IDENTIFICATION OF DRIVELINE PARAMETERS USING AN AUGMENTED NONLINEAR MODEL

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Abstract: Modern approaches for engine control assume the knowledge of the dynamic properties connecting the engine via driveline to the road. A formally identical problem arises when a combustion engine is operated on a test bench, with an electrical machine simulating the wheel load. While in some cases design information allows sufficient estimation of the parameters, in many other cases it may prove more adequate to determine them using measurements. This is usually complicated by the fact that measurements of driveline quantities are disturbed by gas exchange in the cylinder. As this paper shows, a suitable representation of the plant allows to concentrate the disturbances arising from the compression into a nonlinear periodic term acting in parallel to a static nonlinear feedback caused by friction. If standard linear parameter identification methods are used, the effect of this concentrated nonlinearity corresponds to a possibly infinite number of additional poles, which, however, depend on the rotational speed of the shaft. Using this property, it turns out possible to use a standard ARMAX identification approach to determine the model of the driveline. This is confirmed by measurements performed on an engine test bench.

Keywords: Driveline control, transmission modeling, system identification.

1. INTRODUCTION

Driveline control represents one of newest topics in automotive applications. Using such control permits a new trade off in driveline design between vehicle performance, minimizing fuel consumption and emissions and drivability.

Various different controller designs like model reference control (Schwenger, Hinrichsen and Henn, 2004), predictive control (Baumann, et al., 2004) and LQ control (Garofalo, et al., 2002), or even state estimation like Kalman filters (Schwenger, Hinrichsen and Henn, 2004) deal with the problems of driveline actuating. Hereby the given complex driveline is modelled using first principles as simple two mass oscillator which usually sufficiently reflects the first resonance frequency, and so the dominant poles of the real system.

Other approaches (Fredriksson, Weiefors and Egardt, 2002; Kiencke and Nielsen, 2000; Magnus and Nielsen, 2003) combine parameter estimation methods with parameters of inertias and shafts material known *a priori*. Also nonlinear effects, e.g. due to backlash (Lagerberg and Egardt, 2004), can be considered.

In spite of their possible complexity, all these models are rather simple if compared to the possible complexity of a driveline. In particular linear models are a mostly sufficient approximation of the real system. Against this background it may prove sensible to use identification for the whole driveline without relying on a priori information. As linear identification of a nonlinear system boils down to optimal linear approximation, better or equivalent results than with the other approaches can be expected. Furthermore, online identification can be applied to track parameter changes due to wear or failure and so applicable for control and fault detection.

A basic problem for the identification of driveline dynamics is the intermittent nature of gas exchange which shows in the form of torque peaks both in the fired and unfired engine condition and so in the corresponding movement irregularities.

This paper shows a possible description of this effect in terms of a periodic feedback "extension" to the linear model to be identified. As a consequence, a new augmented nonlinear model arises, on which a standard linear identification approach (ARMAX) can be applied. An observation of the poles corresponding to the system extensions shows a variation with the shaft speed, so it is possible to distinguish them from the invariant poles of the target second order system.

As mentioned in the abstract, the problem setup for a vehicle driveline or for its reproduction on a test bench is identical. In this work an AVL dynamical engine test bed has been used with a BMW M47d production engine, whereas correct load torque estimation is assumed.

2. PROBLEM ANALYSIS

The basic model-structure of a driveline consists of the tyre connected via flexible shafts and gearbox with the combustion engine where detailed modelling leads to a complicated high order system.

But for standard-control, just the dominant (representative) system which contains the low frequencies is relevant. So, as far as a low order model (here second order is sufficient) is aimed at, the complex subsystems (for instance gearbox) are included in the damping and resistance and must be treated as linear lumped elements.

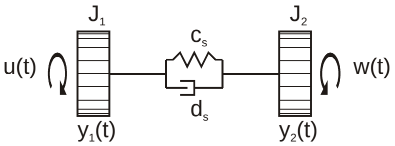

Fig. 1. Mechanics

Fig. 1 shows this linear two mass system with reduced inertias of tyre and drive shaft to inertia J_1 on the left side and the inertias of transmission, propeller shaft and engine reduced inertia J_2 on the right side. Further $u(t)$ represents the torque-input of the road load, $y_1(t)$ the tyre speed, $w(t)$ the torque-input of the engine, $y_2(t)$ the engine speed and c_S, d_S the spring-damper constants of the system.

2.1 Design of the input torque u(t)

In order to obtain the load torque $u(t)$, the well known load equation (1) described by (Kiencke and Nielsen, 2000) with m_v the vehicle mass, v_v the vehicle speed, g the gravity constant, F_{aero} the air resistance, F_{roll} the rolling resistance and α the road grade, can be used.

$$u(t) = M_{load}(t) = -F_{roll}(v(t))r - F_{aero}(v(t))r - m_v g r \sin(\alpha(t)) \quad (1)$$

2.2 Design of the disturbance torque w(t)

On the engine side, the engine torque $w(t)$ will act over the inertial moment J_2 and consist of a static frictional term $w_S(t)$ and a periodic term $w_P(t)$ for compression.

$$w(t) = w_S(t) + w_P(t) \quad (2)$$

Both effects have to be regarded separately. The friction torque is a rotational speed depending force generically approximated by a polynomial of 2nd degree as in the following equation (3) for $y_2(t)>0$.

$$w_S(y_2(t)) \approx b_0 + b_1 y_2(t) + b_2 y_2(t)^2 \quad (3)$$

The effect of the engine temperature is not considered separately, because it varies too slowly to influence the identification process which lasts some seconds. In other words, the temperature effect is absorbed in parameter b_0.

The complex periodic function $w_p(t)$ could be approximated by an angular dependent combination of exponential and sinusoidal function with constant amplitude for unfired engine.

In Fig. 2 the superposition (2) of both effects in $w(t)$ is shown for constant speed.

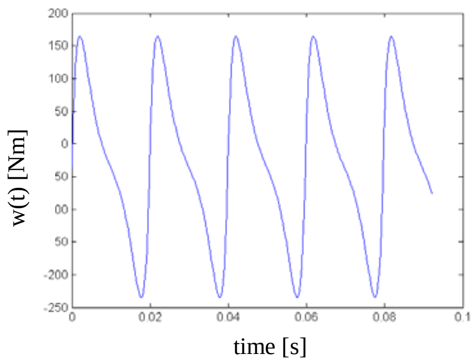

Fig. 2. Engine torque *w(t)*

Periodicity allows considering just sinusoidal disturbances, indeed just a single sine turns out to be sufficient. So to model this process we shall first derive an oscillator whose frequency is input dependent and inserted it in the linear of figure 1.

Notice that alternatively mean value models as done *e.g.* in (Karlsson and Fredriksson, 1999), filtering the speed signal with a time domain or a crank angle based filter (Schmidt and J, 1999) which eliminate the combustion irregularities could be used. But problems caused by such approaches can occur because of nonlinear signal-modification in time domain, which is problematic for identification using linear methods.

Fig. 3 shows a simple oscillator with input dependent varying frequency in time domain. Hereby the input of this system is the engine speed $y_2(t)$. The same integrated gives the angular position, multiplied with the number of pressure pulses f per revolution gives the basis function $v(t)$.

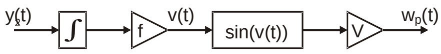

Fig. 3. Oscillator with input depending frequency

V represents a constant gain which reflects the amplitude of periodic torque pulses.

2.3 The linear model connecting engine and tyre

The common state space representation of the linear model in Fig. 1 is given by (4).

$$\begin{aligned}\dot{x}(t) &= Ax(t) + BU(t) \\ y(t) &= Cx(t)\end{aligned} \tag{4}$$

Hereby $x(t)^T=[\varphi(t), y_1(t), y_2(t)]$ describes the state vector with the shaft-torsion $\varphi(t)$, input $U^T=[u(t), w(t)]$ and output $y(t)^T=[y_1(t), y_2(t)]$.

$$A = \begin{bmatrix} 0 & 1 & -1 \\ c_S/J_1 & -d_S/J_1 & d_S/J_1 \\ -c_S/J_2 & d_S/J_2 & -d_S/J_2 \end{bmatrix}$$
$$B = \begin{bmatrix} 0 & 0 \\ -1/J_1 & 0 \\ 0 & -1/J_2 \end{bmatrix}, \ C = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{5}$$

A generic MIMO representation of discretized system (5) can be given by (6) .

$$A(q^{-1})\begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \begin{bmatrix} B_{11}(q^{-1}) & B_{12}(q^{-1}) \\ B_{21}(q^{-1}) & B_{22}(q^{-1}) \end{bmatrix}\begin{bmatrix} u \\ w \end{bmatrix} \tag{6}$$

Setting the engines input $w(t)$ to zero, we obtain the transfer function from $u(t)$ to $y_1(t)$ and using Laplace transforming gives the 2nd order approach (7) plus integrator. Here the polynomial $A(s)$ (which we can also find in $A(q^{-1})$) shows us the representative information which we want to get out of identification.

$$\begin{aligned}\left.\frac{\hat{y}_1}{\hat{u}}\right|_{\hat{w}=0} &= G_{11}(s) = \frac{B_{11}(s)}{A(s)} \\ &= \frac{J_2 s^2 + d_S s + c_s}{s\left(J_1 J_2 s^2 + (J_1+J_2)d_S s + (J_1+J_2)c_S\right)}\end{aligned} \tag{7}$$

Applying measurements taken with unfired engine, we could use the load torque $u(t)$ as input and the load speed $y_1(t)$ as output and try to identify the corresponding transfer function (7) or its discrete time form. Unfortunately, this turns out to yield extremely poor results due to the pressure peaks due to the compression and static nonlinear feedback which do not fit in the classical identification noise framework, as they are highly correlated and periodic (see Fig. 2 for an example).

In both cases as well as for their sum the system structure can be described in terms of the augmented nonlinear system shown in figure 4 with $w(t)=g(y_2(t))$.

The total system consists of a linear part with a nonlinear (dynamic) feedback $w(t)$ correlated to output $y_2(t)$.

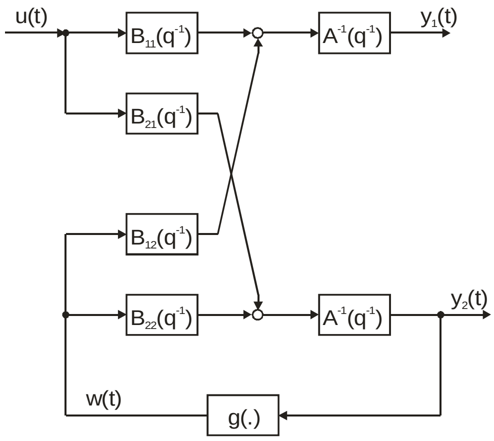

Fig. 4. Augmented nonlinear system

2.4 Driveline – test bench analogy

As mentioned in the introduction, we can find the same structure of the simplified driveline in a test bench system. There, the engine is coupled via a flexible shaft with an electrical machine for load simulation. So, using the inertia of the electrical machine instead of the reduced inertias of tyre and drive shaft we obtain the same system at the test bench as in the driveline. Further the torque input $u(t)$ at the test bench is the electrical torque instead of the road-load torque. It is clear that the parameters for inertias J_1, J_2, spring c_s and damping coefficient d_s are different, but the structure, which is crucial for identification, is still the same.

Because of this analogy, the systems of driveline and test bench are discussed equivalent in the following sections.

3. LINEAR SYSTEM REPRESENTATION

3.1 Effect of nonlinear friction torque (static nonlinear feedback)

The nonlinear friction torque is a speed $y_2(t)$ dependent function $w_S(t)$ which can be described as generically form of eq. (3) as static function in (8)

$$w_S(t) = g_1\left(y_2(t), b\right) \tag{8}$$

where b is a parameter-vector $[b_0, b_1, b_2]$. For sufficient small variations of $y_2(t)$, (8) can be approximated at a working point y_{20} and yields to equation (9).

$$\tilde{w}_S(y_2(t)) \approx \beta_0 + \beta_1 y_2(t) + \Delta_S(t) \tag{9}$$

Hereby an error $\Delta_S(t)$ caused by approximation is introduced into the system. Inserting $w = \tilde{w}_S$ into (6) the transfer function with approximated feedback at working point y_{20} can be derived as (10).

Equation (10) shows that the augmented system has a higher number of poles, of which:

- $Deg(A(q^{-1}))$ number of poles of the linear system at the original position $A(q^{-1})$ and
- $Deg(A(q^{-1})-B_{22}(q^{-1})\beta_1)$ additional poles appear as an effect of the feedback, but their position will depend on β_1 and thus on the operation point y_{20}.

$$\left(A(q^{-1})-B_{22}(q^{-1})\beta_1\right)A(q^{-1})y_1(t)=$$
$$\left(\left(B_{12}(q^{-1})B_{21}(q^{-1})\text{-}B_{11}(q^{-1})B_{22}(q^{-1})\right)\beta_1\right.$$
$$\left.\text{-}A(q^{-1})B_{11}(q^{-1})\right)u(t) \quad (10)$$
$$+B_{12}(q^{-1})A(q^{-1})\left(\beta_0+\Delta_S(t)\right)$$

Notice also that the constant term $B_{12}(q^{-1})\beta_0A(q^{-1})$ (offset) vanishes in the identification routine and $B_{12}(q^{-1})A(q^{-1})\Delta_S(t)$ has the role of approximated disturbances which introduced to the system, so that an ARMAX assumption is a sensible choice.

3.2 Effect of periodic disturbance

The oscillator of Fig. 3 represents a speed $y_2(t)$ dependent function $w_P(t)$ can be described as generically form (11).

$$w_P(t)=g_2\left(y_2(t),\delta,q^{-1}\right) \quad (11)$$

A common approximation based on a finite sum is impossible, because the integrated input (mean value not zero) leaves the region of approximation. So it is easier to map the system in discrete domain (12).

$$w_P(t)=\frac{q^{-1}\sin\left(f\ y_2(t)\right)Vf\ y_2T_S}{1-2q^{-1}\cos\left(f\ y_2(t)\right)+q^{-2}} \quad (12)$$

Linearizing system (12) again at working point y_{20} gives an oscillator with constant frequency $f\ y_{20}$, sample time T_S and amplitude V. The simplified form is shown in (13) with $\delta_1=\sin(f\ y_{20})Vf\ \ y_{20}T_S$ and $\delta_2=2\cos(f\ y_{20})$.

$$\tilde{w}_P(t)\approx\frac{q^{-1}\delta_1+\Delta_P(t)}{1-q^{-1}\delta_2+q^{-2}} \quad (13)$$

Hereby an approximation error $\Delta_P(t)$ is assumed to be correlated the dynamics of the oscillator. The input output representation of the linear system with periodic feedback is derived by substituting $w=\tilde{w}_P$ into (6) results in at a working point y_{20} which yields to (14).

$$\left(1-q^{-1}\delta_2+q^{-2}\right)A(q^{-1})y_1(t)$$
$$=\left(1-q^{-1}\delta_2+q^{-2}\right)B_{11}(q^{-1})u(t) \quad (14)$$
$$+B_{12}(q^{-1})\left(q^{-1}\delta_1+\Delta_P(t)\right)$$

Here we can find the polynomial $A(q^{-1})$ again, this time multiplied by denominator of the oscillator. The constant term $q^{-1}\delta_1$ is expected to vanish in the ARMAX identification routine again.

3.3 Combination of disturbances

For receiving the full model, both disturbances $w=\tilde{w}_S+\tilde{w}_P$ have to be combined and inserted into (6) and so one obtains (15)

$$F_1(q^{-1},y_{20})A(q^{-1})y_1(t)=$$
$$F_2(q^{-1},y_{20})u(t)+F_3(q^{-1},y_{20}) \quad (15)$$
$$+F_4(q^{-1},y_{20})\Delta_S(t)+A(q^{-1})B_{12}(q^{-1})\Delta_P(t)$$

with terms of F_1 (16) related to the output.

$$F_1(q^{-1},y_{20})=$$
$$\left(A(q^{-1})+B_{22}(q^{-1})\beta_1\right)\left(1-\delta_2q^{-1}+q^{-2}\right) \quad (16)$$

F_2 in (17) related to the input,

$$F_2(q^{-1},y_{20})=$$
$$\left(B_{11}(q^{-1})A(q^{-1})+B_{12}(q^{-1})B_{21}(q^{-1})\beta_1\right. \quad (17)$$
$$\left.-B_{11}(q^{-1})B_{22}(q^{-1})\beta_1\right)\left(1-\delta_2q^{-1}+q^{-2}\right)$$

F_3 in (18) related to the offset,

$$F_3(q^{-1},y_{20})=$$
$$A(q^{-1})B_{12}(q^{-1})\left(\beta_0\left(1-\delta_2q^{-1}+q^{-2}\right)+q^{-1}\delta_1\right) \quad (18)$$

and F_4 in (19) related to the static feedback disturbance.

$$F_4(q^{-1},y_{20})=A(q^{-1})B_{12}(q^{-1})\left(1-q^{-1}\delta_2+q^{-2}\right) \quad (19)$$

So for the ARMAX identification routine a model order of least $deg(F_1(q^{-1},y_{20})A(q^{-1}))$ has to be used. The constant term $F_3(q^{-1},y_{20})$ is expected to vanish in the identification process again.

4. IDENTIFICATION

One can now apply the ARMAX approach (Ljung, 1999) for the identification of our problem. We recall that the ARMAX algorithms are designed for plants of the form

$$A^*(q^{-1})y_1(t)=B^*(q^{-1})u(t)+C^*(q^{-1})e(t) \quad (20)$$

As we have seen, the problem can be put in form (15) where term C^* comprises the term of uncertainty. While ARMAX has been conceived for filtered white noise, the iterative nature of the algorithm as well as recurring to the minimization of a robustified quadratic prediction error criterion using an iterative search algorithm, whose details are governed by the properties in (MATLAB), allows to achieve consistent and correct estimations (something witch would be impossible with ARX), albeit with the acceptable price of a high order of the polynomials A^*, B^* and C^*. So a standard robust ARMAX approach as implemented in MATLAB yields an unbiased solution.

There exist indeed methods for identification of (feedback) Hammerstein and Wiener systems as described in (Greblicki and Pawlak, 1986; Guo, 2003; Vandersteen and Schoukens, 1999; Vörös, 1999), but they are not necessary here, because we are not interested in the model of the nonlinearity but only in eliminating its effects.

(Goodzeit and Phan, 2000) proposed a special ARX model based identification algorithm which can separate periodic disturbances. But in presence of friction feedback, such a method is not applicable, because one expects in such a case the nonlinear frictional torque feedback to cause biases in identified parameters.

Due to the integrating characteristic of the plant, system identification has to be done in closed-loop. To excite the system in a broad frequency band, a discrete binary test signal (Isermann, 1991) has been used with a period of one and two seconds.

The ARMAX identification will yield the right values of the poles (plus the additional poles).

In contrast to the dependency on the working point, only the polynomial $A(q^{-1})$ keeps constant. So, the gain of the identified system changes in the same way as the numerator. For obtaining the gain of the transfer function, two possibilities are for disposal: Inertias from manufacturer, or a drag and roll-out experiment developed by (AVL, 2004) can be used.

To identify the periodically disturbed system, some constraints must be kept in mind:

- The amplitude of the disturbance should not be dominant. In simulations it was possible to have disturbance amplitudes up to half of excitation.
- The frequency of the disturbance (excitation) must not lie too near to the resonance frequencies of the system (what anyway would be a very poor choice in real applications).

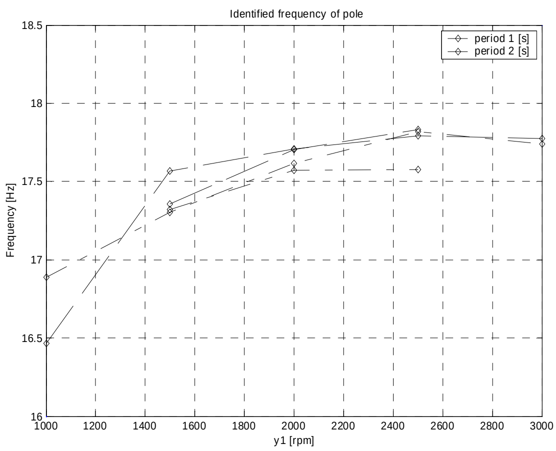

Fig. 5. Frequencies of identified system

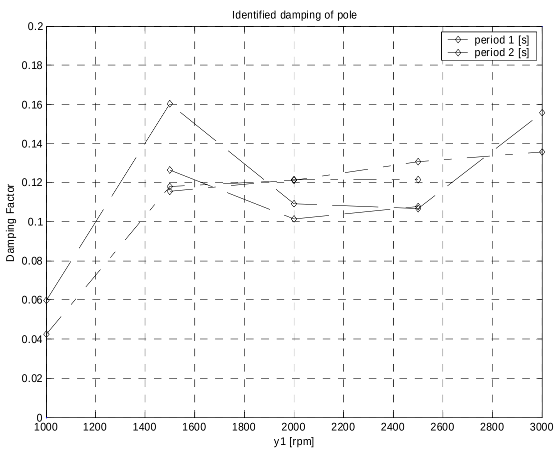

Fig. 6. Damping of identified system

The identification measurements are done with two different excitation periods (one and two seconds) and in different controlled operation points.

The Akaike final prediction error (MATLAB, 2002) for the measured system defined in (21) V_{FPE} is the loss-function, d_{FPE} the number of estimated parameters and N_{FPE} is the number of estimation data was about 0.5 except one at 1.2.

$$FPE = V_{FPE} \frac{1 + \frac{d_{FPE}}{N_{FPE}}}{1 - \frac{d_{FPE}}{N_{FPE}}} \qquad (21)$$

The last two figures show frequency and damping of the poles in different operation points (engine speeds). Hereby a slightly nonlinear dependency on the operation point is obvious. But this behavior is confirmed by the manufacturer of the test-system.

5. CONCLUSIONS

In spite of its rather long derivation, the proposed method allows simple, fast and consistent estimation of driveline parameters. From the used point of view, the method essentially boils down to determine an overparametrized model for a few operating points, to sort out the “good” from the “spurious” poles. The proposed method guarantees robustness and convergence which cannot be shown by the known nonlinear identification methods up to now. Further this method is capable of being used as online application because the calculation effort much lower than those for nonlinear methods.

As mentioned, this method was tested at on a dynamical test bench on which the drag force was simulated by the load of the electrical machine, and provided very good results.

In Fig. 5 and 6 the operating point (speed y_{20} dependent) frequency of the linear system is obvious too. It can be seen as quasi-linear.

Experiments with fired engine did not give convergent identification, because the disturbance amplitude *w(t)* is much higher than load torque *u(t)*.

Further work is needed to make this approach applicable to production driveline systems, because it relies on a correct estimation of both load torque and speed, which are strongly affected by sensor precision.

Another problem in driveline application is the complexity of system excitation. In principle, excitation can be provided both by brakes and engine, but still work is necessary to elaborate a driver and passenger friendly approach.

ACKNOWLEDGEMENTS

The idea of this approach is derived from a testbench identification and control project done in cooperation with LCM (Linz Center of Mechatronics) GmbH and AVL List GmbH Graz.

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