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Inverse Torque Control of Hydrodynamic Dynamometers for Combustion Engine Test Benches

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Abstract— **Hydrodynamic dynamometers can be used for the entire range of combustion engines from cart engines up to large ship engines, are inexpensive and have a small moment of inertia. Due to their strong nonlinearities and the absence of precise models, they are still rarely used for dynamic testing. Against this background, this paper proposes an inverse control of an approximate form determined experimentally. As the paper shows using measurements on a dynamic truck engine test bench, the proposed approach is able to offer a significantly better performance with respect to the classical implementation thus opening a new path for the intended use for dynamic testing.**

I. Introduction

Dynamic combustion engine test benches are used for many purposes, e. g. for calibration of the maps of an engine control unit but also for development of new engine control concepts. If the dynamics of the torque delivered by the brake is sufficiently high, they can be used instead of measurements in a vehicle with strong advantages in terms of reproducibility, costs and time.

Fig. 1 shows the classic setup of such a test bench. The internal combustion engine is the device under test, a load to the engine is applied by a dynamometer to simulate the load the engine would experience in a vehicle. Usually the torque of the internal combustion engine T_E, the dynamometer T_D and the one transfered in the connection shaft T_{st} can be measured next to the engine speed n_E and the dynamometer speed n_D.

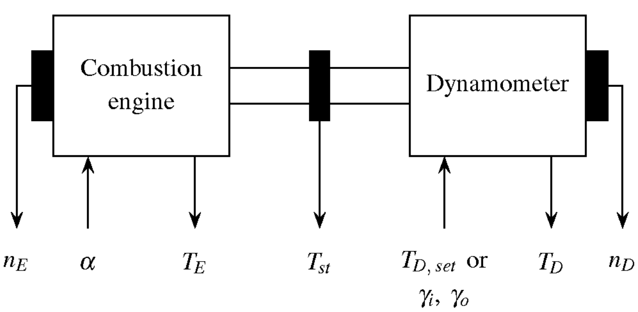

Fig. 1. Setup of a combustion engine test bench

In most cases, the accelerator pedal position α and the set value for the torque of an electric machine $T_{D,set}$ constitute the input variables of the engine test bench. (For more information on modeling of combustion engine test benches with an electric dynamometer see e. g. [1].)

Instead of electric dynamometers also hydrodynamic dynamometers can be used. Today test benches equipped with such a brake (see Fig. 2) are mainly used in stationary tests. The inputs of a water brake are given by the inlet valve position γ_i as well as the outlet valve position γ_o. They combine high power ratings with a low moment of interia and offer especially in case of high power ratings a good alternative.

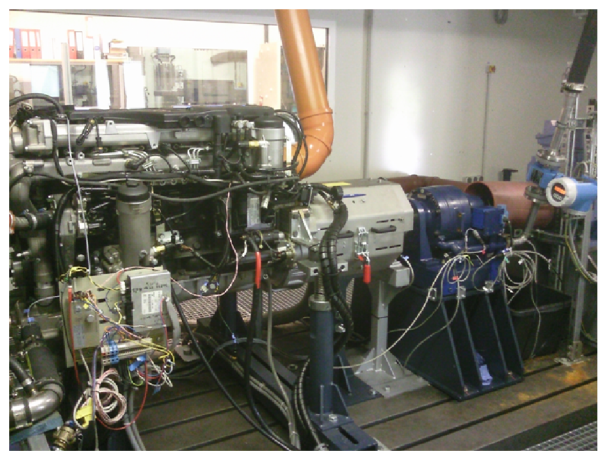

Fig. 2. Combustion engine test bench equipped with hydrodynamic dynamometer (left: combustion engine, right: water brake)

Despite these advantages water brakes are not widely used in dynamic testing as modeling and control is due to their nonlinearities a nontrivial task.

Although the underlying principle is not new, models for systems with variable fill level are rare. The fill level is the key quantity for the resulting dynamometer torque, unfortunately it can not be measured directly. In recent years the focus has changed from first principles models to computational fluid dynamic (CFD) models. The intended use for such models lies in the development of the brake itself, due to the complexity it is (often) not possible to design a controller on this basis.

In [2] a first principles approach is compared with a black box model and a gray box model. Best results were achieved by using the gray box model which combines the simple structure of a first principles model with a data-based part. In [3] the authors developed a stationary model for a variable fill hydrodynamic brake, extended this model in [4] to transients

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and finally designed a closed loop control in [5].

The performance of combustion engine test benches can for example be significantly improved by model reference adaptive control (see e. g. [6]), robust multivariable feedback control (see e. g. [7]) and robust inverse control (see e. g. [1]). Controlling the hydrodynamic brake itself has rarely been treated in the past. Common implementations at test benches make use of simple feedback controllers extended by some maps to compensate deviations.

The design of a controller for the torque of a water brake is described in this work. An approximative inversion of the plant is generated by means of a feedforward control, while a feedback controller compensates uncertainties and disturbances. Only one simple feedback controller needs to be tuned after measuring the nonlinear static maps. The developed controller has been tested on a test bench with a hydrodynamic dynamometer with a maximum power of 420 kW used to load a heavy duty truck engine with an output of approximately 250 kW running at a constant speed of 1000 rpm.

The paper is organized as follows: A brief description on the working principle and the estimation of a Wiener type model of a water brake dynamometer is given in Section II. The following section deals with the developement of the controller for constant speed. Finally, a comparison is made between the developed controller and a standard implementation. Conclusions including future aspects will complete the paper.

II. Hydrodynamic Dynamometers

This section deals with the working principle of a water brake and the developement of a simple model based on real measurements. It turns out, that the behaviour can be described by means of a Wiener type model.

A. Working principle

A hydrodynamic brake consists of two essential parts (see Fig. 3): The rotor driven by the device under test, composed by the turbine wheel and the shaft, and the stator unit composed of the housing and the supply.

A water brake operates on the Föttinger principle (see [8]). In contrast to torque converters or clutches this type of dynamometer has only one rotating part. As a consequence the slip calculated by the relative difference of the rotor and stator speed is always 100%. This results in a large thermal impact on the working fluid making a fluid exchange indispensable.

The energy dissipation process occurs entirely in the working chambers. From entering the water brake through the inlet valve until leaving it by the outlet valve, the fluid passes the gap between rotor and stator unit several times. Most of the thermal impact on the working fluid results from this incidence losses. Furthermore, friction and secondary circulation losses lead to a warming of the fluid.

An electric machine offers sometimes the possibility to feed the braking energy back to the net. The appropriate efficiencies of the machine, the frequency converter, etc. have

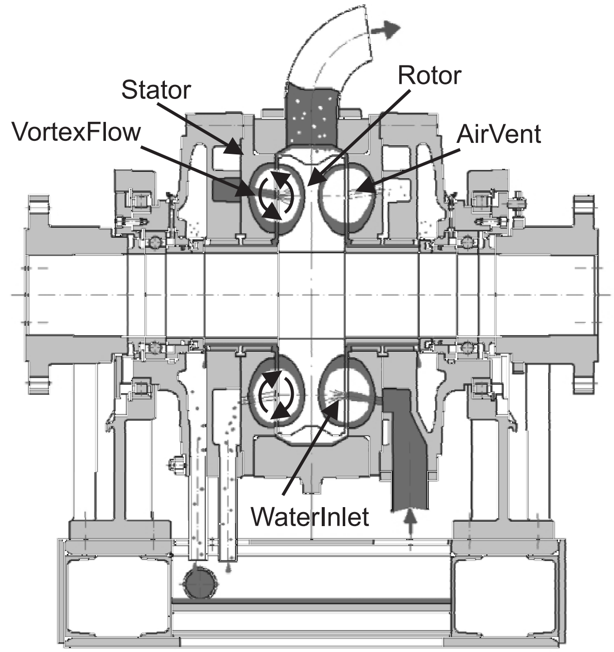

Fig. 3. Section drawing of a hydrodynamic brake

to be taken into account. However, the electrical energy is transformed by resistors into heat in many cases. Using a water brake the whole mechanical energy transmitted by the connection shaft is converted into heat.

A detailed overview of the energy dissipation as well as on the structural design of a hydrodynamic brake is given in [2] and [9] respectively.

B. Modeling of a hydrodynamic dynamometer

Fig. 4 shows the stationary dynamometer torque T_D as a function of the inlet γ_i and the outlet valve position γ_o for a constant speed of $n = 1000$ rpm. The average of the measured torque over several seconds has been calculated after reaching the operating point to achieve accurate results even in the presence of noise.

A valve position of 100% of both valves corresponds to the maximum possible torque. In this case the inlet valve is fully opened and the outlet valve almost closed. Especially for lower speeds a significant leakage flow can be determined for these and similar valve positions.

Fig. 4 and Fig. 8 indicate a band of high gradients in the dynamometer torque T_D, while in other regions the torque T_D is saturated (see Fig. 4). A low fill level according to a nearly closed inlet valve while the outlet valve is wide opened will in any case give a low dynamometer torque T_D. The difference between the actual torque and the maximum possible torque at a wide opened inlet valve and a nearly closed outlet valve is fairly limited.

Similar maps can also be recorded for other speeds. However, for higher speeds the steep rise of the dynamometer torque T_D is shifted to higher values of both valve positions. The insensitive area next to the origin becomes bigger. In this case also the maximum torque is higher.

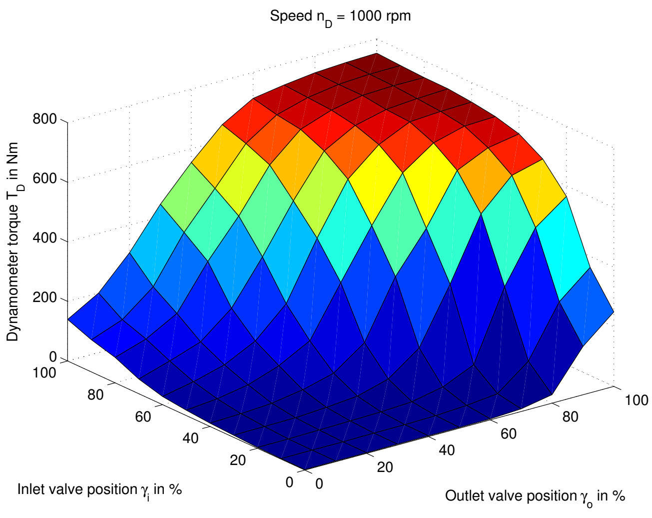

Fig. 4. Dynamometer torque T_D as a function of the valve positions for a constant speed

To increase the performance of the subsequent developed inverse torque control and to obtain more realistic simulation results, it is necessary to combine the above described maps with dynamics. Fig. 5 shows the response of the hydrodynamic dynamometer to steps of the inlet valve position γ_i for different speeds n_D.

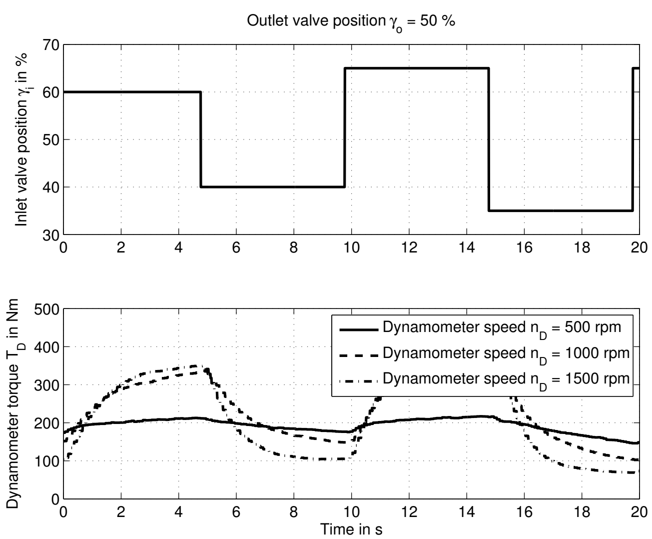

Fig. 5. Response of hydrodynamic dynamometer to steps of the inlet valve position γ_i

The water brake including the valves and the actuators show a low pass behaviour. This behaviour is not only related to steps of the inlet valve position γ_i but also caused by steps of the outlet valve position γ_o and the dynamometer speed n_D.

However, the low pass characteristics shown in Fig. 5 changes with the dynamometer speed n_D. Due to an increased centrifugal force at higher dynamometer speed n_D, a change of the fill level occurs within a shorter time. Therefore, the response time of the water brake to changes of the operational point decreases with higher speed.

In a first attempt, an estimation of the linear dynamics was performed to determine the worst case scenario. For this scenario the controller has been developed and tuned in simulation.

As the characteristic of the water brake changes with the inlet γ_i and the outlet valve position γ_o as well as the speed n_D, a further extension of the linear dynamics in form of LPV contributes to an improvement. For more information on LPV modelling, identification and control see e. g. [10], [11] and [12].

Above observations suggest the usage of a Wiener type model (see Fig. 6) where the linear dynamics is downstreamed by a nonlinear static map $\Psi(\cdot)$. It is

$$\dot{x} = A\,x(t) + B\,u(t) \tag{1}$$

$$z = C\,x(t) \tag{2}$$

$$y(t) = \Psi(z) \tag{3}$$

with x representing the state of the linear dynamics, z the output of the linear dynamics and y the output of the Wiener type model.

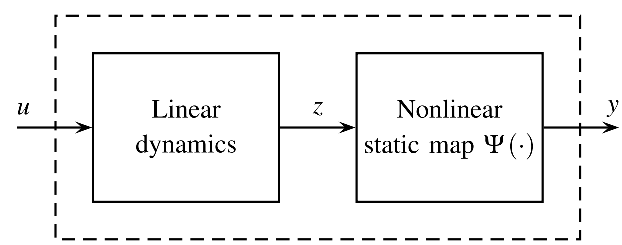

Fig. 6. Wiener type model

The input u consists of the inlet valve position γ_i, the outlet valve position γ_o and the speed n_D. The output y is given by the dynamometer torque T_D.

Common identification procedures for Wiener type models are based on two step approaches, see for example [13], [14] and [15].

III. Inverse Torque Control

Fig. 7 shows the scheme of the inverse torque control. On the basis of the nonlinear static map $\Psi(\cdot)$ the set values for both valve positions are calculated from the desired dynamometer torque $T_{D,\,set}$. A feedback controller is used to compensate uncertainties and disturbance effects. These effects may include model-plant-mismatch as well as disturbances to the hydrodynamic dynamometer caused by an operation point change of the combustion engine.

The determination of the various parts of the controller will be shown for a constant speed below. The extension to variable combustion engine test bench speeds will be discussed in a later paper.

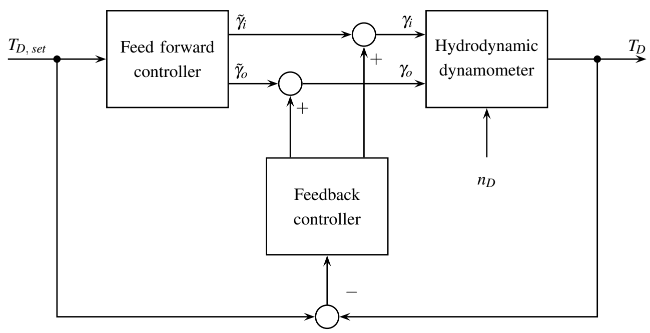

Fig. 7. Control scheme of inverse torque control

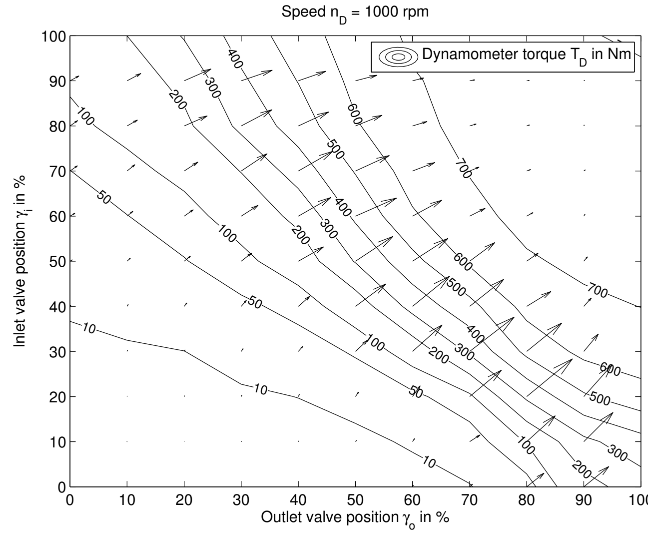

Fig. 8. Contour plot of dynamometer torque T_D and vector field representing the gradient

A. Inversion of the nonlinear static map

As shown in Fig. 4 and Fig. 8, a specific torque can be achieved with an infinite number of combinations of the inlet γ_i and the outlet valve position γ_o. Thus, this system offers an additional degree of freedom for control of the dynamometer torque T_D.

The introduction of an additional condition is required to obtain a unique control input $(\gamma_i,\ \gamma_o)$ achieving a given criterion of optimality. This condition can either be based on measurement quantities, on constraints imposed during the design of the controller or on a combination of both.

The design goals for the controller contain both the tracking of the dynamometer torque T_D as well as constraints of the temperature and the flow rate. For example, the maximum temperature at the outlet as well as the maximum temperature difference between inlet and outlet is specified by the manufacturer of the brake.

Additional conditions may include the least possible movement of one or of both valves compared with the actual configuration or the least possible difference between both valve positions. A further condition could be the operation of both valves closer to the center of the operational area.

In simulation as well as on the combustion engine test bench the following condition was implemented

$$\min_{x} \left\{ \|x - x_0\|_p \right\} \tag{4}$$
$$\text{s. t.} \quad \begin{aligned} T_D &= T_{D,\,set} \\ \underline{x} &\leq x \\ x &\leq \overline{x} \end{aligned}$$

with $p \in \mathbb{N}$ and $x = \begin{bmatrix}\gamma_i & \gamma_o\end{bmatrix}$.

$\underline{x} = \begin{bmatrix}\underline{\gamma}_i & \underline{\gamma}_o\end{bmatrix}$ and $\overline{x} = \begin{bmatrix}\overline{\gamma}_i & \overline{\gamma}_o\end{bmatrix}$ respectively describe some boundaries. $\underline{x}$ and $\overline{x}$ allow on the one hand an avoidance of specific operational areas as those specified by the manufacturer. A certain operation range for the feedback controller persists on the other hand.

The argument in (4) describes the difference between the actual valve positions $x = \begin{bmatrix}\gamma_i & \gamma_o\end{bmatrix}$ and the design point $x_0 = \begin{bmatrix}\gamma_{i,0} & \gamma_{o,0}\end{bmatrix}$. In the implemention p is set to 2 corresponding to the usage of the euclidean norm. Thus, the valves are operated near to the center of their working range offering large movement in both directions.

The gradient of the dynamometer torque T_D for a constant speed $n_D = 1000$ rpm is shown in Fig. 8. In the center of the operational area of both valves the gradient is not maximum, but reaches a high value and offers a trade-off between susceptibility to disturbances and sensitivity.

On the other hand some freedom in the control of the dynamometer torque T_D remains due to the design parameter x_0. For instance, x_0 allows to consider the temperature of the working fluid at the outlet in the control loop.

B. Robustifying feedback control

Neglecting the linear dynamics in Fig. 5 and disturbance effects, it is possible to control the plant solely by means of the above described inversion of the nonlinear static map. However, to take these phenomena into account a feedback controller is introduced subsequently.

The calculation of two control variables based on one error signal is described with reference to Fig. 8. All possible combinations of inlet γ_i and outlet valve position γ_o that lead to one and the same dynamometer torque T_D lie on a continuous curve. Partial deviations from this property can only be observed for very high and very low values for a given speed and the marginal areas.

Notice also that these lines are almost parallel especially in the area with the largest gradient and tilted by about -45° to the axes of abscissae in Fig. 8. Thus, the shortest connection between two of these lines is given by the orthogonal direction – under $+45^\circ$ to the axes of abscissae.

The feedback controller exploits the previous consideration. If the achieved dynamometer torque T_D is less than the desired one $T_{D,\,set}$, certain values are added to both value positions $(\tilde{\gamma}_i,\ \tilde{\gamma}_o)$ calculated by the inversion of nonlinear static map $\Psi(\cdot)$. The situation is reversed for a negative error.

It is

$$\gamma_i = \tilde{\gamma}_i + \Delta\gamma_i \tag{5}$$

$$\gamma_o = \tilde{\gamma}_o + \Delta\gamma_o \tag{6}$$

with

$$\Delta\gamma_i = C_{\left(T_{D,\,set}-T_D\right)\to\gamma_i}(s)\ e \tag{7}$$

$$\Delta\gamma_o = C_{\left(T_{D,\,set}-T_D\right)\to\gamma_o}(s)\ e. \tag{8}$$

While one valve is further closed, the other one is further opened. The fill level of the hydrodynamic dynamometer and therefore the torque T_D can be changed faster compared to an approach using only one valve.

The choice of $C_{\left(T_{D,\,set}-T_D\right)\to\gamma_i}(s) = C_{\left(T_{D,\,set}-T_D\right)\to\gamma_o}(s)$ is a further simplification. If the movement of one of the two valves – caused by rejecting errors in the inversion of the nonlinear static map $\Psi(\cdot)$ or disturbance effects – should for example be limited, different choices of the controllers are possible. Using only one valve is the extremum.

The following simple structure of the feedback controller has been chosen for measurements on the test bench:

$$C(s) = k_P\ (T_{D,\,set} - T_D) + \frac{k_I}{s}\,(T_{D,\,set} - T_D) \tag{9}$$

with $k_P = 0.18$ and $k_I = 0.07$.

Especially fast changes of the operating point result in control variables slightly smaller or larger than the (soft) boundaries specified in the inversion of the nonlinear static map $\Psi(\cdot)$. Under infrequent circumstances the controller would require control variables smaller than 0% or greater than 100%. In this case, problems can be resolved by introducing an anti-windup-loop.

IV. Measurements at the Combustion Engine Test Bench

The objective of an engine test bench is to operate an internal combustion engine like in a car or a heavy duty truck. This objective is equivalent to the tracking of a torque and a speed profile at the crank shaft of the engine. The engine to be tested as well as the dynamometer are the actuators.

In the present case the speed of the test bench is controlled with the standard controller acting on the combustion engine. The torque at the crank shaft is controlled via the dynamometer.

The results when using the standard controller for the torque and the developed control structure are compared in the following: The desired $T_{D,\,set}$ and the measured dynamometer torque T_D using both control concepts are shown in the first plot of Fig. 9. The effects of controlling the dynamometer torque T_D on the speed of the test bench are depicted in the second plot. Furthermore, a comparison is given between the valve positions and the temperature of the working fluid at the outlet.

The tracking of the reference torque can significantly be improved by using the proposed controller. The rise time is shortened by an rapid movement of both valves. In case of an emptied brake, this also leads to a reduction of the response time.

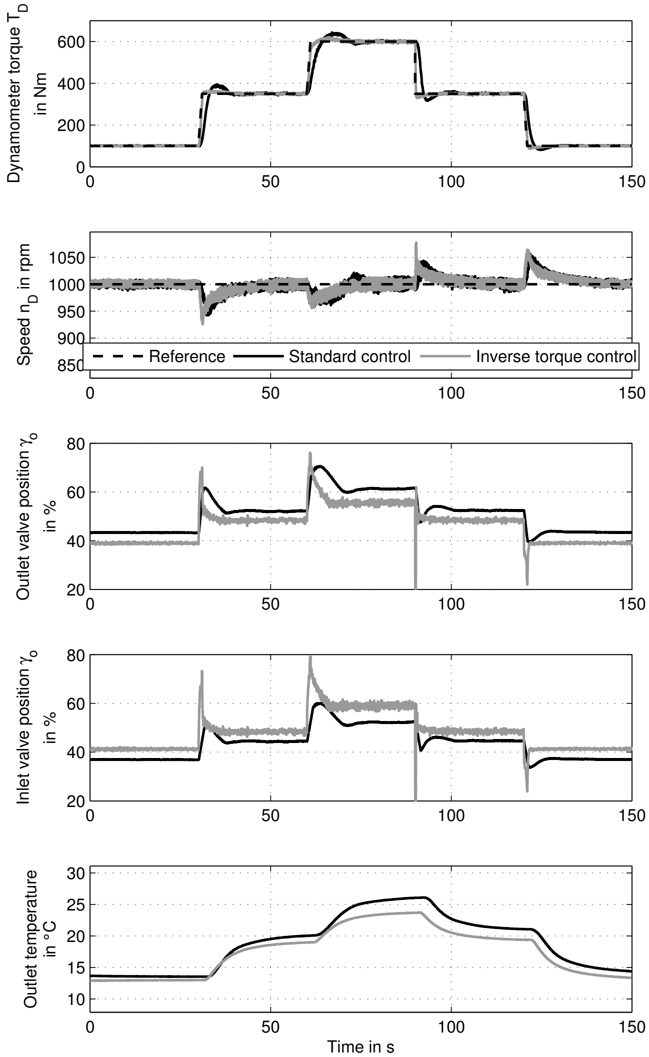

Fig. 9. Comparison between developed controller and standard implementation – Trial 1

While over- and undershootings can be determined using the standard control, the developed control shows this behaviour hardly. There is only a short, limited undershoot after a sudden and large reduction of the desired dynamometer torque $T_{D,\,set}$.

Furthermore, it should be noticed that the temperature at the outlet is less using the proposed control. More water flows through the brake according to a more opened inlet and outlet valve. However, an adjustment is possible by a shift of the "design point".

A simple comparison between the results suggests an increased performance using the controller developed in this paper. However, the control of a combustion engine test bench is always a trade-off between the two outputs. A quantitive comparison using the weighting functions

$$J_{T_D} = \sqrt{\frac{1}{N}\sum_{k=1}^{N}\left|\frac{T_D - T_{D,\,set}}{T_{D,\,set}}\right|^2} \tag{10}$$

and

$$J_{n_D} = \sqrt{\frac{1}{N}\sum_{k=1}^{N}\left|\frac{n_D - n_{D,\,set}}{n_{D,\,set}}\right|^2} \tag{11}$$

is done and summerized in a normalized fashion in Table I. N characterizes the total number of measurement points.

TABLE I
COMPARISON BETWEEN STANDARD CONTROL AND INVERSE TORQUE CONTROL

	Costs	
	J_{T_D}	J_{n_D}
Standard control	100	100
Inverse torque control	25.4	98.6

Using the developed inverse torque control leads to nearly the same values of the weighting function of the test bench speed, while the improvement in tracking the torque is significant.

However, the parametrization of the feedback controller affects the coupling from torque control to test bench speed. Through a more smooth setting of the control the impact is reduced, but this also decreases the torque tracking performance.

Other test runs allow the same conclusions. The overall performance of the system is greatly increased.

V. Conclusion and Outlook

Measurements on a real engine test bench indicate that a water brake can be described by a Wiener type model. This paper describes the design of a controller for the torque of such a plant. The feedforward controller calculates the control signals for inlet and outlet valve according to the inversion of the nonlinear static map $\Psi(\cdot)$ of the Wiener type model. A feedback controller is used to compensate model-plant-mismatch, errors in the inversion of the nonlinear static map $\Psi(\cdot)$ and disturbance effects.

Compared with available standard implementations operating the developed controller leads to an increased performance. The usage of a hydrodynamic brake is not limited to stationary measurements. The presented novel approach offers the opportunity to run dynamic tests like the Heavy-Duty FTP Transient Cycle (see [16]) on test benches equipped with hydrodynamic dynamometers.

To release the limitation to fixed speeds a map approximating the dynamometer torque T_D as a function of the valve positions and the actual speed of the test bench n_D is calculated by interpolation between the recorded nonlinear static maps $\Psi(\cdot)$. First results both in simulation as well as on the test bench show a promising behavior.

To handle all the above mentioned design goals the proposed Wiener type model has to be extended to the temperature at the outlet (or the difference temperature between inlet and outlet). Improved results should also be possible by using dynamics depending on the operating point in the control design instead of the worst case scenario.

The employment of other structures for the feedback control and the proof of robustness could also be of interest.

However, the next step in the overall process is given by the development of a control for the speed of the test bench. Subsequently, as described in [1], the couplings between internal combustion engine and dynamometer will be considered by using a multi-input multi-output model in control design.

VI. Acknowledgments

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