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Fully Variable Intake Valve Train – Advanced Air Management for Improved Dynamic Performance of Large Bore Gas Engines

10 - Latest Engine Component Developments - Turbochargers & Air-/Exhaust Management

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ABSTRACT

As the share of renewable sources in the electricity mix grows, flexible power generation units are increasingly required to respond to the highly fluctuating availability of renewable energy. Modern reciprocating engine power plants are able to ramp up electricity generation within a short time period. Gas engines in particular provide power (electricity, heat) at high efficiency while complying with strict emission legislation. Given the need for grid stability, the requirements for transient operation will continue to increase and must be addressed in gas engine development with improved charging and combustion concepts as well as control strategies. Compliance with emission limits has to be guaranteed.

A variable valve timing system on the engine intake (ABB's VCM®) for power control has been shown to improve engine efficiency and allow for increased power density. Additionally, it enhances flexibility under ambient conditions and fuel quality in steady-state operation. Furthermore, improved transient response can be expected in engines equipped with a variable intake valve train.

This paper describes investigations on a lean burn gas engine equipped with a variable valve train on the intake side and demonstrates how transient engine performance can be improved in comparison to a conventional state-of-the-art application with constant Miller timing. The transient performance and control strategies are assessed for power generation applications in grid parallel operation as well as in island grid operation.

Following the LEC's transient development methodology, simulations are carried out using a detailed 0D/1D model of the corresponding multi cylinder engine (MCE) to pre-assess the transient performance potential. The basis for model calibration is provided by a large number of steady-state measurement points from a single cylinder research engine. In the next step, a fast calculating 0D gas exchange model derived from the detailed model is used to set up and calibrate controller structures for power, speed and emissions control using a variable intake valve train. Testing of the concepts developed using simulation is conducted on a single cylinder test bed that allows transient operation. Multi cylinder engine behavior is simulated on the SCE using a Hardware-in-the-Loop approach. This provides insight into the real limitations (knocking, misfire) and real emissions during transient engine operation.

1 INTRODUCTION

The stabilization of the power grid is a key challenge as the share of renewable energy sources will further increase in future. The grid operators and administrators need to purchase balancing power, which is used to quickly restore the supply-demand balance in the grid. Balancing power is used to stabilize the active power balance of integrated power systems on short time scales from seconds to hours [1]. In Europe, balancing power is called “control power”, and is divided in three different types: primary control, secondary control, and tertiary control. The time-based specifications of each are shown in Figure 1.

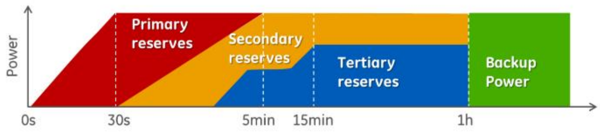

Figure 1. Classification of electricity reserves within ENTSO-E [2]

To ensure that large bore gas engines will furthermore play an important role to stabilize the power grid, the transient performance (ramp up time, block load capability) gets increasingly into focus. This contradicts the request for high electric and thermal efficiencies as well as high operational flexibility (e.g. gas quality) for these engines enabled by Miller timing, high compression ratio and high power density. Furthermore, certain exhaust limits, e.g. maximum permissible values for nitrogen oxide (NO_x) emissions, must be fulfilled also during transient engine operation [3]. This limits the transient performance especially for load ramp up after start with a preheated engine. Exhaust after-treatment devices like SCR catalysts (selective catalytic reduction) need time to heat up before they start to work and hence fuel enrichment as a lever for increased transient performance is restricted.

The use of a variable intake valve train is an option to increase transient engine performance while keeping high electric and thermal efficiency as well as meeting the transient emission requirements. Results with respect to the efficiency potential by introduction of a variable intake valve train were presented in [4], [5], [6] and [7].

This paper presents studies on the transient behavior of a lean burn gas engine equipped with a variable intake valve train. It is shown how the engine control by means of variable intake valve timing differs from a state-of-the-art application with constant Miller timing and which advantages can be achieved. After an overview of possible engine control systems for highly turbo charged lean burn gas engines, the reference engine and the variable valve train system used are described (Chapter 1).

On the basis of simulation models of the multi cylinder engine (MCE), the fundamental differences of the investigated control strategies are presented and the potential for improving the transient performance in grid-parallel and island operation is demonstrated in Chapter 2.

To evaluate the transient performance when using a variable intake valve timing for speed or power control, measurements were performed on a transient capable single cylinder research engine (SCE). For this purpose, the SCE was operated in a hardware-in-the-loop (HiL) mode. Here, the HiL methodology is described first. The structure of the real-time capable full engine model required for HiL operation and the integration of the model into the test bench environment are described. The assumptions and limitations of the HiL methodology are listed. Subsequently, the results of the transient test runs in island operation as well as in grid-parallel operation are presented. A comparison of different control strategies reveals the potential for increasing transient performance and shows the influence on transient NO_x emissions in Chapter 3.

Finally, the implications of using a variable valve train for engine control are discussed and a conclusion is given.

1.1 Gas engine speed and power controls

Modern lean-burn gas engines offer diesel-like power densities at very high levels of thermal efficiency. This became mainly possible with the introduction of the Miller valve timing in combination with high pressure turbocharging; whereas today’s high-end gas engines are equipped with two-stage turbocharging systems.

Gas engines are operated in compliance with a given emission regulation (e.g., TA-Luft). For this reason, speed and/or power is not solely controlled by the admitted quantity of gas fuel. In order to be able to control the engine at a desired air/fuel ratio level, the turbocharging system has to be designed such – mainly via the selection of the turbine specification(s) – that a surplus of charge air pressure can be provided at all time. Designated control elements adjust the pressure to the load and air/fuel ratio corresponding value.

In [4], common control elements of modern gas engines are described. They are categorized in elements that act on the hot side (exhaust waste gate, variable turbine geometry) and on the cold side of the engine (compressor recirculation, throttle, variable valve train). The elements on the exhaust side have the advantage of allowing for better fuel efficiency (reduced gas exchange losses), while the air/mixture-side elements provide better controllability and faster transient responses (reduced dead time). However, it was also demonstrated that control via a variable intake valve train allows for exceeding the efficiency benefit of the exhaust control elements as well as the transient responsiveness of throttle and bypass elements on the compressor side.

In the present paper, the scope of investigated control elements is limited to the reference case of compressor bypass control vs. the control with a variable inlet valve train system, i.e. ABB's VCM® system. Furthermore, the content is limited to premix gas engines.

During transients the change in engine torque can be further supported by enrichment of the fuel/air mixture and the IVC timing beyond the reference timing (increased volumetric efficiency). However, both too low air-excess ratio as well as too high values of volumetric efficiency are prone to knocking combustion which must be avoided.

Before going to the in-depth investigation in terms of simulation and measurement the main differences as well as the limitations of the two control approaches are visualized based on a basic correlation: by using the formula for the calculation of the brake mean effective pressure (bmep) based on receiver conditions, air-excess ratio $\lambda_{V,tot}$ and the volumetric efficiency λ_R, the maximum immediate load step from idle can be calculated.

$$bmep = \frac{p_{Rec}\lambda_R}{(\lambda_V L_{st}+1)} \frac{H_u \eta_{th}}{R_{Rec} T_{Rec}} \quad (1)$$

Therefore:

$$\frac{bmep_2}{bmep_{Ref}} = \frac{\lambda_{R,2}}{(\lambda_{V,2} L_{st}+1)} \frac{(\lambda_{V,Ref} L_{st}+1)}{\lambda_{R,Ref}} \quad (2)$$

$$\frac{bmep_2}{bmep_{Ref}} \approx\sim \frac{\frac{\Delta\lambda_R}{\lambda_{R,Ref}}+1}{\frac{\Delta\lambda_V}{\lambda_{V,Ref}}+1} \quad (3)$$

The boundary assumption is that the initial reaction of the turbocharging system is slow and an ambient pressure of 1 bar is immediately available in the intake receiver. With additional assumptions on gas properties (NG), engine thermal efficiency and the intake receiver temperature the stated equation can be evaluated for various levels of air excess ratio and – considering the case of variable intake valve timing – volumetric efficiency. The resulting diagram is depicted in Figure 2. As reference values the nominal values of the baseline engine introduced in the subsequent section are considered. The maximum load step from idle is shown as function of the air-excess ratio enrichment from the nominal value. Towards enriched air-excess ratio larger initial load steps can be achieved. For this example, an enrichment by $\Delta\lambda_V$ = 0.4 leads to an increase of the initial load step of roughly 5 %pts. With a much higher impact, the change of volumetric efficiency can substantially increase the load step; with an increment of the volumetric efficiency by $\Delta\lambda_R$ = 0.4 the initial load step is increased by more than 10% points. Of course, a combination of enrichment and de-Millering achieves the highest load step values. However, the region in the upper right of the diagram is limited by knocking, whereas very low air-excess ratios inherently cause high thermal load to the engine and turbocharger turbine. It is also worth noting that the benefit of a variable volumetric efficiency becomes more pronounced, if the reference inlet valve train features an earlier Miller timing.

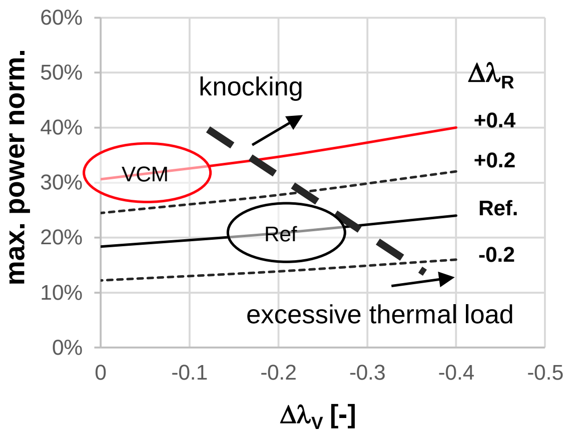

Figure 2. Immediate load step from idle, nominal bmep = 24 bar

In real world applications the described torque step cannot be immediately realized when suddenly demanded. The air path control elements and the fuel gas admission feature an inherent dead time. Nevertheless, and despite the additional well-known fact of the turbo lag, even higher load steps can be realized when allowing the engine speed to drop (island mode operation). If the generator load step of a genset cannot be immediately satisfied by a corresponding engine torque step, the missing energy is then provisionally provided by the engine rotational kinetic energy; the engine speed drops.

Meanwhile the TC system accelerates because of the initial engine torque step. Consequently, with increasing boost pressure the generator torque can be overcome and the engine speed can be brought back on track.

1.2 Description of the baseline engine

The Jenbacher Type 6 gas engine has been part of the product program since 1989. In 2007 the J624 as the world's first 24-cylinder 4MW engine was introduced. Based on this, GE launched the J624-H version as the world's first gas engine with two stage turbo charging in 2010. This engine version is used to point out the advantages of a variable intake valve train for the transient engine performance. The following table shows the major technology concepts of the J624-H version:

Table 1. Technology concepts of the INNIO J624-H engine

	J624-H
Engine process	Four stroke spark ignited with lean A/F mixture
Mixture preparation	Gas mixer upstream of turbocharger
Charging concept	Two stage (4 TC) with two stage mixture cooler
Gas exchange	Single four valve cylinder heads Advanced early Miller timing
Ignition	MORIS high energy ignition system
Combustion concept	Scavenged prechamber with passive prechamber gas valve
Power / speed control	Compressor bypass and throttle valve

The combination of these technologies results in high power density as well as in high electric and thermal efficiency. Furthermore, engine operation at very high altitude, humid and hot ambient conditions and low NO_x emission levels is enabled without power de-rating. The Table 2 lists the key technical data for the natural gas engine version of J624-H which was used as reference baseline.

In the meantime, the development of two stage charged J624 was ongoing and the newer J624-K version was presented offering higher electric and total efficiencies ($\Delta\eta_{Electric}$ = +0.6 %pt., $\Delta\eta_{Total}$ = +0.6%pt.) as well as a higher power output ($\Delta P_{Electric}$ =+0.1MW) [8].

Table 2. Technical data for the INNIO J624-H engine

	J624-H
Bore [mm]	190
Stroke [mm]	220
Displacement per cyl. [dm^3]	6.24
BMEP [bar]	24
Rated speed [rpm]	1500 (50Hz)
Engine power [kW_{el}]	4400
Electric efficiency[1] [%]	46.3 @ MN > 83
Total efficiency[2] [%]	90.3

1 NO_x = 500 mg/m^3 (norm) @ 5 % O2 in exhaust gas, 50 Hz operation

2 Electrical and thermal efficiency, combined heat and power (CHP), 50 Hz operation

1.3 Used variable intake valve control system

The variable intake valvetrain system used for the presented investigations is ABB's VCM®. It is an electrohydraulic valvetrain system that allows stepless variation of valve timing and valve lift [9].

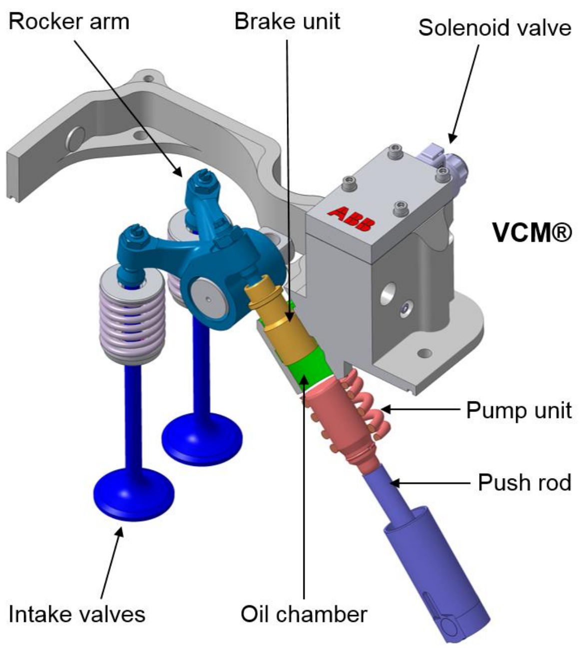

Figure 3. Main components of VCM®

The main components of VCM are a hydraulic pump, oil chambers, a hydraulic brake and a solenoid switch valve (see Figure 1). A detailed description of the operating principle of VCM is found in [5][9][10].

VCM allows to adjust the timing of intake valve closing on a cylinder individual basis and from cycle to cycle. The use of a variable intake valve train permits to transfer power or speed control from the turbocharging system to the cylinder intake process which results in an increased engine efficiency [6]. Furthermore, by suddenly increasing volumetric efficiency, a faster load response can be expected [7].

2 ASSESSMENT OF THE POTENTIAL FOR TRANSIENT ENGINE OPERATION BASED ON SIMULATION

2.1 Model description

For the prediction of the potential of the assessed technologies with respect to transient engine operation, ABB's in-house calculation suite ACTUS was used. Thereby, the simulation models introduced in [7] were used as a basis. The model performance was further enhanced by the introduction of an improved knock model based on [11] and validated on a large number of measured operating points.

In all shown examples, the reference control strategy with compressor bypass and throttle feature the same control margin as the cases with variable intake timing (VCM®). The control margin describes the difference between the charge provided by the compressor (pressure ratio, mass flow) and the charge actually required by the engine to maintain the desired engine operation (power, air-excess ratio). Furthermore, the simulations were performed with an ambient temperature of 30 °C and an altitude of 500m. Lower ambient temperatures and altitudes would result in better transient performance.

For the engine cold start in net parallel mode the thermal inertia of the exhaust manifold and the turbocharger were considered.

2.2 Potential in island grid operation

2.2.1 Maximum block load steps

In this investigation the maximum achievable load steps depending on the initial engine operating point are determined. For this purpose, the load step size was incrementally increased in steps of 10 % leading to increasing engine speed drops. Based on the defined benchmarks, the maximum possible load steps where the engine speed did not fall below the limit were then found by interpolation. As a benchmark, maximum engine speed drops during load steps according to [12], i.e. ISO classes G1 and G3 were used considering maximum engine speed drops of 250 rpm, and 160 rpm respectively.

In the case of VCM control, three control strategies were applied. In case of engine speed drop due to increased generator load the speed control reacts with

Strategy 1)

- Retardation of IVC timing towards optimum filling
- No enrichment of air-excess ratio

Strategy 2)

- Retardation of IVC timing towards optimum filling
- Shift of start of ignition to late timing
- No enrichment of air-excess ratio

Strategy 3)

- Retardation of IVC timing towards optimum filling
- Shift of start of ignition to late timing
- Additional enrichment of air-excess ratio

The results, are depicted in Figure 4. The diagram demonstrates the advantage of VCM control over the reference case. Over the whole load range the achievable load steps turn out to be roughly 10 % higher (absolute referring to 100 % engine power). Towards part load, where the control margin is inherently lower than at the nominal load point, the difference between VCM and the reference becomes even more pronounced.

In general, larger load steps are possible towards higher engine power, because the turbocharging system provides more control margin and its time constant becomes much smaller. On the other, the knock margin is much closer at high cylinder output. Therefore, the IVC shift is more limited to reduced volumetric efficiency and also the amplitude of mixture enrichment needs to be kept smaller to avoid knocking.

Since the knock limit substantially reduces the potential of the VCM solution, simulations with retarded ignition timing were conducted. The shift of the combustion to a later timing leads to an increased margin towards knocking. By applying the control *strategy 2* load steps from low part low points can be carried out with maximum volumetric efficiency without coming close to the knock limit yet. Therefore, the control *strategy 3* can be used and a further increase of the load step size can be realized. Mainly at intermediate loads, where the knock limit is close therefore limiting the IVC timing towards maximum volumetric efficiency, still an increase in load step can be achieved with control *strategy 2*. Towards low load *strategy 2* converges towards the values of *strategy 1*, because the knock margin is not limiting the IVC timing. There, an additional enrichment – *strategy 3* – can still give a relevant boost at idle and very low load points.

Summarizing it can be stated that with a reasonable combination of the three presented strategies, VCM enables increased blockload steps between 15 to 20 % in the relevant load range.

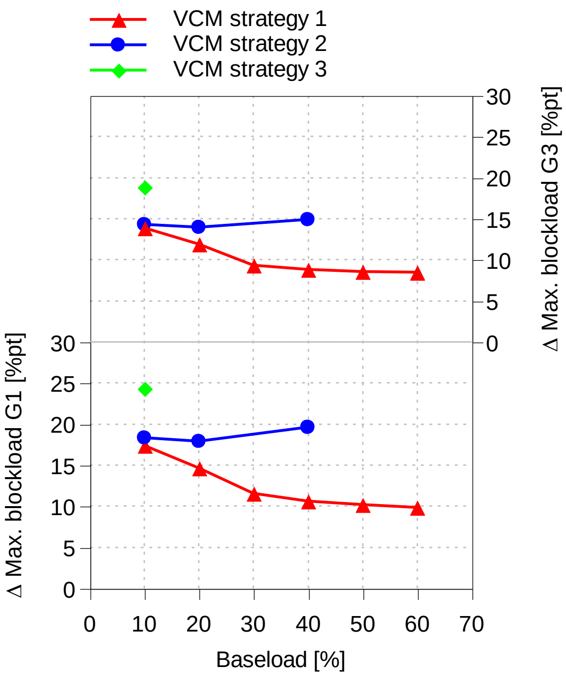

Figure 4. Island grid load acceptance – comparison of control strategies

2.2.2 Exemplary comparison of VCM and CB control during load step

In order to visualize the different behavior of the reference and the VCM control strategy the transient simulation results of some relevant variables of the same load step are shown in Figure 5. The time was scaled to show the difference of the investigated variants relative to the reference configuration (this procedure was applied to all results presented). The IMEP curves at the top of the graph show the required immediate torque step of 30 % at time = 0 %. The faster the engine is able to provide the required torque, the lower the engine speed drop shown at the bottom of the diagram.

The reference strategy reacts to the speed drop by closing of the compressor bypass valve (and if not already open, a complete opening of the throttle valve). In addition, the gas admission valves enriches the air-excess ratio. Despite the very fast reaction of the gas mixer, the enrichment reaches the cylinder with a small delay. Due to this delay, the torque increase cannot happen immediately and the engine speed drop is much more pronounced than in the following example with VCM. It has to be stated that this load step does not meet the ISO G1 requirements by a narrow margin.

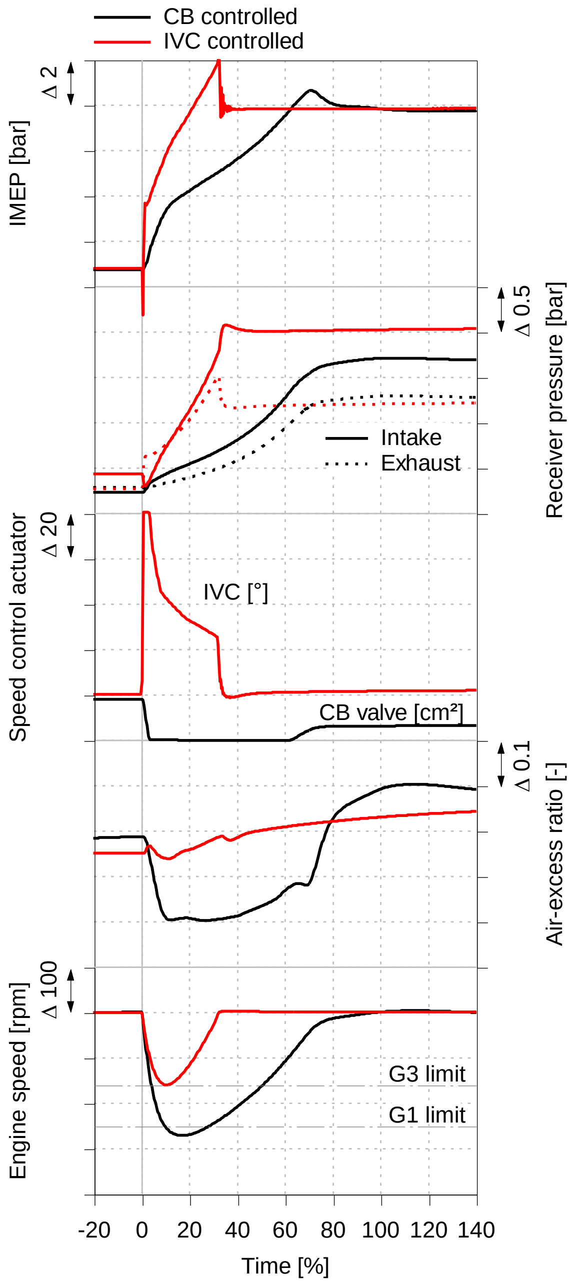

Figure 5. Load pickup in island operation – comparison of CB control and IVC control of engine speed: step from 20 % to 50 % load, VCM strategy 1 only

In contrast to the reference case, the VCM approach with control *strategy 1* does not feature any enrichment during the load step. The air-excess ratio slowly increases to the leaner value at the higher load point at the end of the step. The torque increase is realized by a sudden increase of the volumetric efficiency. For this purpose, the

IVC timing is shifted to a filling optimized timing. Since the IVC timing can be changed from one working cycle to the other, the change in IMEP can be realized almost immediately. The change in volumetric efficiency drastically increases the engine mass flow and hence leads to a sudden increase of the turbine inlet pressure, while the intake pressure is temporarily lowered. The step in turbine pressure causes a very fast ramp-up of the turbocharging system which then leads to high rates of charge air pressure and consequently engine power. However, towards higher IMEP, the maximum IVC timing is limited by the knock border. This can be clearly seen in the fast reduction of the IVC timing as the charge air pressure increases. As soon as the engine speed is recovered, the IVC timing can be switched very fast to the required position. Due to its proximity to the cylinder the engine speed control is easier since almost no dead time has to be considered.

2.3 Potential in grid parallel operation

The power ramp-up rate in grid parallel operation strongly depends on the turbocharging system characteristics (time constant) and the resulting control margin (mainly given by the turbine specification). Small turbine flow areas result in a high control margin and improved transient behavior with the downside of reducing nominal engine efficiency due to increased gas exchange losses. Consequently, the control margin is usually designed as low as possible. However, this can cause very slow power ramp-up especially under preheated engine start conditions. Corresponding simulation results are shown in Figure 6, whereas for the VCM cases the control *strategy 1* was applied. For the CB controlled cases, an enrichment up to a level between 0.2 and 0.3 units in air-excess ratio was enforced.

It can be read from the diagram of engine power over time that the initial power step is identical for a given type of control strategy. In the case of a cold engine start, the control margin becomes very small and leads to very long ramp-up times for the reference control strategy because exhaust components (walls, turbines, …) are heated up first. The exhaust enthalpy available for the turbine is reduced which leads to a reduced turbine power output. The variability in volumetric efficiency with VCM obviously allows for a much larger initial power step and a highly increased engine mass flow, causing a much faster raise of the turbine inlet pressure and therefore a much faster acceleration of the turbocharging system.

In the shown example, the power ramp-up time of the cold engine increases by more than 100 % compared to the hot engine start of the reference case. With VCM the reference ramp-up times can be massively reduced for both hot and cold boundary conditions. Roughly half of the reference time is required. The results are summarized in Table 3.

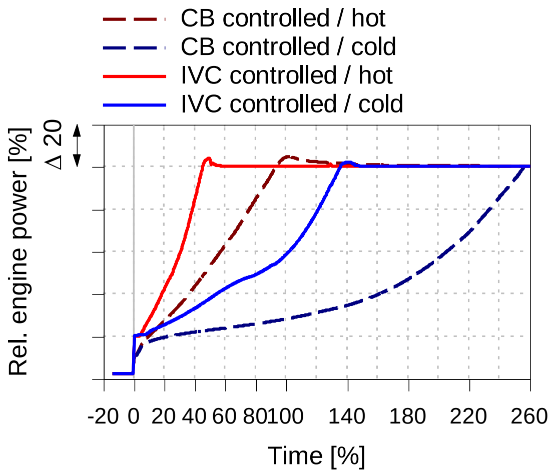

Figure 6. Load ramping capability for hot engine conditions and cold/pre-heated conditions

Table 3. Comparison of grid parallel ramp-up time with CB vs. IVC control

	CB [%]	IVC [%]
cold	260	140
hot	100	50

In the shown investigations, both strategies (CB control and VCM control) always featured the same reference control margin resulting in a benefit in engine efficiency of 0.7 %pts at ambient conditions (cf. [5][6][7]). Nonetheless, a further interesting approach could be to compare the transient behavior when equipping the VCM approach with a highly increased control margin, achieving the same thermal engine efficiency as in the reference case.

3 EVALUATION OF TRANSIENT PERFORMANCE BASED ON SCE MEASUREMENTS

In order to investigate the transient behavior of the combustion, the turbocharger system and different control concepts on a single-cylinder engine test bench, the transient behavior of multi-cylinder engines must be simulated. This requires the application of a hardware-in-the-loop configuration to the SCE. The intake and exhaust pressures of the SCE and MCE differ considerably in transient operation, in particular due to the completely different piping system and the absence of a turbocharger in the SCE. Therefore, the boundary conditions (e.g. boost pressure and exhaust back pressure) of an MCE in transient operation must be set in real time on the SCE test bench. For this reason, the subsystems for pressure control (charge air, exhaust gas, gas supply) and the dynamometer on the test bench must be sufficiently dynamic to meet the transient requirements.

3.1 General setup of the MCE HiL model

Based on a detailed 1D simulation model, a real-time capable 0D gas exchange and powertrain model was generated keeping it as simple as possible to ensure real time capability and as accurate as required to represent to dynamic behavior of the engine. This was achieved by combining appropriate groups of pipe elements to keep the derived model as simple as possible. The subsequent larger volumes of certain elements reduce model complexity and increase model stability without significantly affecting accuracy. Flow coefficients, effective flow areas and extrapolated turbocharger maps are transferred from the detailed 1D model to the 0D model. The elements contain the required calculation formulas and are interconnected according to the engine arrangement. The tubes and cylinders are described in 0D using the comparatively simple and fast “Fill and Drain” method. The 1D effects in the piping system and the detailed modeling of the gas supply system including possible pulsation effects are not investigated to achieve real-time capability.

The transient behavior of the system is described by solving a set of linear differential equations. At each time step a change of state of the total system is calculated by a flux vector. The flux vector is used to calculate the change of state. The flux vector contains mass and enthalpy fluxes. The change of state of the total system contains the mass change and the change of the inner energy of the volumes. This change of state is then integrated with the explicit Euler method.

Compressors and turbines are based on standard SAE turbocharger maps [13]. Two-dimensional look-up tables contain the reduced mass flow and efficiency. The turbocharger speed, the adjacent pressures and the adjacent temperature in front of the turbine are input values for the turbine element. After the reduced speed and the pressure ratio have been calculated, the efficiency and the reduced mass flow are interpolated from these variables in the look-up tables. Turbine power, enthalpy flows and mass flow are the output variables required in the adjacent volume elements. The speed of the turbocharger is calculated based on the power equilibrium between turbine and compressor considering the inertia of the turbocharger shaft. Therefore, the inertia of the turbocharger shaft must be modelled; friction can also be taken into account in an optional high-pressure stage [14]. The cylinder modules are set up to calculate the exhaust enthalpy flow through the exhaust valves; this is necessary because the turbocharging system is driven by exhaust enthalpy. The mass flow and enthalpy flow through the inlet and outlet valves is described using the nozzle equation and the flow function. The cylinder pressure is calculated according to the 1st law of thermodynamics. The powertrain is modeled considering the inertia of the whole crankshaft including the generator. Depending on the difference in power between the consumer and the engine, the powertrain either accelerates or decelerates.

The 0D model must be calibrated with measurement data from the reference MCE or simulation data from a detailed 1D engine model. The heat transfer coefficients of the wall can be adjusted so that the same pressure values are achieved in the intake and exhaust pipes as in the reference data. The simplified MCE model can also be used for controller design and calibration within a reasonable time period when extended by a combustion model. Cycle-to-cycle variations can be easily imposed to the model by applying random number fluctuations to the indicated mean effective pressure (IMEP). This permits to transfer of the controller structure and the controller settings to the virtual multi-cylinder engine for SCE testing.

A more detailed description of the modelling and validation of the HiL approach can be found in [15] and [16].

3.2 Integration of the HiL model

The most comprehensive application of the HiL methodology encompasses the entire virtual multi-cylinder engine, including the powertrain and MCE control system (cf. Figure 7). To enable SCE operation with this variant, the real-time capable 0D-MCE model was configured and calibrated according to the methodology described in the previous section. Safe SCE operation in HiL mode requires the controller structure and parameters to be predefined.

By implementing the models and controllers in the test environment, only the setpoints for engine power or engine speed and generator load need to be defined. The engine operation is realized by the input values of the test bench and the output values of the models, controllers and the test bench itself. In the cylinder modules only the gas exchange, compression and expansion have to be calculated. The cylinder pressure is re-initialized in each cycle by the measured pressure at exhaust valve opening in order to take into account the influence of the increased combustion pressure in the expansion phase. The input value for the powertrain model is the indicated power. It is calculated from the indicated mean effective pressure in each cycle based on the indicated cylinder pressure curve.

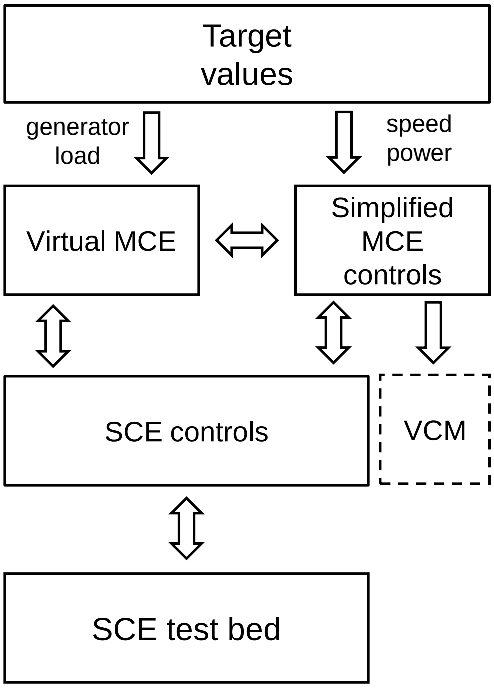

Figure 7. Hardware-in-the-Loop Methodology – Setup for SCE testing

Engine speed, boost pressure and exhaust back pressure are the output values of the model required as input for the SCE test bench. The real-time models can also be used to obtain numerous other actuation signals, e.g. gas supply pressure, gas valve actuation or actuation of a variable intake valve train.

3.3 Assumptions and limitations of the HiL methodology

Before the results of the investigations at the SCE are presented, the limitations of the HiL methodology will first be discussed. On a full engine, fluctuations in combustion from cycle to cycle are compensated for and averaged out by cylinder-specific differences. The cycle-individual transfer of the cylinder pressure at exhaust valve opening and the indicated mean effective pressure of the SCE to all the cylinders of the full engine model causes an amplification in cycle-to-cycle fluctuations. Thus, a misfire cycle of the SCE acts as if all cylinders of the MCE have no combustion. This behavior must be taken into account when parameterizing the speed or power controller. Here a compromise must be found between a short response time of the controller in transient operation and a stable stationary operation.

The speed controller in island operation and the power controller in grid-parallel operation were designed as simple PID path without additional feed-forward controllers. Only a dependency of the control parameters on the engine power was implemented. The basic parameters were set using the offline simulation of the full motor model. In order to ensure the above-mentioned compromise between short response time and stability, however, the control parameters had to be readjusted on the SCE.

The use of the variable valve train for speed and power control required the incorporation of inlet valve lift curves in the 0D MCE model. The set of curves was derived from stationary measurements at nominal speed. A consideration of the closing behavior at speeds deviating from the nominal speed was therefore not taken into account in the gas exchange calculation.

The fuel mass was introduced directly into the inlet port of the engine by means of a port injection valve on the test bench. The reference engine is mixture charged. The time offset between the introduction of fuel into the fresh air upstream of the low-pressure compressor and the arrival of the charge in the cylinder was derived on the basis of 1D MCE simulations in the form of a constant delay time and a first order filter element and implemented at the SCE. A differentiation of the time constants for different cylinders was not considered.

The control of the air-excess ratio to maintain NO_x emissions was based on the Leanox® approach used in the baseline engine [17]. The control algorithm of the reference MCE could not be implemented one-to-one. An important component for the implementation of the emission control is the volumetric efficiency. In the presented test bench investigations, a data-based model of the volumetric efficiency was stored in the emission controller both for the reference engine with compressor recirculation control and for the variants with IVC control. The data for this comes from a large number of stationary operating points. However, there may be variations between the model and the actual volumetric efficiency due to speed, scavenging gradient and the actual time of inlet valve closing. This causes differences in the measured fuel quantity and thus deviations between the target air-excess ratio and the actual air-excess ratio.

Finally, the regulation of boost pressure, exhaust back pressure and gas supply pressure should also be mentioned. Despite a very fast, model-based controller, implemented in the test bench automation system, an immediate conversion of the set values from the HiL model is not possible and a small time delay had to be accepted. The faster the target values are changing, the more pronounced is the effect of this delay.

3.4 Results in island grid operation

In a first step, the evaluation of the transient engine behavior is to be presented on the basis of the engine operated in island mode. In the reference case (J624-H; CB controlled), the engine load, and thus the engine speed, is controlled via the compressor bypass. In addition, in the event of a speed drop caused by a generator load step, the mixture is deliberately enriched in order to have more fuel energy available in the combustion chamber [17]. In the comparative cases (J624- VCM; IVC controlled), the speed is controlled by shifting the timing of the intake valve closing event.

Figure 8 shows the results of a 25 percent load step over time. It can be seen that in the case CB controlled the compressor bypass valve closes completely immediately after load is applied (time = 0 %) to limit the speed drop. This causes an increase in boost pressure. In combination with the enrichment, which is dependent on the speed drop, this results in the increase of the indicated mean effective pressure. The maximum speed drop is limited to about 150 rpm. In the IVC controlled case, the engine reacts – with the compressor bypass valve fully closed – by retarding the IVC after load is applied. As a result of the sudden increase in the volumetric efficiency, the boost pressure initially drops and the exhaust back pressure rises, resulting in a brief negative scavenging gradient. The indicated mean effective pressure jumps steeply upwards. This not only limits the speed drop. Another consequence is that the nominal speed is reached after half the time of the reference case. The limitation of the retarded IVC timing was chosen so that no knocking combustion could be detected during operation. The limit was set constant for every test run and was not a function of another variable (a major difference to the simulation cases). In the case of a constant ignition timing (red curves), a retardation of the IVC about 20 °CA was thus possible. If, however, the ignition timing is also set to late if a speed drop is detected, the IVC limit can be retarded even further (blue curves). The indicated mean effective pressure reaches higher values after the load step; actually, the target IMEP can be reached. This does not reduce the immediate engine speed drop, but the recovery time is further shortened compared to the reference.

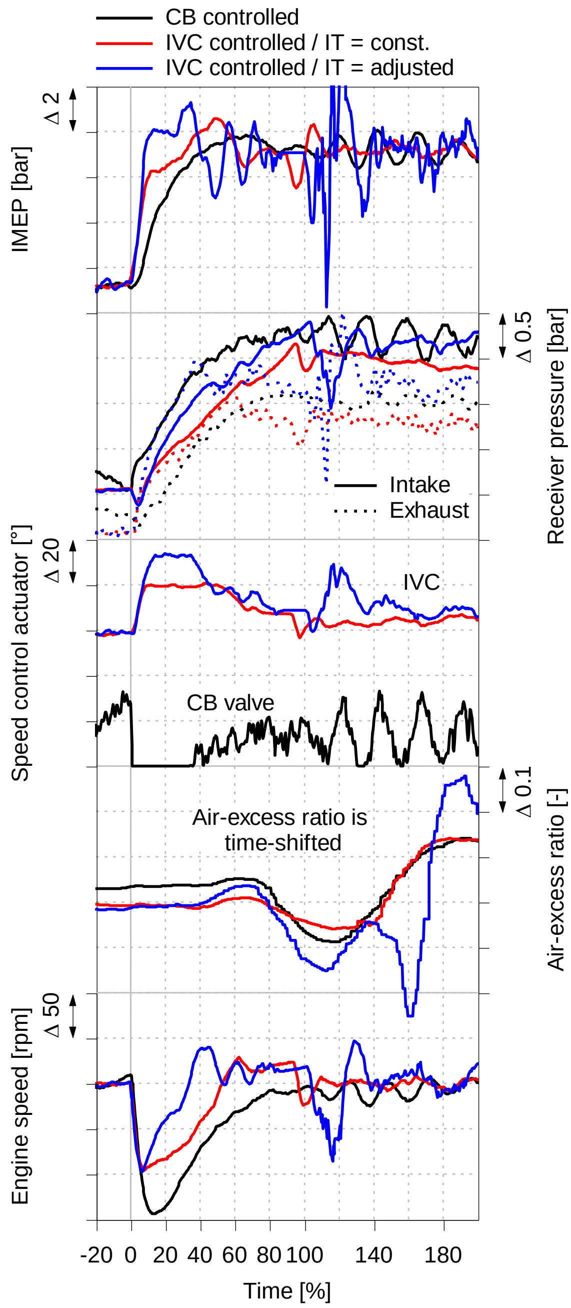

Figure 8. Island grid load steps – Comparison of control strategies

The actual drop in the air-excess ratio during load pickup in the IVC controlled cases - the air-excess ratio was determined from values of the exhaust gas analysis and therefore has a time offset - also leads to a restriction of the latest possible IVC, as the tendency to knocking combustion increases further. Due to an insufficiently tuned controller for the air-excess ratio in combination with the retarded ignition timing, misfire occurs around the timestep 100%. However, engine speed drop can be very quickly counteracted by the speed governor via IVC.

Figure 9 shows the possible increase in load acceptance in island mode for the two IVC controlled cases compared to the CB controlled reference case. Only the maximum speed drop was used as the criterion for distinguishing between ISO class G1 and class G3. The recovery time was not considered. For the baseload range investigated, the maximum possible block load increased by five to ten percentage points for class G1. A further slight increase is possible by retarding the ignition timing. It should be mentioned here that the load steps were only varied in a resolution of five percentage points. In addition, the limiting IVC was also adjusted in steps of five degree crank angle.

For class G3, the maximum block load can also be increased by five to ten percentage points. In the range from 40 to 50 percent baseload even by fifteen percentage points through additional retardation of the ignition timing.

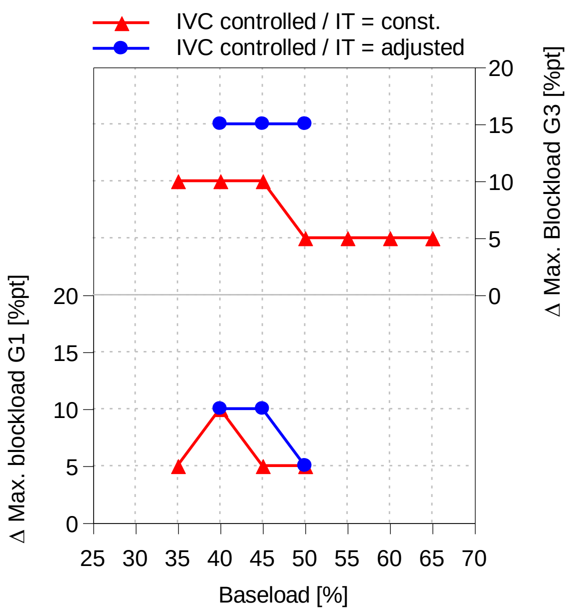

Figure 9. Island grid load acceptance – comparison of control strategies

An investigation of baseloads lower than 35 % was not possible at the SCE, because the current set up of the speed control in combination with the cycle-to-cycle fluctuations of the SCE did not allow for a stable engine operation.

It could thus be shown that the load step behavior depends significantly on the knock-limited, latest possible IVC and that an additional intervention via the ignition timing for knock control has further advantages. In the next step, the influence of the air-excess ratio during load increase was of interest. The CB controlled reference case shows active enrichment as a function of the engine speed deviation.

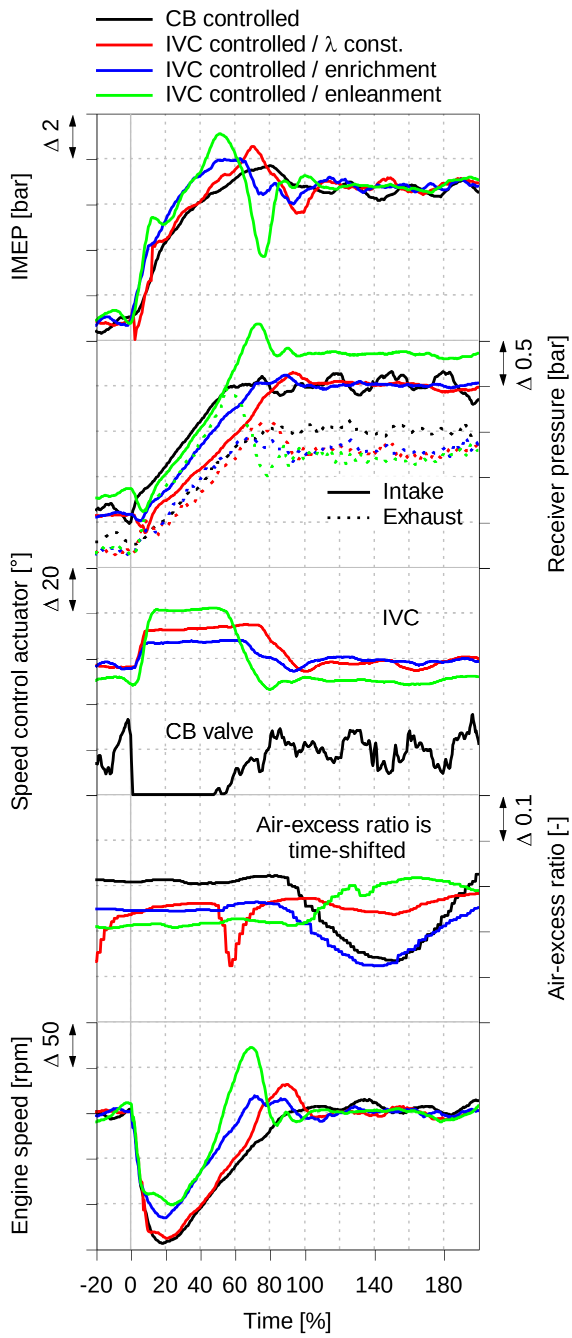

Figure 10. Influence of emission control strategy on load pickup behavior in island grid operation

For the IVC controlled case, three control approaches for the air-excess ratio were developed and compared. First, the air-excess ratio should be kept as constant as possible during load increase (similar to *strategy 1* in Chapter 2.2). For this purpose, two further approaches with enleanment and enrichment were compared. Figure 10 shows, that the latest possible IVC must be set earlier when enrichment is applied than at a constant air-excess ratio for a 25 percent load step. The enleanment allows a more retarded IVC than in the case with constant air ratio. This makes it possible to achieve the same speed drop as with enrichment despite a lower fuel concentration in the combustion chamber. It is also obvious that the speed after reaching the nominal speed strongly overshoots in the case of enleanment.

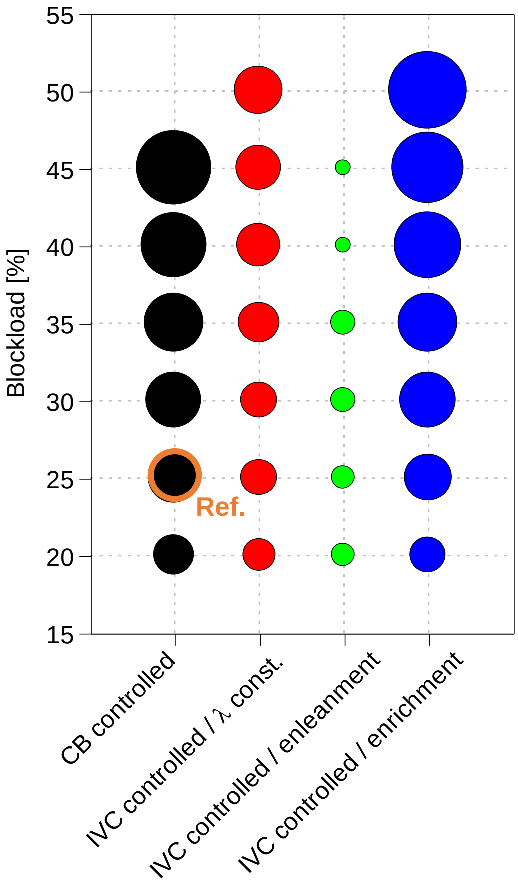

Figure 11. Qualitative comparison of integral NO_x emissions during load pickup for various control approaches

An adjustment of the speed control parameters could reduce this behavior. In the case of a constant air-excess ratio, the course of the air-excess ratio can be used to detect a multiple spontaneous misfire events of the SCE. In the specific case, misfire also occurred immediately after the load step. This caused a larger speed drop compared to the other IVC-controlled cases. The reason for this behavior was found in the prechamber gas supply which, in some cases, was not able to appropriately follow the high dynamics in boost pressure. This might pose an additional challenge to the implementation in an MCE application.

Of particular interest were also the NO_x emissions during load increase, taking into account the different control approaches. Figure 11 compares the integral NO_x emissions for varying block loads and the different control approaches. The size of the bubbles represents the integral NO_x emission relative to the integral NO_x emission of the reference case at 25 % load step. All shown load steps exhibit engine speed drop to fulfill the speed drop criterion of class G1. It has been shown that nitrogen oxide emissions can be significantly reduced by using a variable intake valve timing for speed control. Only in the case of enrichment during the speed drop do the emissions reach the same or slightly higher level than in the reference case. However, the recovery time can be reduced in comparison to the reference. In the case of a constant air-excess ratio, which is also described in Chapter 2, a 5 %pts higher load step can be achieved with a reduction of almost 55 % in cumulated NO_x emissions. The enleanment strategy enables the same maximum load step as the reference. The reduction in NO_x emissions is roughly 93 % for the maximum load step.

3.5 Results in grid parallel operation

Finally, results of the SCE in HiL mode for grid parallel operation are shown in Figure 12. In this mode, the frequency is applied to the generator by the grid, keeping the engine speed constant.

The power is again controlled by the compressor bypass in the reference case or by adjusting the inlet valve closing time in the VCM case. Load ramps from medium partial load (35 % of nominal load) to full load are shown. It should be noted here that a load ramp from idle to full load was not possible due to the restrictions mentioned in Chapter 3.3. Nevertheless, a comparison was made between hot engine conditions and cold but preheated engine conditions. In the HiL-model, the temperature of the exhaust pipe system was artificially lowered by 600 °C for this purpose. As a result, the heat losses to the pipe walls increase and a lower exhaust enthalpy for the turbines is available in the model. As a consequence, boost pressure build-up is limited. For the hot reference case (CB controlled / hot), the compressor bypass in the model is completely closed shortly after the new engine target power is set and only opens again when the rated power is reached. In the cold reference case (CB controlled / cold) double the time of the hot reference is required.

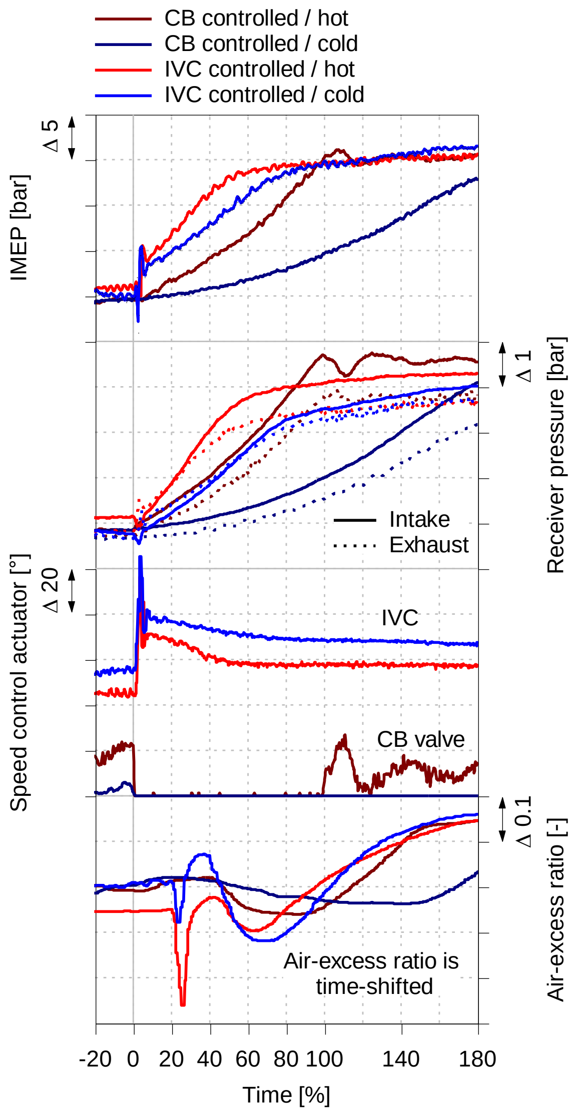

Figure 12. Ramping to full load in grid parallel operation – comparison of control strategies

On the other hand, in the case of IVC control, the intake valve closing timing will be retarded in order to increase the volumetric efficiency and increase the engine power. This causes the already mentioned, sudden initial step of the indicated mean effective pressure. As in island operation, sudden misfire was detected right after the initial retardation of IVC. This leads to the characteristic drops in air-excess ratio. Under hot conditions the power ramp rate can be substantially increased. Rated power is already reached after 80 % of the hot CB controlled reference. However, the particular advantage of control via IVC can be seen under cold engine conditions (IVC controlled / cold). Here, the sudden increase in the volumetric efficiency not only causes a large initial step in indicated mean effective pressure. Further, the boost pressure build-up is also accelerated as a result. As a consequence, the nominal load can be reached within the same time period as in the hot reference case.

4 CONCLUSIONS

The paper summarized investigations on the improvement of the transient performance of a large bore premix gas engine by applying a variable intake valve train for speed and power control in comparison to a state-of-the art engine configuration with compressor bypass and constant Miller timing. Based on a simulation study the thermodynamic effects of using a variable intake valve train for engine control and the potential for improvements were presented. Depending on the applied control strategy (taking also enrichment and an adjustment of ignition timing into account) the maximum block load steps in island grid operation can be increased by 15 to 20 percentage points compared to the reference engine. In grid-parallel operation the ramping time from idle to full load can be reduced to 50 % of the reference engine.

The simulation results were consequently validated by measurement results from a transient single cylinder research engine. The SCE was operated using real-time capable MCE models in a hardware-in-the loop methodology. The general potential for improvements could be demonstrated. Furthermore, it was possible to make statements about knocking and emissions in transient operation. By using a variable intake valve train, it is possible to increase the maximum block load steps in island grid operation by up to 15 percentage points depending on the tolerated engine speed drop (ISO class). In addition, a reduction in integral NO_x emissions during load pick up to one half of the reference engine can be accounted for. In grid parallel operation, ramp rates can be increased substantially. In particular, under cold/pre-heated engine conditions ramping times as with the reference engine in hot conditions can be achieved.

The results from simulation and SCE testing generally fit quite well. Minor deviations can be explained by the slightly different modelling and control approaches and the coarser discretization in terms of limited IVC and block load during SCE testing.

5 OUTLOOK

The use of a variable intake valve train for speed and power control enables to significantly increase the engine transient performance. This comes along with potential challenges regarding engine controls. Model based controllers can be introduced to simultaneously address multiple actuators. Gas admission to scavenged prechambers has to react quickly in order to follow the rapidly changing state of the cylinder and deliver the right amount of fuel.

Future generations of gas engines may be able to achieve higher nominal specific power output by consequently introducing more aggressive Miller timings. Here, a variable intake valve train has an even higher potential for transient improvements in the low load range in particular.

The used simulation models and the applied HiL methodology are subject to assumptions and certain limitations. A demonstration on a multi-cylinder engine will reveal the full potential of a variable valve train on the intake side.

6 DEFINITIONS, ACRONYMS, ABBREVIATIONS

bmep	Brake mean effective pressure
CB	Compressor bypass
HiL	Hardware-in-the-Loop
H_u	Lower heating value
IMEP	Indicated mean effective pressure
IT	ignition timing
IVC	Intake valve closing
L_{st}	Stoichiometric air-to-fuel ratio
MCE	Multi-cylinder engine
NO_x	Nitrogen oxides
$P_{Electric}$	electric power
p_{Rec}	Intake receiver pressure
R_{Rec}	Air/fuel mixture gas constant in receiver
SCE	Single cylinder engine
SCR	Selective catalytic reduction
T_{Rec}	Intake receiver temperature
VCM®	Valve Control Management
$\eta_{Electric}$	Generator set electrical efficiency
η_{th}	Engine thermal efficiency
η_{Total}	CHP plant total efficiency
λ, λ_v	Air-excess ratio
λ_R	Volumetric efficiency

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