Development of a Dynamic Engine Brake Model for Control Purposes

X.L.J. Seykens[1*], R.S.G. Baert[1]

F.P.T. Willems[2], W. Vink[2]

I.T.M. van den Heuvel[3]

1 Eindhoven University of Technology, The Netherlands
2 TNO Automotive, The Netherlands
3 DAF Trucks, The Netherlands
** Corresponding author ; e-mail : X.L.J.Seykens@tue.nl*

Abstract — This paper presents the extension of an existing mean value dynamic engine model with new models for the combination of a compression release brake and an exhaust valve brake. The focus is on the prediction of engine brake torque, exhaust gas temperatures and mass flow rates. The implemented models are tuned on the basis of steady-state engine data. Transient simulation results, i.e. for the activation of the engine brake, show good agreement with measurements. The developed model has been used to analyze the possibility to use the engine brake to facilitate automatic gear switching. This has been verified by measurements on an actual vehicle and resulted in much smoother and faster gear shift.

INTRODUCTION

Engine brakes serve as a necessary alternative to wheel brakes for vehicle speed reduction. By applying the engine brake wheel brakes are relieved. This reduces wheel brake maintenance frequency and costs, helps to prevent brake overheating and allows for a faster reduction in engine speed. The faster reduction in engine speed may also be used to facilitate automatic gear switching.

Traditionally, engine braking is performed by closing a throttle (in the form of a butterfly valve) in the exhaust system. Compression release brakes, in which the exhaust valve is opened at the end of the compression stroke, are more and more applied, [1][2],[3]. During compression release braking, the fuel injection is stopped and the engine functions as an air pump. By opening the exhaust valve at the end of the compression stroke the work performed by the piston during the compression stroke is not (partially) regained during the expansion stroke. The resulting negative work causes the engine speed to decrease. This technology is closely related to variable valve train operation, which is of great interest in engine development.

During engine braking, manifold conditions (temperatures and pressures) and mass flow rates differ significantly from the conditions during normal engine operation. These conditions greatly influence the efficiency of the engine aftertreatment system. Therefore, for engine controller development, not only an accurate simulation of the engine brake torque and engine speed, is required, but also the prediction of exhaust temperatures and mass flow rates.

Dynamic engine models used for powertrain simulations that include an engine brake are most frequently crankangle based models, see e.g. [4] and [5]. The justification for this is that the relevant time scales of the relevant processes for switching from normal engine operation to braking operation are small and in the order of a single cylinder event. However, the higher accuracy obtained by applying a greater temporal resolution comes at the cost of an increase in computational effort. Therefore, in order for a model to be useable in real-time applications, as are typical for control algorithm validation (e.g. as part of a Hardware-In-the-Loop environments), the temporal resolution has to be decreased to limit the computation time. Here, mean value engine models, such as TNO's DYNAmic engine MOdel DYNAMO [6], are a valuable tool. DYNAMO is implemented in a Matlab/Simulink environment and relies heavily on engine measurement data. It is a tool for concept studies and (model-based) controller development rather than a tool for fundamental analysis.

Without significant modeling efforts, DYNAMO is already capable of simulating engine brake torque with good accuracy. However, for the simulation of mass flow rates and exhaust temperatures additional modeling is necessary. Accurate modeling of the engine dynamics during engine braking is of great importance for controller development in the area of exhaust gas aftertreatment systems and, as a first application of the model used in this study, automatic gear switching.

In this paper, the extension of TNO's existing dynamic engine model with new models for the combination of compression release brake and exhaust valve brake, is presented. The focus will be on the improvement of mass flow rate prediction, which is closely related to the simulation of turbine behavior, and exhaust temperatures. First, the implementation of the new engine brake models will be presented. The brake models are tuned on the basis of steady-state engine data. The tuning process is not described in detail. Only the final results will be shown as a model validation. The complete engine model is then validated on the basis of both steady state and transient, i.e. for the activation of the engine brake, engine data. Finally, an example of model application will be presented: the developed model is used to examine the influence of engine braking on automatic gear switching. Measurements on an actual vehicle are performed and are shown for verification.

1 IMPLEMENTATION OF BRAKE MODELS

The engine brake models are implemented in a DYNAMO model of a 12.9 liter 6 cylinder DAF MX diesel engine. The used approach and technology can, however, also be applied to conventional diesel engines provided they have variable valve train equipment. In figure 1 the architecture of the considered engine is depicted.

The exhaust valve brake is positioned just downstream of the turbine. The exhaust valve brake is explicitly added to the current model as an additional sub-model. Since DYNAMO is a mean value model, the resolution in time is larger than one engine cycle, i.e. in-cylinder phenomena are not explicitly modeled. As a result, the motion of the exhaust valve (gas exchange) is not described. The compression release brake can therefore be modeled implicitly only, i.e. by capturing its influence on the important engine variables. Table 1 shows an overview of the adaptations made to the existing mean value model with the required quantities that need to be determined.

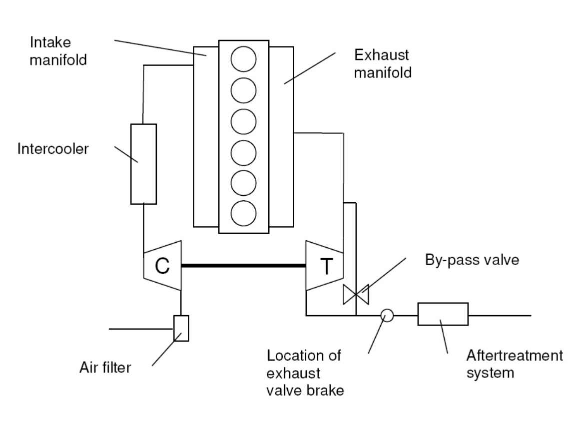

Figure 1
Engine architecture.

TABLE 1

Overview of new brake models and required quantities

Exhaust valve brake (ExhB)	
Module	**Required quantities**
Exhaust valve brake component	$A_{eff} = C_d \cdot A$; K_{pipe}
Diesel Nucleus	η_{vol} ; P_{pump} ; ΔT_{engine}
Turbine	A_{eff} ; N_{turbo} ; $\Delta T_{turbine}$
Compression release brake (CRB)	
Module	**Required quantities**
Diesel Nucleus	η_{vol} ; P_{CRB} ; ΔT_{engine}
Turbine	A_{eff} ; N_{turbo} ; $\Delta T_{turbine}$

1.1 Exhaust valve brake component model

As mentioned, the exhaust valve brake is added as a new component to the mean value model. This component consists of a volume and a restriction with a variable flow area. An exhaust valve brake is a device that essentially creates a major restriction to the compressible flow in the exhaust system, thus causing a substantial exhaust backpressure. This backpressure remains in the cylinder for the entire exhaust stroke (exhaust valve open) and performs negative work on the piston. The result is a negative torque that slows down the engine.

If the exhaust throttle is in the open position, the part of the exhaust system containing the exhaust valve brake is modeled straightforwardly as a duct with a 'pipe loss' coefficient K_{pipe}:

$$\Delta p = \frac{1}{2}\rho K_{pipe} v^2 = \frac{1}{2}\frac{RT}{p}\dot{m}^2 \frac{K_{pipe}}{A_{pipe}^2} \tag{1}$$

where ρ is the gas density, v is the gas velocity, $\dot{m}$ is the mass flow rate, A_{pipe} is the flow area of the exhaust duct, p is the pressure, T is the temperature and R is the specific gas constant. The pressure is taken equal to the mean pressure over the restriction.

To model the resistance to the flow as performed by the closed exhaust throttle (butterfly valve with a small orifice), the mass flow rate through the exhaust valve brake is described by the compressible valve flow equation:

$$\dot{m} = C_d A\, p_1 \sqrt{\left(\frac{2\kappa}{\kappa-1}\right)\frac{1}{RT_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{2}$$

where C_d the discharge coefficient, A the equivalent geometrical flow area and κ the ratio of the specific heats. Pressures and temperatures in (2) are total pressures and temperatures. However, the dynamic part of these quantities can be neglected. Index 1 and 2 refer to respectively the upstream and downstream conditions. The discharge coefficient C_d has to be tuned to measurement data. An instantaneous switch is made between equation (1) and (2) when the exhaust valve brake is activated.

Equation (1) and (2) require the pressure upstream of the butterfly valve to be known. However, in current DYNAMO, the output of the turbine model is a mass flow rate. In order to be able to connect the butter-fly resistance model to the turbine model, a volume element, i.e. the duct between the two components, has to be added. In this volume, pressure is calculated using the ideal gas law:

$$\frac{dp_{volume}}{dt} = \frac{RT_{volume}}{V}\cdot\left(\dot{m}_{volume\ in} - \dot{m}_{volume\ out}\right) \tag{3}$$

where V is the volume.

Since the exhaust valve brake is positioned very close to the turbine outlet, the heat loss to the surroundings in the connecting duct is neglected. The temperature in the volume is assumed to be identical to the temperature of the incoming exhaust gas, i.e. turbine out temperature.

1.2 Adaptation of diesel nucleus model

The diesel nucleus component in DYNAMO calculates the engine torque generation, the engine mass flow rates to and from the engine and the temperature of the exhaust gas entering the exhaust manifold. Both the exhaust valve brake and the compression release brake (always used in combination with the exhaust valve brake) have great influence on these variables. First, the engine torque is considered followed by the engine mass flow rate and the temperature difference across the engine, i.e. the exhaust gas temperature.

1.2.1 Engine brake torque

In current DYNAMO, the engine torque is computed from:

$$Torq = \frac{\left(P_{ind} + P_{pump} - P_{friction}\right)60}{2\pi N_{engine}} \tag{4}$$

where P_{ind} is the indicated work by combustion, P_{pump} is the pumping work performed by the engine, $P_{friction}$ are the frictional losses and N_{engine} is the engine speed. Because the fuel injection process is stopped during engine braking, the indicated power is zero. The frictional power is mainly a function of engine speed. It is assumed that the frictional power during braking is the same as for normal engine operation. The computation of the frictional power is therefore not altered. The exhaust valve brake interacts with the torque generation through the pumping work. The increase in backpressure results in an increase in the pumping work. The pumping work during normal engine operation is modeled as:

$$P_{pump} = \left[\left(p_{int} - p_{exh}\right)\cdot PMEP_1 + PMEP_2\right]\cdot V_{Swept}\cdot\frac{N_{Engine}}{120} \tag{5}$$

where p_{int} and p_{exh} are respectively the pressure in the intake and exhaust manifold and V_{swept} is the total swept volume of the engine. The coefficients $PMEP_1$ (*P*umping *M*ean *E*ffective *P*ressure) and $PMEP_2$ have to be fitted to engine data. This model is also used during activation of the exhaust valve brake. Preliminary simulation results indicated that the values of both the coefficients $PMEP_1$ and $PMEP_2$ used for normal engine operation need not be changed to model the pumping power during engine braking. However, the increased backpressure during braking, i.e. negative pressure difference across the engine, required the operating range to be extended to larger negative pressure differences across the engine.

In addition to the increased pumping losses by the increased backpressure, resulting from the activation of the exhaust valve brake, the compression release brake further increases the braking power (i.e. larger negative engine torque). The pumping losses are assumed to be unaffected by the application of the compression release brake. The influence of the compression release brake on the engine torque is captured by adding a new power term P_{CRB}. By opening the exhaust valve at the end of the compression stroke, the engine performs compression work which is not (partially) regained during the subsequent expansion stroke. The performed compression work is mainly dependent on the amount of mass inside the cylinder at the start of compression. This in-cylinder mass is determined by the gas-exchange process, which is greatly affected by the pressure difference across the engine. The compression release brake power is therefore made a function of the pressure difference across the engine:

$$P_{CRB} = MEP_{CRB}\left(\Delta p_{engine}\right)\cdot V_{swept}\cdot\frac{N_{engine}}{120} \tag{6}$$

where MEP_{CRB} is the mean effective pressure corresponding to the compression release brake. It is implemented as a one dimensional table in DYNAMO using the pressure difference across the engine as an input. The table is filled on the basis of steady-state engine measurements. It has to be noted here that the timing of the compression release brake is held fixed. The influence of compression release brake timing on engine behaviour was no subject in current study.

1.2.2 Engine aspirated mass flow rate – volumetric efficiency

In the original model, the aspirated mass flow by the engine $\dot{m}_{aspirated}$ is taken proportional to the volumetric efficiency η_{vol} and a theoretical mass flow rate $\dot{m}_{theoretical}$ based on intake manifold conditions. The pressure difference across the engine is decisive for the gas exchange process and thus for the degree of filling of the cylinders with fresh charge. Pressure oscillations in the manifolds (i.e. manifold tuning), which are dependent on engine speed, also influence the volumetric efficiency. Therefore, in general, the volumetric efficiency is implemented as a map with the pressure difference across the engine and engine speed as input. The used model is given by:

$$\dot{m}_{aspirated} = \eta_{vol}\cdot\dot{m}_{theoretical} = \eta_{vol}\left(\Delta p_{engine}, N_{engine}\right)\cdot\frac{V_{swept}N_{engine}p_{intake}}{120RT_{intake}} \tag{7}$$

For Heavy Duty applications, engine speed effects on volumetric efficiency are limited. Therefore, only the pressure difference across the engine is used as an input.

To capture the influence of engine braking on the volumetric efficiency, only the operating range of the used map has to be extended to larger negative pressure differences. No significant differences are found between the volumetric efficiencies for the exhaust valve brake and the combination of the exhaust valve brake and the compression release brake. Therefore, one operating map is used for both engine brake configurations.

1.2.3 Exhaust gas temperature

In current DYNAMO, the temperature of the gas leaving the engine is computed as the sum of the intake manifold temperature and a certain temperature rise over the engine. For a given fuel, this temperature rise is dependent on the mass of fuel injected, engine speed and charge composition. The latter is modeled through an adjusted air-fuel ratio λ in which the amount of exhaust gas residual in the charge is incorporated. Furthermore, a correction for the start of injection is implemented. During engine braking, the fuel injection is instantaneously stopped. Extrapolation of the current model to the limits $\dot{m}_{fuel}\rightarrow 0$ and $\lambda\rightarrow\infty$ gives inaccurate results. Therefore, as a first attempt, the temperature rise across the engine is directly taken from measurements and implemented in a table as function of engine speed.

The engine condition can change stepwise from firing condition to engine braking, but the engine temperature will change more smoothly. Some dynamics has therefore been added to the engine out temperature model to capture these dynamics. When switching to “engine braking” the engine out temperature changes with a first order response from the current temperature to the steady-state “engine braking” temperature T_{EB_steady}:

$$\dot{T} = \frac{1}{\tau}\left(T_{EB\_steady} - T\right) \tag{8}$$

The time constant τ of this first order filter has to be fitted to transient test results.

1.3 Adaptation of turbine model

As mentioned before, the existing DYNAMO engine model is not capable of accurately simulating the mass flow rates during engine braking. This is mainly caused by the fact that the behavior of the turbo is not correctly captured. Analysis of preliminary simulation results learned that the pressure ratio over the turbine (p_{in}/p_{out}) is very low when using the exhaust brake. Consequently, the turbo speed is very low (10000 – 40000 RPM) too. These two factors cause the turbine to operate far outside the operating range described in the turbine manufacturer's map. The operating range described by the compressor operating map is somewhat larger. For the steady-state measurements used in this study, the operating range lies just inside the map. However, it is possible that during transient operation, the compressor operates outside of the operating range for which data is available. At present, no corrections are made for this.

The fact that the turbine operates outside of the turbine map, is the main reason for the deviations between measured and simulated mass flow rates during engine braking. The turbine efficiency derived from the maps is not correct which results in a severe under-prediction of turbospeeds. This finally results in too low mass flow rates delivered by the compressor and inaccurate simulation of the manifold conditions.

For a good verification of engine behavior during exhaust brake operation, compressor and turbine maps should be made available for very low turbo speeds and pressure ratios. However, this data was not available in this project. As a pragmatic alternative to obtaining compressor and turbine maps for wider operating ranges, a model transition is proposed here. If the turbo speed drops below 42,000 RPM and the pressure ratio over the turbine drops below 1.2, the turbine model changes into a simple compressible flow resistance. The mass flow rate through this restriction is again given by (1). The compressor model has not been changed for aforementioned reasons.

The effective flow area $A_{eff}\left(=C_d \cdot A_{geo}\right)$ in the compressible valve flow equation, see (1), describing the mass flow rate through the turbine, is tuned on the basis of steady state measurements for the combination of the exhaust valve brake and compression release brake. This effective flow area is held constant for all operating conditions. Figure 2 gives a comparison between the resulting simulated mass flow rates and the measured mass flow rate for different engine speeds. An acceptable accuracy is present (max. deviation <5% with respect to measured value). Data presented in the figures is normalized on the basis of the maximum value from the corresponding measurements.

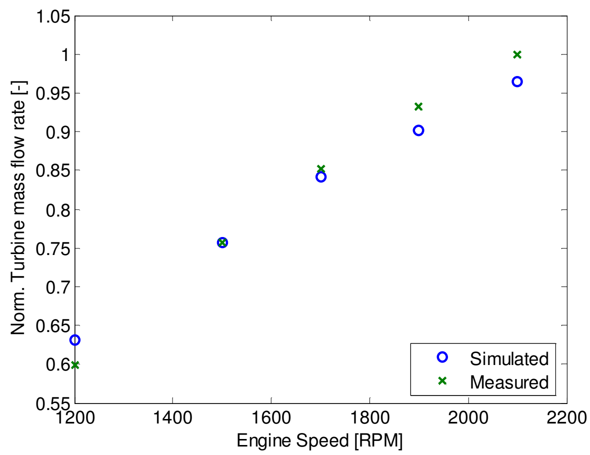

Figure 2
Steady-state simulated and measured normalized turbine mass flow rate

Apart from the turbine mass flow rate, the turbospeed and temperature drop over the turbine must also be described. The turbospeed is required in the compressor model to compute the compressor mass flow rate. The turbine out temperature is of great importance for the exhaust aftertreatment system. Both variables are fitted directly on steady-state measurement data. The determination of the turbospeed, which is proportional to the mass flow rate through the turbine, is based on the engine speed. The turbine out temperature is computed as a function of the exhaust manifold temperature.

2 SIMULATION RESULTS

The implemented engine brake models are tuned on the basis of steady-state measurement data. In this section, first steady-state simulation results for the complete engine model will be compared with measurements for model validation. Here, results will be shown for both the exhaust valve brake and the combination of exhaust valve brake and compression release brake. Thereafter, the dynamics of the model will be validated through a comparison between transient simulation results and measurements, i.e. for the activation of the engine brake. Transient measurement data was only available for the configuration of the compression release brake (CRB) in combination with the exhaust valve brake (ExhB). The important variables to be considered here are the engine brake torque, which is of great importance for the engine braking efficiency, the aspirated mass flow rates and exhaust gas temperatures which are of great significance for the exhaust gas aftertreatment system.

2.1 Simulation results steady-state operation

The steady-state results shown in this section are obtained by using the full engine model in DYNAMO. The only predefined model inputs are the engine speed, fuel mass flow rate (zero during engine braking) and ambient conditions. These inputs are taken from the engine measurements.

Figures 3 to 5 show a comparison between measurements and simulations for respectively the engine torque, aspirated air mass flow rate and exhaust manifold temperature. In the figures, the results for both the configuration of the exhaust valve brake (EhxB) and the combination of the exhaust valve brake and the compression release brake (CRB) are shown.

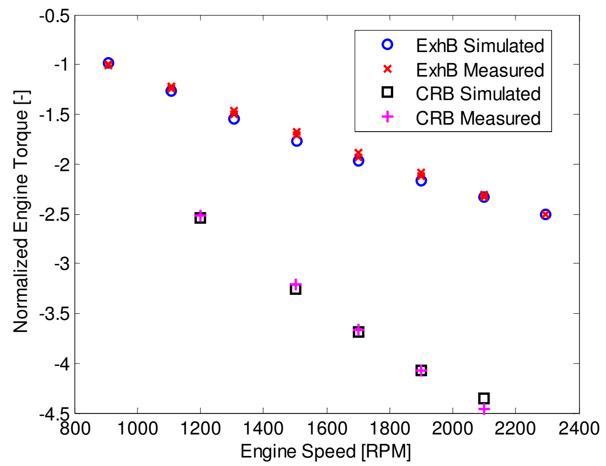

Figure 3
Comparison steady-state simulated and measured engine torque for both the exhaust valve brake and the combination of the exhaust valve brake and compression release brake.

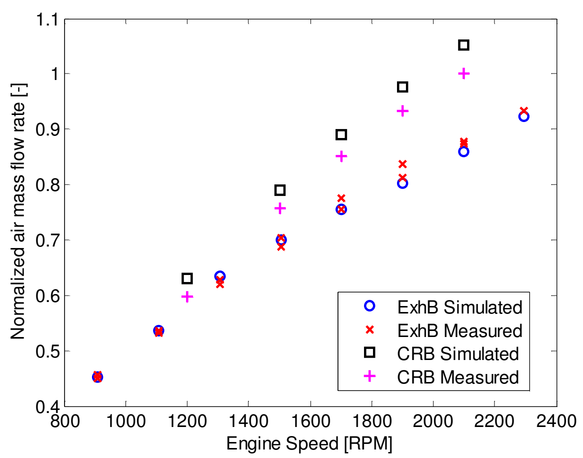

Figure 4
Comparison steady-state simulated and measured aspirated air mass flow rate for both the exhaust valve brake and the combination of the exhaust valve brake and compression release brake.

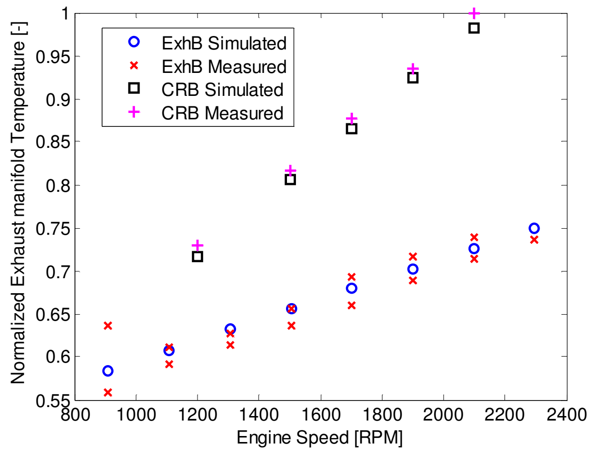

Figure 5
Comparison steady-state simulated and measured exhaust manifold temperatures for both the exhaust valve brake and the combination of the exhaust valve brake and compression release brake.

The figures show a good agreement between measurements and simulations for steady-state engine braking over a large range of engine speeds. The deviations between measured and simulated mass flow rate and engine torque are within 5% of the simulated values. For engine torque and aspirated air mass flow rate, these results confirm that they are mainly dependent upon the pressure difference across the engine as is assumed in the engine brake models.

2.2. Simulation results for transient operation – activation of engine brake

In order to validate the dynamic behavior of the model, transient measurements have been performed on the engine. During these tests, the engine is operating normally at a predefined steady-state operating point before the engine brake, i.e. the combination of the exhaust valve brake and compression release brake is activated. In these experiments, first the fuel flow is stopped. After one second, both the compression release brake and the exhaust valve brake are activated. For the exhaust valve brake it is assumed that the closing of the exhaust valve requires 0.3 seconds. The compression release brake is activated instantaneously. The engine speed in the final steady state is also predefined. In table 2 an overview of the initial and final steady states of four of the performed transient tests, for which the results will be shown, is presented. In practice, transients A and D can be seen as activating the engine brake to maintain a constant velocity when going down a hill. Transients B and C can be considered as activating the engine brake to decrease vehicle speed, for example when stopping for a traffic light.

TABLE 2

Overview of transients – activation of engine brake

Transient	**Initial state**	**Final state**
A	100% load ; 2100 rpm	2100 rpm
B	100% load ; 2100 rpm	1500 rpm
C	70% load; 1500 rpm	2100 rpm
D	70% load; 1500 rpm	1500 rpm

The actuation signals to the fuel injection system, exhaust valve brake, compression release brake and the engine speed are used as input variables in the DYNAMO simulations. It has to be noted here, that the actuator signals to the compression release brake and exhaust valve brake are not actually measured but have been numerically constructed. The input signal to the exhaust valve brake directly describes the opening percentage of the exhaust throttle. Figure 6 shows the used inputs for transient A.

As can be seen in figure 6, the engine speed drops as soon as the fuel flow is stopped. However, the dynamometer connected to the crankshaft has a set-point of 2100 rpm. During the time period between the stopping of the fuel flow and the activation of the engine brake, the engine is driven by the dynamometer and the engine speed starts to increase. The activation of the engine brake results in a rapid decrease in engine speed before the dynamometer comes into play and speeds-up the engine to the desired final steady-state engine speed of 2100 rpm.

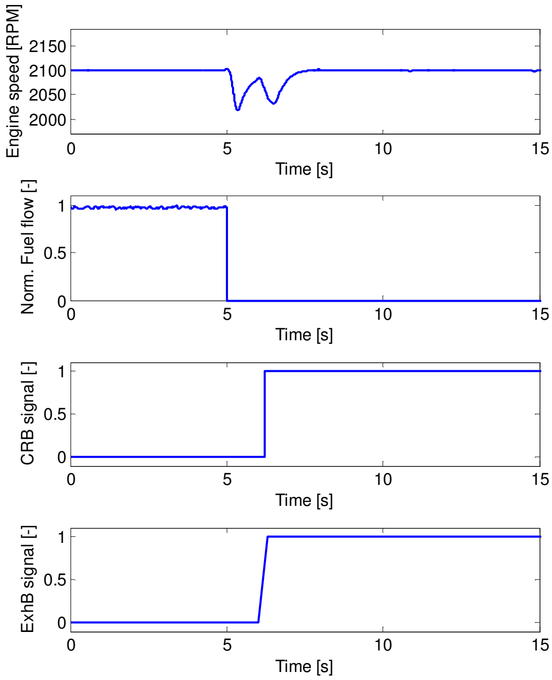

Figure 6
Input signals for transient simulations for transient A.

Figures 7 to 10 show the results for respectively engine torque, aspirated air mass flow rate, turbine-out temperature and turbo speed. It has to be noted here, that the deviation present in the initial state for transients A and B are caused by inaccuracies in the original DYNAMO engine model and are not the result of the implemented engine brake models. The origin of the deviations in the initial conditions is an incorrect simulation of the exhaust manifold temperature, i.e. the temperature rise across the engine. In all figures the instantaneous events of stopping the fuel flow rate and activating the engine brake can be clearly seen as discontinuities in both the measurements and simulation results. As can be concluded from these figures, the predicted dynamic behavior is in good agreement with the measurements. The largest deviations are present in the turbo speed and exhaust manifold temperatures. The turbo speeds drops too slowly after the fuel flow has stopped. This is most probably the result of a defined turbo inertia which is too high in the original model. This also causes the mass flow rates to remain too high during the time period between stopping of the fuel flow and activation of the engine brake. However, after the activation of the engine brake, the turbo speeds and mass flow rates are in good agreement with the measurements. This is an important conclusion for controller development for, amongst others, automatic gear switching algorithm development. If the engine brake is deactivated, the turbo speed is a decisive factor in the engine response. The good agreement in mass flow rates and also in engine torque indicate that the manifold pressures are correctly simulated (not shown).

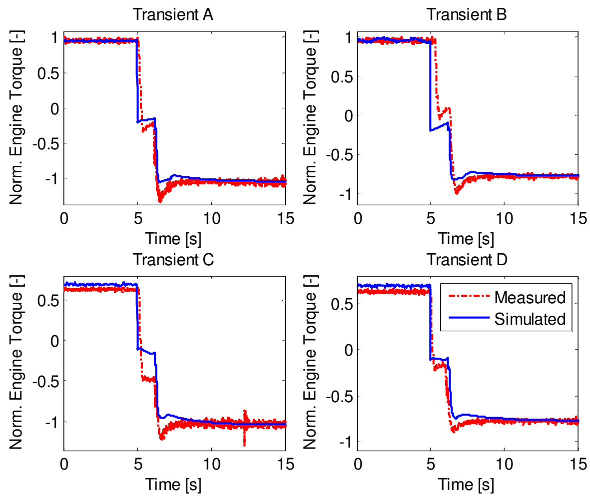

Figure 7
Comparison transient simulated and measured engine torque for the four selected transients and the combination of the exhaust valve brake and compression release brake.

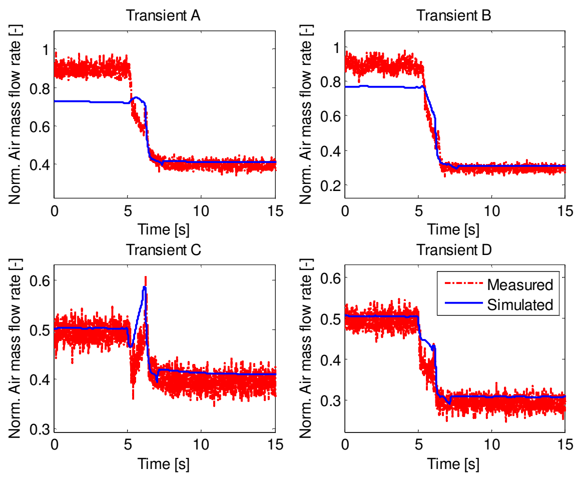

Figure 8
Comparison transient simulated and measured aspirated air mass flow rate for the four selected transients.

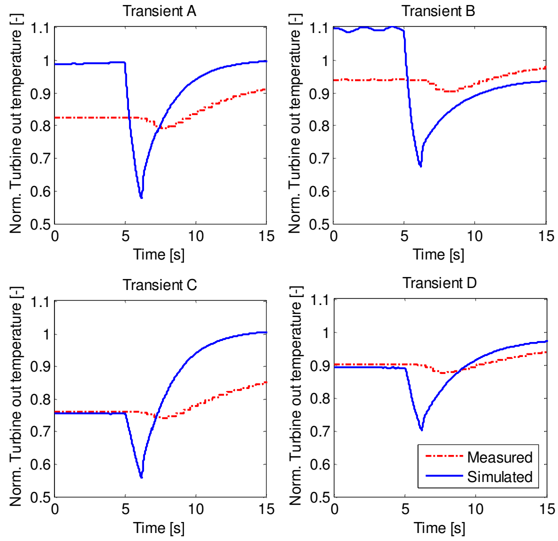

Figure 9
Comparison transient simulated and measured exhaust gas temperature for the four selected transients.

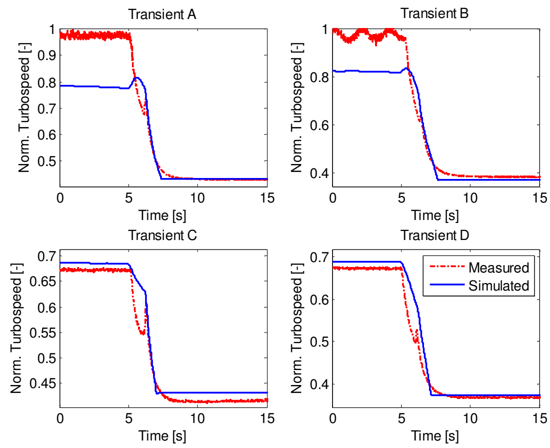

Figure 10
Comparison transient simulated and turbospeed for the four selected transients.

The turbine-out temperatures in figure 9 show the largest deviations between simulations and measurements when the exhaust valve brake is activated. This is only partially caused by the large response time of the used thermocouples in the exhaust system. The final steady states (not shown) show an acceptable accuracy with deviations < ~5% of the measured value. The used models for the determination of the exhaust manifold temperature and turbine-out temperature therefore need to be extended with a more physically based model to capture all of the influences and give a better approximation of the dynamic behavior.

3 MODEL APPLICATION

The good agreement between measured and simulated mass flow rates and the reasonable agreement in exhaust gas temperature make the developed model a valuable tool for exhaust gas aftertreatment controller development. The accurate simulation of the engine torque during the activation of the engine brake is beneficial for power train controller development. This has also been one of the first application of the developed engine model. More specifically, the model has been used to examine the possibility of using the exhaust valve brake to facilitate automatic gear switching.

The dynamic model has been used as a tool to verify that by activating the exhaust valve brake during gear switching the gear switching process can occur faster. The activation of the exhaust valve brake results in a more rapid decrease in engine speed. With the aid of the model a control algorithm (of the open loop type, i.e. feedforward) has been developed for this, which has subsequently been tested on an actual vehicle, i.e. complete power train.

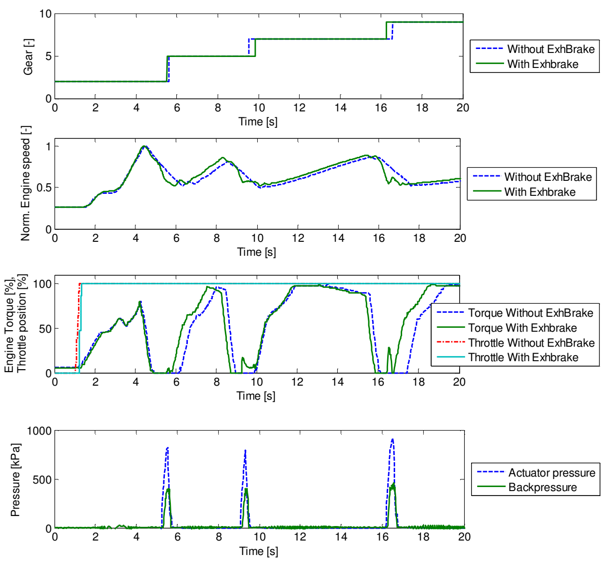

Figure 11
Comparison between automatic gear switching measurements with and without the use of the exhaust valve brake.

Figure 11 shows the measurements for both the situation with and without the application of the exhaust valve brake for up-shifting from 3rd to 9th gear. At every gear switch the exhaust valve brake is activated, which can be seen in the bottom graph which shows the pressure in the exhaust valve brake actuator and the exhaust manifold pressure. Both pressures increase during braking. The accelerator pedal remains at 100% (full load) during the whole shifting sequence. During activation of the exhaust valve, the engine torque and engine speed rapidly decrease. As can be observed from the figure, the decrease in engine speed is much faster when the exhaust valve brake is used. This will result in a much smoother and faster gear shift.

4 CONCLUSIONS AND OUTLOOK

In this study, the implementation of engine brake models in an existing engine simulation tool, DYNAMO, is described. Although the implementation only requires minor adjustments to the current engine model through mostly empirical correlations, the model gives satisfactory results for both steady-state and transient (i.e. activation of engine brake) engine behavior during engine braking. The implemented models are based on steady-state measurement data. During braking the engine operates outside of the operating range for normal engine operation, e.g. higher backpressures. As a consequence, additional engine measurements need to be performed. However, no quantities are required that are not already measured in case of normal engine operation. The required data is therefore relatively easy to obtain.

The predicted mass flow rates are greatly dependent on the turbo model (i.e. compressor and turbine). During engine braking, the turbine operates outside of its normal operating range for which data is available through the turbine manufacturers operating map. A very pragmatic model for simulating the mass flow rate, turbine-out temperature and turbo speed is currently implemented as a first attempt. Results however show that already satisfactory agreement on mass flow rates and turbo speed are present with the use of these models. Only for the turbine-out temperature an extension of the current model, i.e. a more physically based model, is requested. Deviations present are not only coupled to the turbine model, but mainly to the prediction of the exhaust gas temperature exiting the engine.

The benefits of the model are that it is relatively simple, computationally very efficient (real-time) and it able to calculate quantities during engine braking which are of great significance for exhaust gas aftertreatment system control, i.e. engine mass exhaust gas flow rate and temperature. This is of great importance since in the future exhaust gas aftertreatment system controls will become more and more integrated in the engine control system. The accurate prediction of dynamic engine behavior during engine braking makes the model a valuable tool for controller development. In the present study, the model has been used to examine the possibility to apply the exhaust valve brake during automatic gear switching. With the aid of the model it was verified that the exhaust valve brake can facilitate the gear switching by decreasing the engine speed more rapidly. This has subsequently been checked on an actual vehicle.

The present engine simulation tool, DYNAMO, is a mean value model. This means that in-cylinder phenomena are not explicitly modeled. The temporal resolution is greater than one engine cycle. Compression release braking and, most importantly, combustion and emission formation, are occurring on a much smaller time scale. Therefore, current research focuses on extending DYNAMO with a real-time in-cylinder model for combustion and emission formation evaluation. This will also include the incorporation of more advanced combustion concepts, such as Homogeneous Charge Compression Ignition Combustion.

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