Inverse Torque Control of Hydrodynamic Dynamometers for Combustion Engine Test Benches


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 transferred in the connection shaft $T_{st}$ can be measured next to the engine speed $n_E$ and the dynamometer speed $n_D$.

In most cases, the accelerator pedal position $\alpha$ 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 $y_I$ as well as the outlet valve position $y_O$. 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)](image)

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 cannot 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...
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 development of the
controller for constant speed. Finally, a comparison is made
between the developed controller and a standard implemen-
tation. Conclusions including future aspects will complete
the paper.

II. HYDRODYNAMIC DYNAMOMETERS

This section deals with the working principle of a water
brake and the development 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
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

B. Modeling of a hydrodynamic dynamometer

Fig. 4 shows the stationary dynamometer torque \( T_D \) as
a function of the inlet \( \gamma_i \) and the outlet valve position \( \gamma_o \)
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.
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 $\gamma_i$ for different speeds $n_D$.

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 $\gamma_i$ but also caused by steps of the outlet valve position $\gamma_o$ 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 $\gamma_i$ and the outlet valve position $\gamma_o$ 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

$$\dot{x} = Ax(t) + Bu(t) \quad (1)$$

$$z = Cx(t) \quad (2)$$

$$y(t) = \Psi(z) \quad (3)$$

The input $u$ consists of the inlet valve position $\gamma_i$, the outlet valve position $\gamma_o$ 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.
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 \( \gamma_i \) and the outlet valve position \( \gamma_o \). 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

\[
\min_x \left\{ \|x - x_0\|_p \right\}
\]

\[
T_D = T_D, set
\]

s. t. \( \bar{x} \leq x \leq \bar{x} \)

with \( p \in \mathbb{N} \) and \( x = [\gamma_i \ \gamma_o] \).

\( \bar{x} = [\bar{\gamma}_i \ \bar{\gamma}_o] \) and \( \bar{x} = [\bar{\gamma}_i \ \bar{\gamma}_o] \) respectively describe some boundaries. \( \bar{x} \) and \( \bar{x} \) 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 \( x = [\gamma_i \ \gamma_o] \) and the design point \( x_0 = [\gamma_i, 0 \ \gamma_o, 0] \). In the implementation \( p \) 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 \( n_D = 1000 \text{ rpm} \) 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 \( \gamma_i \) and outlet valve position \( \gamma_o \) 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 \(-45^\circ\) 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 valve positions \((\gamma_i, \gamma_o)\) calculated by the inversion of nonlinear static map \( \Psi(\cdot) \). The situation is reversed for a negative error.
It is
\[ \gamma_i = \tilde{\gamma}_i + \Delta \gamma_i \] (5)
\[ \gamma_o = \tilde{\gamma}_o + \Delta \gamma_o \] (6)
with
\[ \Delta \gamma_i = C(TD, set - TD) \rightarrow \gamma_i(s) e \] (7)
\[ \Delta \gamma_o = C(TD, set - TD) \rightarrow \gamma_o(s) e. \] (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 \( C(TD, set - TD) \rightarrow \gamma_i(s) \) 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:
\[ C(s) = k_p (TD, set - T_D) + \frac{k_i}{s} (TD, set - T_D) \] (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 \( TD, set \) 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.
and

\[ J_{\text{IP}} = \sqrt{\frac{1}{N} \sum_{k=1}^{N} \left| \frac{n_D - n_{D, \text{set}}}{n_{D, \text{set}}} \right|^2 } \]  \hspace{1cm} (11) 

is done and summarized in a normalized fashion in Table I. \( N \) characterizes the total number of measurement points.

**TABLE I**

<table>
<thead>
<tr>
<th>Method</th>
<th>Costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard control</td>
<td>100</td>
</tr>
<tr>
<td>Inverse torque control</td>
<td>25.4</td>
</tr>
</tbody>
</table>

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.

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**REFERENCES**