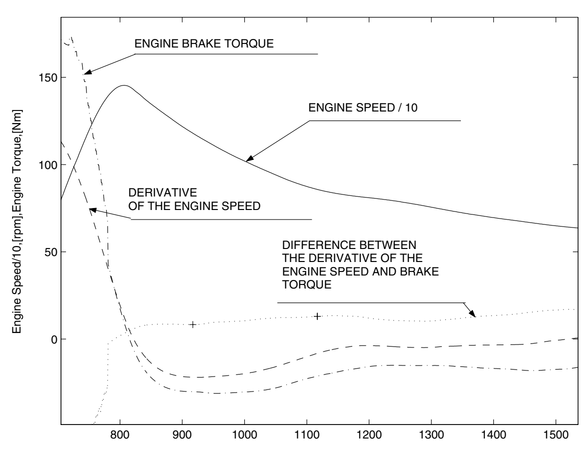

Figure 2

Measurements with the step of 4 ms on the prototype engine. The friction losses are overestimated by $10[Nm]$. Engine speed during a start is plotted with solid line. The values of the engine speed are divided by ten. Engine brake torque is plotted with dashdot line. The derivative of the engine speed multiplied by the inertia moment $J\dot{\omega}$ is plotted with dashed line. The difference $e(t) = J\dot{\omega} - T_{brake}$ is plotted with dotted line. The points where $e(t_p)$ is evaluated are indicated with plus signs.

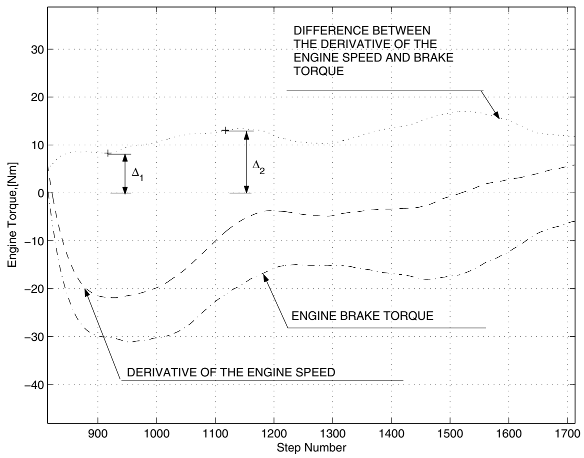

Figure 3
Measurements with the step of 4 ms on the prototype engine (the same as Fig. 2). The friction losses are overestimated by $10[Nm]$. Engine brake torque is plotted with dashdot line. The derivative of the engine speed multiplied by the inertia moment $J\dot{\omega}$ is plotted with dashed line. The difference $e(t) = J\dot{\omega} - T_{brake}$ is plotted with dotted line. The points where $e(t_p)$ is evaluated are indicated with plus signs. The left point is evaluated at $\omega = 1180[rpm]$,$T_{ind} = 23[Nm]$, and the right point is evaluated at $\omega = 860[rpm]$,$T_{ind} = 43[Nm]$. The friction torque at idle is evaluated at $\omega = 650[rpm]$, $T_{ind} = 34[Nm]$.

V. ADAPTATION OF THE FRICTION LOOK-UP TABLE

1) Adaptive Problem Statement : In Figure 4, engine friction torque is plotted as a function of engine speed and indicated torque. The friction torque is overestimated by $10[Nm]$. Two points representing the estimated friction torque from the start are plotted with plus signs. The point that represents the estimated friction torque at idle is plotted with round and plus signs. The problem statement is the following. It is necessary to design adaptation algorithms for the nodes of the look-up table by using three measured points of the actual friction torque. As indicated above, the estimation of the engine friction torque at engine start provides less consistent estimates than estimates of the friction torque at engine idle. Therefore, the measurements of the friction torque at idle and at start should be treated differently by assigning different weighting factors in the adaptation algorithms.

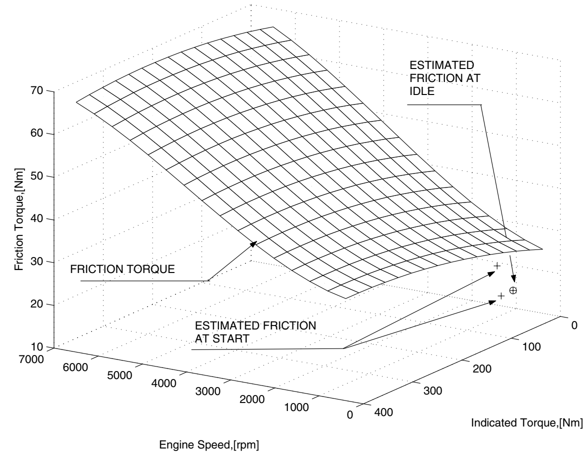

Figure 4
Engine friction torque is plotted as a function of engine speed and indicated engine torque. The friction torque is overestimated by $10[Nm]$. Two points which represent the estimated friction torque at start (see Figures 2 and 3) are plotted with plus signs. The point which represents the estimated friction torque at idle is plotted with round and plus signs.

2) Adaptation Algorithms for Look-Up Tables : The algorithm of the adaptation of the nodes of two dimensional tables can be divided into three steps. In the first step, the output of the look-up table is approximated by a polynomial of two independent variables in the least-squares sense. In the second step, a recursive procedure is designed for adaptation of the part of the coefficients of the polynomial when new data is added. In the third step of the algorithm, the approximation error is canceled. Namely, the differences between the polynomial approximation of original table and the polynomial approximation after adaptation are evaluated at every node and added to the nodes of original look-up table. This allows a cancellation of the approximation error and usage of low order polynomials, which are more robust with respect to the measurement errors. Only the nodes of the look-up table are adapted as a result of the application of the algorithm described above. The values of the friction torque between the nodes are obtained by linear interpolation.

Suppose that there is a look-up table describing the variable z as a function of two variables x and y. The look-up table is presented as a number of nodes (x_h, y_p) , $h = 1, ...D$, $p = 1, ..., G$, where the output variable $z_{h,p}$ is defined. The values of the variable z between the nodes are computed via a linear interpolation. The problem of the adaptation of a look-up table is reduced to the adaptation of $z_{h,p}$.

As it was mentioned above, the problem can be solved in three steps as follows.

Step 1. Polynomial Approximation.

In this step, the look-up table is approximated by the following polynomial:

$$\hat{z} = \sum_{i=0}^{n} \sum_{j=0}^{n} a_{i,j} x^i y^j \tag{6}$$

where n is the order of the polynomial, $a_{i,j}$ are the coefficients of the polynomial. The polynomial model (6) can be written in the following vector form:

$$\hat{z} = \varphi^T \theta \tag{7}$$

where

$$\begin{aligned} \varphi &= [1, y, y^2, ..., y^n, x, xy, xy^2, ..., xy^n, ..., \\ &\quad x^n, x^n y, x^n y^2, ..., x^n y^n]^T \\ \theta &= [a_{00}, a_{01}, a_{02}, ..., a_{0n}, a_{10}, a_{11}, a_{12}, ..., a_{1n}, ..., \\ &\quad a_{n0}, a_{n1}, a_{n2}, ..., a_{nn}]^T \end{aligned} \tag{8}$$

The performance index to be minimized is the following:

$$S = \sum_{l=1}^{N} (z_l - \hat{z}_l)^2 w_l \tag{9}$$

where N is the number of the nodes (sites) of the look-up table, $l = 1, ..., N$,$N = D \times G$, w_l is the weighting factor at every node of the table. The parameter θ, which minimizes index (9) is computed as follows:

$$\theta = [\sum_{l=1}^{N} (\varphi_l \varphi_l^T w_l)]^{-1} \sum_{l=1}^{N} z_l \varphi_l w_l \tag{10}$$

Assuming, the parameter vector θ has been computed according to the formula (10) and memorized in the memory of the electronic control unit, the problem of the adaptation of the look-up table can be stated as the problem of the adaptation of the parameter vector θ for new measured data. The values $\hat{z}_{(h,p)}$ at the nodes (x_h, y_p) are computed according to (7).

Step 2. Adaptation of the coefficients. In this step of the algorithm, the vector θ is adapted for new data. Assuming, new measured data x_m, y_m, z_m with the weighting factor w_m is added to the data set, the parameter vector $\theta \in R^{(n+1)^2}$ is divided into two parts: the first part $\theta_c \in R^{(n+1)^2 - q}$ remains unchanged from the previous step, and the second part $\theta_a \in R^q$ which should be adapted, where q is the number of the parameters to be adapted. Then,

$$\theta = [\theta_c \quad \theta_a]^T \tag{11}$$

$$\varphi = [\varphi_c \quad \varphi_a]^T \tag{12}$$

where φ_c is the part of the regressor, which corresponds to the parameter vector θ_c, and φ_a is the part of the regressor corresponding to the parameter vector, θ_a. New measured data x_m,y_m,z_m is added to the data set. The performance index to be minimized is the following:

$$S_1 = \sum_{l=1}^{N} (z_l - \hat{z}_l)^2 w_l + (z_m - \varphi_m^T \theta)^2 w_m \tag{13}$$

where

$$\begin{aligned} \varphi_m &= [1, y_m, y_m^2, ..., y_m^n, x_m, x_m y_m, x_m y_m^2, ..., \\ &\quad x_m y_m^n, ..., x_m^n, x_m^n y_m, x_m^n y_m^2, ..., x_m^n y_m^n]^T \end{aligned} \tag{14}$$

$$\varphi_m = [\varphi_{cm} \quad \varphi_{am}]^T \tag{15}$$

The adaptive parameter θ_a is computed according to the following equation $\dfrac{\partial S_1}{\partial \theta_a} = 0$, i.e.,

$$\theta_a = [\sum_{l=1}^{N} (\varphi_{al} \varphi_{al}^T) w_l + \varphi_{am} \varphi_{am}^T w_m]^{-1}$$

$$\sum_{l=1}^{N} (z_l - \varphi_{cl}^T \theta_c) \varphi_{al}^T w_l + (z_m - \varphi_{cm}^T \theta_c \varphi_{am}^T) w_m \tag{16}$$

In order to reduce the computational burden on the engine controller, the adjustable parameter is computed recursively. The vector of the adjustable parameters is computed according to the following formula at step $(k-1)$

$$\theta_{a(k-1)} = [\sum_{l=1}^{N} (\varphi_{al} \varphi_{al}^T w_l)]^{-1} \sum_{l=1}^{N} (z_l - \varphi_{cl}^T \theta_c) \varphi_{al}^T w_l \tag{17}$$

and the adjustable parameter θ_{ak} at step k should be updated recursively via $\theta_{a(k-1)}$ as soon as new data z_m, φ_m with the weighting factor w_m are available. Applying the matrix inversion relation to (16) and taking into account (17) one gets the following adjustment law for the parameter θ_{ak} at step k:

$$\begin{aligned} \theta_{ak} &= [I - \frac{\Gamma_{k-1} w_m \varphi_{am} \varphi_{am}^T}{(1 + w_m \varphi_{am}^T \Gamma_{k-1} \varphi_{am})}](\theta_{a(k-1)} + \\ &\quad \Gamma_{k-1}(z_m - \varphi_{cm}^T \theta_c) w_m \varphi_{am}^T), \end{aligned} \tag{18}$$

$$\Gamma_k = \Gamma_{k-1} - \frac{w_m \Gamma_{k-1} \varphi_{am} \varphi_{am}^T \Gamma_{k-1}}{(1 + w_m \varphi_{am}^T \Gamma_{k-1} \varphi_{am})}, \tag{19}$$

where $\Gamma_{k-1} = [\sum_{l=1}^{N} (\varphi_{al} \varphi_{al}^T w_l)]^{-1}$, I is $q \times q$ identity matrix, and the following condition for convergence of the algorithm imposes restrictions on the weighting factors:

$$-\Gamma_{k-1} < \Gamma_{k-1} - \frac{w_m \Gamma_{k-1} \varphi_{am} \varphi_{am}^T \Gamma_{k-1}}{(1 + w_m \varphi_{am}^T \Gamma_{k-1} \varphi_{am})} < \Gamma_{k-1}. \tag{20}$$

Algorithm (18), (19) is easily implementable since the dimension of the vector θ_a is low. As a rule, only the offset and the slope in one of the directions are updated; i.e., $q = 2$. The values $\hat{z}_{a(h,p)}$ at the nodes (x_h, y_p) are computed according to the following formula $\hat{z}_{ak} = \varphi_{ck}^T \theta_c + \varphi_{ak}^T \theta_{ak}$. The vector θ_c is not updated. That, in turn, allows the shape of the manifold to be maintained.

Step 3. Cancellation of the approximation error. Low order polynomials (6) is advisable to use for approximation of look-up tables. Low order polynomials are robust with respect to the measurement noise compared with the polynomials of a high order. However, approximation of a look-up table using low order polynomials could also give a relatively large approximation error. In order to cancel the approximation error, the following differences $\hat{z}_{a(h,p)} - \hat{z}_{(h,p)}$ between the polynomial approximation of the adapted table and the polynomial approximation of the original table are computed at every node $h = 1, ...D$, $p = 1, ..., G$ and added to the values $z_{(h,p)}$ of the original look-up table. Namely, the values of variable z at the nodes of the look-up table are updated as follows:

$$z_{f(h,p)} = z_{(h,p)} + (\hat{z}_{a(h,p)} - \hat{z}_{(h,p)}) \qquad (21)$$

In other words the approximation error which is present in the $\hat{z}_{a(h,p)}$ and $\hat{z}_{(h,p)}$, is canceled since only the difference $(\hat{z}_{a(h,p)} - \hat{z}_{(h,p)})$ (not the absolute values) is used for adaptation of the nodes of the look-up table. In the next subsection the algorithm proposed above is applied to the adaptation of the friction torque look-up table.

3) Adaptation of the Friction Torque Look-up Table : Suppose that the engine friction torque is overestimated with an offset of $10[Nm]$. Actual values (two values) of the engine friction torque as a function of speed and indicated torque were obtained during an engine start (see Figures 2 and 3). A third value of the friction torque was obtained at idle by averaging the values of the friction torque over a certain interval. Weighting factors were assigned to all the values of the measured engine friction torque. The algorithm described above was applied for adaptation of the friction look-up table. The order of the approximating polynomial is two. Only the offset parameter a_{00} was adapted. The result is plotted in Figure 5. The friction torques before and after adaptation were plotted as white surfaces, and actual friction torque is plotted as black surface. The difference between actual friction torque and the friction torque after the adaptation is $0.77[Nm]$.

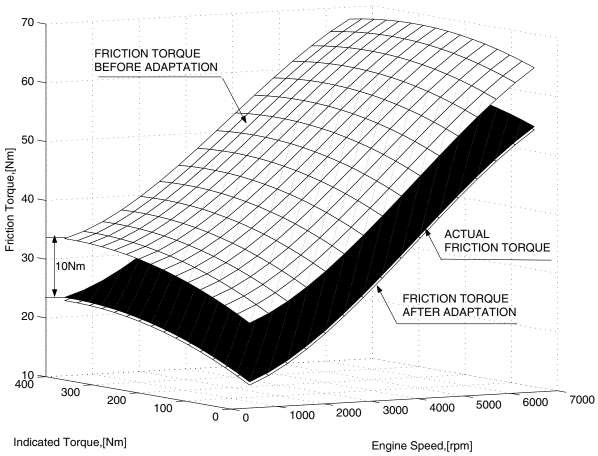

Figure 5
The friction torque is plotted as a function of engine speed and indicated engine torque. The friction torque before the adaptation and after adaptation are plotted as white surfaces. Actual friction torque is plotted as black surface.

The look-up table for the friction torque was updated in the engine electronic control unit and the measurements of the engine speed and brake torque at the next start are plotted in Figure 6. The behavior of the engine speed and torque before adaptation is plotted in Figure 2. Comparison of the Figures 2 and 6 shows that the error $e(t) = J\dot{\omega} - T_{brake}$ is reduced and the control aim (3) is reached with a sufficiently small Δ.

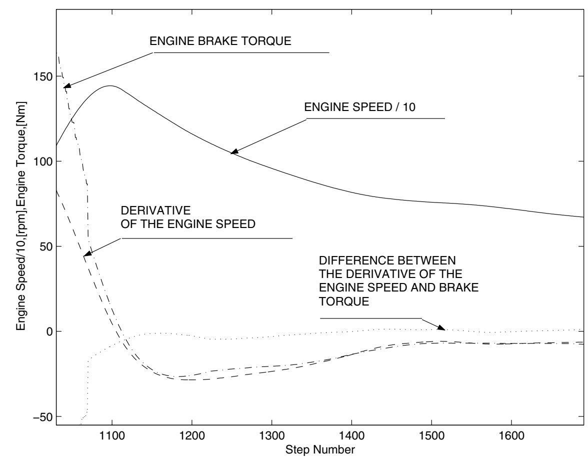

Figure 6
Measurements with the step of 4 ms on the prototype engine. The friction losses have been adapted. Engine speed during a start is plotted with solid line. The values of the engine speed are divided by ten. Engine brake torque is plotted with dashdot line. The derivative of the engine speed multiplied by the inertia moment $J\dot{\omega}$ is plotted with dashed line. The difference $e(t) = J\dot{\omega} - T_{brake}$ is plotted with dotted line.

VI. CONCLUSIONS

New algorithms for real-time estimation of the engine friction torque are developed. Engine friction torque is estimated at start and at engine idle. Recursive and computationally efficient algorithms allow prediction of friction torque for a wide range of speeds and loads even with few new measured points by taking into account physical dependencies used for adaptation of the nodes of the look-up tables. The algorithms make it possible to avoid driveability problems that could result from errors in estimating engine friction torque.

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