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Time to surge concept and surge control for acceleration performance

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Abstract: Surge is a dangerous instability that can occur in compressors. It is avoided using a valve that reduces the compressor pressure. The control of this valve is important for the compressor safety but it also has a direct influence on the acceleration performance. Compressor surge control is investigated by first studying the surge phenomenon in detail. Experimental data from a dynamic compressor flow test bench and surge cycles measured on an engine is used to tune and validate a model capable of describing surge. A concept named *time to surge* is introduced and a sensitivity analysis is performed to isolate the important characteristics that influence surge transients in an engine. It is pointed out that the controller clearly benefits from a feed-forward term due to the small time frames associated with the transition to surge. In the next step this knowledge is used in the design of a novel surge controller. This surge controller is then compared to two other controllers and it is shown that it avoids surge and improves the acceleration performance by delivering both higher engine torque and turbo shaft speed after a gear change.

Keywords: automobile powertrains, engine control, compressor, turbo, system modeling

1. INTRODUCTION

From being used exclusively in sports and performance cars, turbochargers are now common even in ordinary family cars. Ever increasing fuel prices and focus on the environment have moved the automotive industry from using large bigbore engines to using the advantages of downsized and turbocharged engines instead (Guzzella et al., 2000).

An important component in the turbocharger is the compressor as it influences the engine power and thus acceleration performance. In the compressor a dangerous instability phenomenon called surge can occur. It is a small time frame phenomenon caused by the relatively slow dynamics of the turbo shaft. If an automotive turbocharger is driven in deep surge cycles for too long the turbo charger will break down. An effective fail safe method for avoiding surge using a surge valve is implemented in today's production cars. However, in order to ensure safety in all cases, this method wastes much of the valuable pressurized air and turbo shaft speed. Surge control is most important during a gear change since the compressor pressure directly influences the torque available when the new gear is engaged.

The approach is to use experimental surge data from both a compressor surge test stand and an engine test bench to build and validate a compressor and engine model that can describe surge. Then a sensitivity analysis is performed to investigate what properties are most important when considering the time to reach surge and then it is studied what can be done to increase the engine acceleration performance through surge valve control. Based on the knowledge gained from the surge investigation a novel control structure for surge control is developed. One important part is the feedforward term, motivated by the short time frames associated with surge control. The developed controller is then compared to two other controllers.

2. MODELING

To investigate surge and different control strategies a surge capable compressor model has been developed. It is implemented as part of a Mean Value Engine Model (MVEM) (Hendricks, 1989) of a turbocharged Spark Ignited (SI) engine in Simulink. The model structure is component based using restrictions (air filter, compressor, intercooler, throttle, engine, turbine, exhaust system) interconnected with control volumes, and further extended with surge and wastegate valves. Component based MVEM of turbocharged engines is outlined in Eriksson et al. (2002) and Eriksson (2007) while the implemented model is developed and validated in Andersson (2005). The generel structure of the model is shown in figure 1 and the states and notation used is given in the appendix. There are in all 14 states; six pairs of control volume pressures and temperatures, shaft speed and, for the surge capability, compressor mass flow.

2.1 Surge region modeling

The original MVEM has been extended to handle surge (Bergström and Leufven, 2007; Wiklund and Forssman, 2005). The extension utilizes the model by Moore-Greitzer (Greitzer, 1981; Eriksson, 2007) and introduces a compressor mass flow state, given by

$$\frac{dW_c}{dt} = \frac{\pi D_c^2}{4L_c} \cdot (\hat{p}_c - p_c) \tag{1}$$

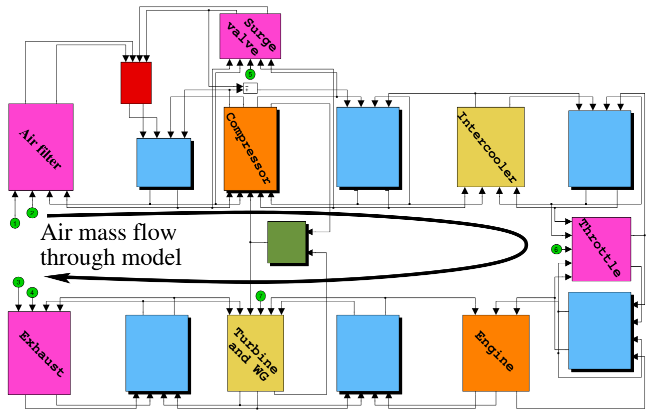

Fig. 1. Engine model with a surge capable compressor model. The mass flow path starts in the air filter (upper left corner) and continues through the compressor, intercooler, throttle, intake manifold, cylinder(s), exhaust manifold, turbine to the exhaust system (lower left corner). These different components are interconnected with control volumes. The turbocharger shaft is seen between the compressor and the turbine. The model also contains a wastegate and a surge valve.

Here $\hat{p}_c = \hat{p}_c(T_{af}, p_{af}, p_c, \omega_{tc}, W_c)$ describes the compressor pressure build up using the parameterized Ellipse model developed in Bergström and Leufven (2007) which handles both forward as well as backward (surging) compressor mass flows. Methods for determining and tuning the parameters of the surge capable compressor model are also described in Bergström and Leufven (2007).

2.2 Surge region validation

To ensure that the model captures real surge phenomena the implemented compressor model is validated against measured data. Validation data from both a separate compressor surge test stand and surge measurements on a real engine are used. The surge test stand validation is shown in figure 2. It is shown that the surge cycle time and compressor pressure ratio behavior during surge is well described by the model. The full MVEM is validated in figure 3 with respect to compressor pressure ratio and turbo shaft speed variations for a rapid throttle closing transient. The validations show that the model gives good description of surge properties, which will be important for the control development, in particular: where surge starts, cycle times, pressure fluctuations and also turbo shaft speed changes.

3. TIME TO SURGE – TTS

As a first step in the controller design the requirements are investigated. The surge phenomenon is very fast and thus puts stringent requirements on the necessary controller reaction time. To facilitate an analysis of this requirement a concept called *Time To Surge* (TTS) is introduced. TTS describes how long time it takes for the compressor to enter surge for every operating point in the compressor map, thus showing the needed response time of the control system. Measured compressor pressure from two fast

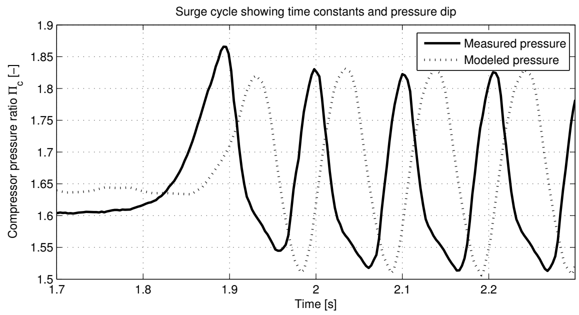

Fig. 2. Measured (solid) and modeled (dashed) surge cycle pressures. The figure shows that the compressor model captures both the surge cycle time as well as pressure dynamics during surge as the measurements.

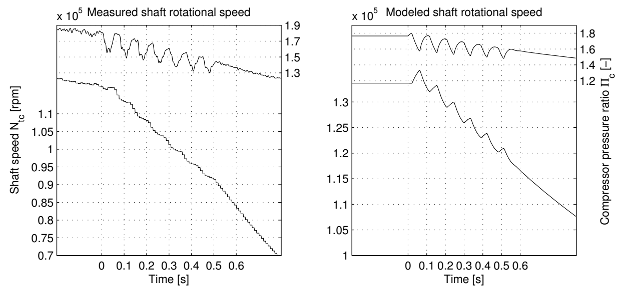

Fig. 3. Turbo shaft speed and pressure variations for a rapid throttle closing transient. Left–measured, right–modeled for an operating point close to measured data. It is seen that the turbo shaft speed does not change much from the surge initiation until the compressor has entered the first surge cycle, which motivates the assumption that the shaft dynamics can be neglected when TTS is investigated.

throttle closings are plotted together with the throttle area reference signal in figure 4. It is seen that the throttle closing is not instantaneous and that the TTS is slightly larger than 0.1s.

The most common cause for surge in an automotive turbocharged engine is a fast throttle closing, e.g. associated with a gear change. Therefore the calculation of TTS is based on the following scenario: the compressor starts in an initial operating point, then there is a sudden drop in throttle mass flow which will lead the compressor into surge. The compressor is said to enter surge when the compressor mass flow equals the surge mass flow for the current shaft speed. The surge mass flow is given by the compressor map and the line formed by the surge mass flows for all shaft speeds is called the surge line, SL.

TTS depends on many system properties and to study what is most important a sequence of increasingly complex systems are studied in the following sections.

3.1 System 1: Instantaneously zero throttle mass flow

In the first system both the intercooler restriction and temperature differences/dynamics are neglected. Furthermore the turbo shaft speed is assumed constant, due to the small time frames of the surge phenomenon compared to the turbo shaft dynamics (see figure 3). The conditions in the air filter control volume are also kept constant

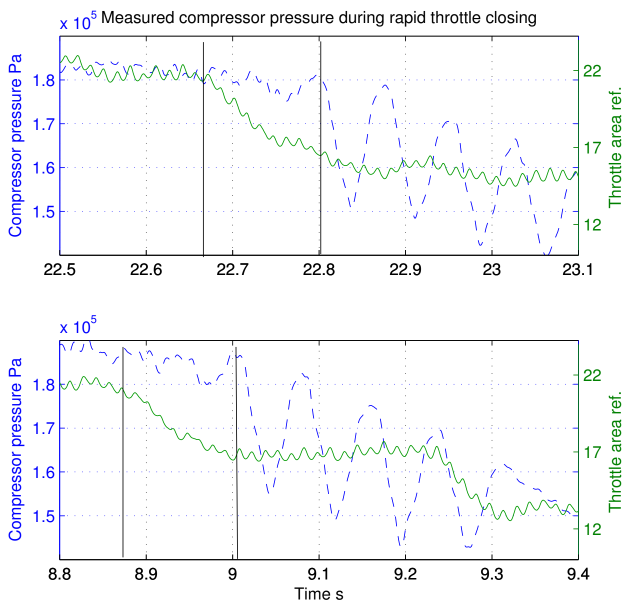

Fig. 4. Pressure measured after the compressor (dashed) from two fast throttle closings are plotted together with the throttle area reference signal (solid). Lines are also shown to emphasize where the step is applied and where the compressor starts surging. For both these finite closing speed transients TTS is about 0.1s. The measured signals are filtered offline using a low pass filter with zero phase shift.

(p_{af}, T_{af}). Finally it is assumed that the throttle mass flow immediately goes to zero in the transient. The differential equations for this simple system now become

$$\frac{dp}{dt} = \frac{RT}{V}(W_c - W_{th}) \tag{2a}$$

$$\frac{dW_c}{dt} = \frac{\pi D_c^2}{4L_c}(\hat{p}_c - p_c) \tag{2b}$$

$$W_{th} = 0 \qquad \text{for } t > t_{init} \tag{2c}$$

where the assumption of an isothermal model, Hendricks (2001), for the lumped control volume is used. A normal temperature increase of 80K over the compressor is assumed as well as a constant control volume temperature before the compressor of 290K. The resulting TTS from this approach is shown in figure 5. The figure shows a worst case, smallest time, originating from a throttle mass flow instantaneously going from $W_{c,init}$ to 0. Even for large mass flows far away from the SL, the time is rather small and a feedback control system thus has to be fast and combined with fast actuators to be able to avoid surge.

3.2 System 2: Dynamic throttle behavior

In the next step the instantaneous stop in throttle mass flow is extended with a first order system for the throttle mass flow. This models the finite response time of a throttle system. The system is now described by (2) but where equation (2c) is exchanged for

$$\begin{aligned} \frac{dW_{th}}{dt} &= -\tau_{th}\, W_{th} + u_{th} \\ u_{th} &= \begin{cases} W_{th,init} & t < t_{init} \\ 0 & t \geq t_{init} \end{cases} \end{aligned} \tag{3}$$

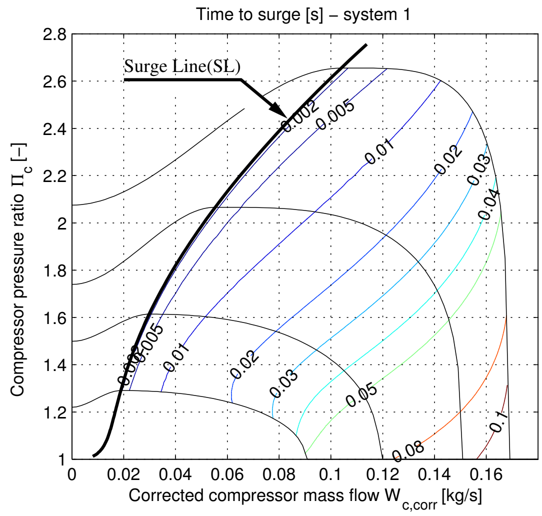

Fig. 5. TTS for system 1. The approach assumes an instantaneous stop in throttle mass flow and underestimates the TTS, showing a worst case scenario.

Since the throttle dynamics is not known exactly, two different first order system time constants, τ_{th}, are used. The results from the calculations for $\tau_{th} = 0.025$s are shown in figure 3.2 Compared to system 1, figure 5, the TTS is larger for every operating point, which is expected. The differences between the two time constants, τ_{th}, are obvious. The behavior is similar but the slowly closing throttle has a much larger TTS. This shows that the throttle closing time constant has a significant impact on TTS.

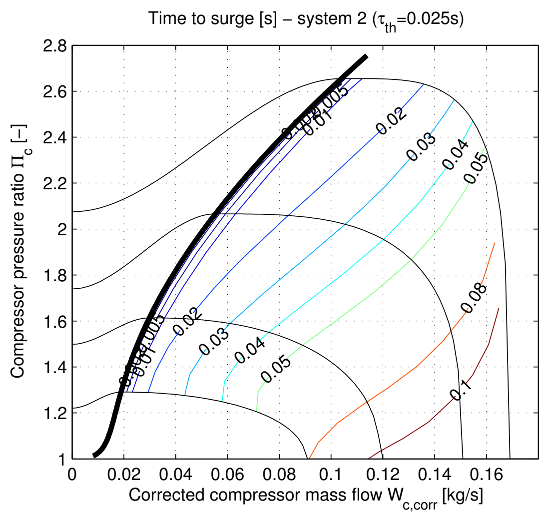

Fig. 6. TTS for system 2. Shown is the TTS map for a throttle time constant of $\tau_{th} = 0.025$s. Larger constants have the same qualitative behavior but differs in the numerical values

3.3 System 3: Temperature dynamics in intermediate control volumes

The next extension is to add the temperature dynamics, i.e. using the adiabatic model (Hendricks, 2001), of the lumped control volume. Compressor upstream conditions

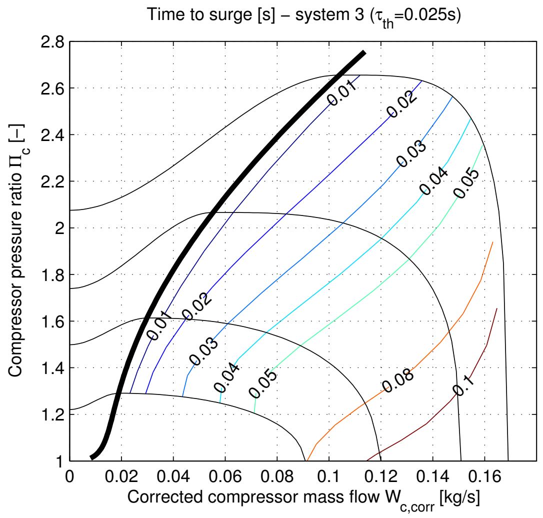

Fig. 7. TTS for Approach 3 for the throttle time constant $\tau_{th} = \frac{1}{40} = 0.025$s. A comparison with figure 3.2 shows that the introduction of control volume temperature dynamics as a state has only very minor effect.

are still kept constant for this system, as is the turbo shaft speed. The states for this system are $W_c, p_{cv}, T_{cv}, W_{th}$. If TTS for this system, shown in figure 3.3, is compared to figure 3.2 it is easy to see that the extra temperature dynamics introduced has only minor effect. Also the qualitative behavior of the TTS is preserved. The temperature dynamics can thus be neglected.

3.4 System 4: Complete 14 states MVEM

As a final investigation the full 14 states MVEM is used to see if there are other effects that have a major impact on TTS. In the previously investigated systems three simplifying assumptions have been made; constant inlet conditions, lumped control volume and constant shaft speed. These will be addressed here. The first by adding an air filter restriction and control volume, which allows the compressor inlet conditions to vary. In particular the inlet pressure will be slightly lowered and mass flow dependent. The second by introducing the intercooler restriction and an extra control volume between compressor and throttle. The third by the introduction of turbo shaft dynamics, allowing the compressor speed to vary (see figure 3).

For the complete model with $\tau_{th} = 0.025$s the TTS-map is shown in figure 8. TTS for $\tau_{th} = 0.1$s is as expected larger. Even for this multi state system it is obvious that the throttle closing speed, essentially τ_{th}, has a major effect on the time it takes before the compressor enters surge.

The differences between the complete MVEM model and the three simple approaches presented earlier are small. TTS is around $10 - 15\%$ larger throughout the compressor map compared to system 2 and 3. Using any of the simpler approaches to calculate TTS would thus give a control system some margin.

3.5 Conclusions of the TTS-investigation

Throughout the investigation a clear trend in parameter sensitivity can be seen. The single most important factor

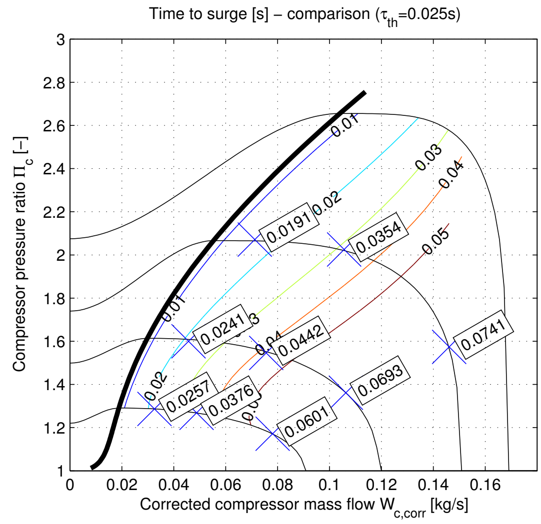

Fig. 8. TTS using the full 14 states model, $\tau_{th} = 0.025$s. Isolines in the background are from system 3 with the same τ_{th}.

is how fast the throttle closes, due to the close connection between effective throttle area and throttle mass flow. In summary, these are the main results from the investigation

- The most important parameter is the throttle time constant τ_{th}.
- Shaft speed variations can be neglected with a good result because of the relatively slow dynamics.
- Temperature dynamics can be neglected unless very small control volumes are being used.

As can be seen in the small differences between the full MVEM model and the three simpler approaches, even a less complex method for determining TTS gives satisfactory performance.

4. CONSTRUCTION OF A SURGE CONTROL SYSTEM

There are two actuators available for surge control, wastegate and surge valve. The wastegate valve effects the driving torque from the turbine and the surge valve releases pressurized air after to before the compressor. The wastegate has almost no effect due to the slow dynamics of the turbine shaft. The main actuator is therefore the surge valve.

4.1 Feedforward or feedback control

The discussion so far supports the claim that a feedforward control system is needed. The largest TTS for the throttle closings studied are all less than around 0.2s. If only real operating points are considered this maximum time is reduced even further and for normal operating points it is closer to 0.05s. The necessary reaction times of a control system are thus even faster. If feedback control is to be used, the reaction time of the control system can be divided into three parts: control system sample time, feedback sensor response time and actuator response time. Reducing any of these parts almost always implies a large increase in production costs.

One way to enhance the possibilities to avoid surge is to add a surge valve controlled not only by the Electronic Control Unit (ECU) but also controlled by a pressure difference. To add even more safety to the system the control system could be constructed to close the surge valve and leave the opening to a pressure difference system, see Bergström and Leufven (2007).

4.2 Surge valve characteristic

The surge valve of the MVEM is implemented as

$$\begin{aligned} T &= T_c \\ W_{sv} &= A_{sv} \frac{p_c}{\sqrt{T_c R}} \sqrt{\frac{2\gamma}{\gamma - 1} \left(p_r^{\frac{2}{\gamma}} - p_r^{\frac{\gamma+1}{\gamma}} \right)} \end{aligned} \tag{4}$$

with $p_r = \max\left(\frac{p_{af}}{p_c}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right)$. A surge valve has a maximum effective opening area and the surge valve maximum mass flow is shown in figure 9 for two different effective areas. Only surge control for cases where the surge valve can recycle the compressor surge mass flow are considered, thus meaning shaft speeds where

$$W_{sv,max} > W_{c,surge,noncorr} \tag{5}$$

effectively giving an upper limit for the shaft speeds to be considered. $W_{c,surge,noncorr}$ is the real mass flow, i.e. not corrected, at the surge line for the given shaft speed. Due to the fact that the maximum mass flow through the surge valve is also inversely proportional to the square root of the compressor control volume temperature, the possible control region is further reduced. The air filter control volume temperature is also assumed to vary little from the ambient temperature due to the heated recycled air, effectively meaning that the recycled air mass is considered small compared to the control volume mass.

4.3 Formulation of control goal

To retain high turbo shaft speeds, during e.g. gear changes, the goal for the control system is to keep the operating point on the highest possible shaft speed. The spool up time for the turbo is then reduced when the new gear is engaged and the demand for engine torque rises again. Retaining turbo shaft speed means keeping the driving torque from the turbine high and keeping the torque consumed by the compressor low. This can be seen in the turbo shaft dynamics

$$\begin{aligned} \frac{d(N_{tc}\frac{\pi}{30})}{dt} &= \frac{d\omega_{tc}}{dt} = \frac{1}{J_{tc}}(Tq_t - Tq_c - Tq_{tc,f}) \\ Tq_c &= \frac{30}{\pi}\frac{(T_c - T_{af}) \cdot c_p \cdot W_c}{N_{tc}} \end{aligned} \tag{6}$$

The consumed torque is proportional to both the temperature difference over the compressor as well as the compressor mass flow. Further it is inversely proportional to the shaft speed. To be able to find the most interesting compressor operating point consumed compressor torque isolines are shown in figure 9. It can be seen that for every shaft speed the least amount of torque is consumed for operating points at the surge line.

4.4 Control algorithm

The control algorithm for the feedforward controller is as follows: when the throttle closing is commanded, calculate the time it takes for the operating point to reach the surge line, TTS. When the surge line (SL) is reached open the surge valve directly to a mass flow according to the following

$$\begin{aligned} W_{sv} &= 0 & t_{init} < t < t_{init} + t_{TTS} \\ W_{sv} &= W_{th} - W_{c,surge} & t = t_{init} + t_{TTS} \end{aligned} \tag{7}$$

meaning that the surge valve is closed for the first TTS seconds and thereafter opened to stabilize W_c at the SL. When this is done, engage a PI(D)-controller that follows the SL.

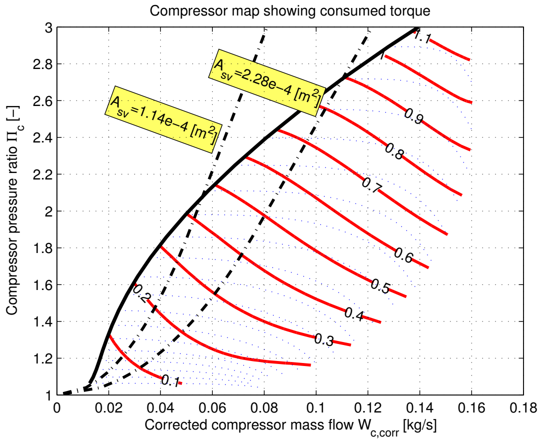

Fig. 9. Consumed compressor torque shown with solid line. Dashed line shows compressor speed line. Seen is that for every speed line the minimum consumed torque is to be found at the surge line (or left of this). Surge valve characteristics are also shown for two different effective areas, $1.14 \cdot 10^{-4}\text{m}^2$ and $2.28 \cdot 10^{-4}\text{m}^2$, from Wiklund and Forssman (2005)

5. CONTROLLER EVALUATION

A test scenario is used for evaluation. It consists of a one second long gear change where the throttle area is dropped to a minimum. The time constant used for the throttle is $\tau_{th} = 0.025$s giving a fast system to control. The controller described in the previous section is compared to two other controllers, one using a simple blow-off-technique that keeps the surge and waste gate valve fully open and one using feedback from mass flows and utilizing a PI-controller. A first order system surge valve, having a time constant of $\tau_{sv} = 0.02$s was used for all three controllers as well as a maximum effective surge valve area of $1.14 \cdot 10^{-4}\text{m}^2$, see Wiklund and Forssman (2005). The effect from more realistic cases having pulse width modulated control signals and time delays in the system is studied in Bergström and Leufven (2007).

The results from the gear change test case is shown in figure 10. It is seen that the feedforward controller has a up to 8% higher engine output torque and up to 20% higher turbo shaft speed. The difference is, as expected, decreasing slowly after the gear change and in the long run the operating points of all three systems will converge.

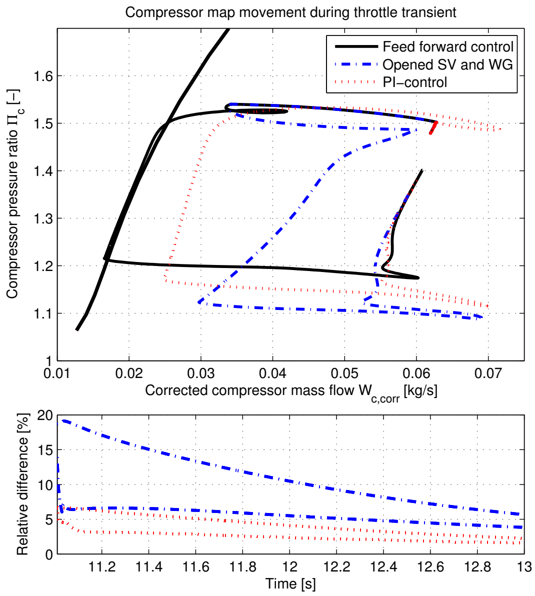

Fig. 10. The figure shows differenct compressor map movements in corrected quantities for the three different controllers described in section 5. The feedforward controller map movement is closest to the SL and the corresponding relative increase in both shaft speed as well as engine torque after the gear change is seen. The shaft speed difference is at most 20% between the feedforward controller and the simple "blow open"-controller.

6. CONCLUSIONS

A measurement, called Time To Surge (TTS), of how long time it takes for a compressor to enter surge after a fast throttle closing is proposed. Different methods, of increasing complexity, for determining TTS are compared through the usage of a validated surge capable mean value engine model, having 14 states. The MVEM model is parameterized and validated using experimental data. The TTS parameter sensitivity is studied and TTS is found to depend mainly on throttle closing speed. There are no significant changes when taking the temperature or turbo shaft dynamics into account. A system that uses only three states (p_c, W_c, W_{th}) and an isothermal assumption gives a good description and has only a maximum deviation of 10% compared to the complete MVEM model. The largest TTS for normal compressor operating points is found to be around 0.1s which gives a strong motive for a feedforward loop in the control system.

An investigation is conducted to find the optimal compressor map movement that will maintain as high turbo shaft speeds as possible during rapid throttle transients. The investigation shows that, for every shaft speed, the path of least consumed torque coincides with the surge line.

A throttle transient during a gear change is used as a test case in a simulation setup. A new controller based on feedforward and the TTS concept is compared to two other controllers; a fast control system using feedback and a simple "blow-open"-controller. Comparisons show the gains with a feedforward system. The increase in shaft speed and engine net torque when the next gear is engaged is found to be 20% and 8% respectively compared to the simplest "blow-open"-controller.

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Appendix A. NOMENCLATURE

State	Description	Unit
p_{af}	Air filter control volume pressure	Pa
T_{af}	Air filter control volume temperature	K
p_c	Compressor control volume pressure	Pa
T_c	Compressor control volume temperature	K
p_{ic}	Intercooler control volume pressure	Pa
T_{ic}	Intercooler control volume temperature	K
p_{im}	Intake manifold control volume pressure	Pa
T_{im}	Intake manifold control volume temperature	K
p_{em}	Exhaust manifold control volume pressure	Pa
T_{em}	Exhaust manifold control volume temperature	K
p_{es}	Exhaust system control volume pressure	Pa
T_{es}	Exhaust system control volume temperature	K
N_{tc}	Turbo shaft speed	$\frac{1}{min}$
W_c	Compressor mass flow	$\frac{kg}{s}$
Symbol	**Description**	**Unit**
V	Volume	m^3
CV	Control Volume subscript	
R	Specific gas constant	$\frac{J}{kgK}$
D_c	Compressor diameter	m
L_c	Compressor duct length	m
η_c	Compressor efficiency	–
γ	Ratio of specific heats	–
Π_c	Compressor pressure ratio	–
W_{th}	Throttle mass flow	$\frac{kg}{s}$
τ	1st order system time constant	
$init$	Initial condition subscript	
Tq_c	Compressor torque	Nm