

Advanced model-based control of B36:45 LNG engines

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Abstract

The framework of model predictive control is used in this paper to optimally control the operation of an B36:45 LNG engine. The model of the engine is based on real life data from an installed B36:45 gas engine in a power plant. Stored data from the plant was used to develop a state space model of the process consisting of 2 manipulatable variables, 3 measured disturbances and 6 measured outputs. The goal is to use global ignition timing and the charge air pressure as control variables to minimize the heat rate while considering constraints on the measured outputs. Heat rate of the engine is directly related to engine performance efficiency. Results show that a model based controller has the potential to be used as an advanced controller for optimal operation of this engine.

Keywords: B36:45 LNG engine, MPC, optimal operation

1 Introduction

Bergen Engines AS is a developer and producer of gas and diesel engines for the marine and land-based power market. The factory is located just north of Bergen on the west coast of Norway and have been since it moved from the city centre of Bergen in 1965. Bergen Engines was part to the Ulstein group from mid-eighties until 1999 from which it has been a part of Rolls-Royce. The latest LNG (Liquefied Natural Gas) fuel engine type developed is the B36:45 engine family, and a graphical representation of the engine is shown in Figure 1.



Figure 1. 20 cylinder B36:45 LNG gas engine with generator. (Courtesy: Bergen Engine AS)

For an LNG gas engine, the engine efficiency is in general as shown in Figure 2 for a given power output. It should be noted that the efficiency is not very good below 20% power output and rises slowly from approx. 30% power output to 100% at which the efficiency is close to 50%. The distribution of the efficiency losses can be seen in Figure 3 where most of the losses are to the exhaust gas.

In the search for increasing the engine efficiency, more

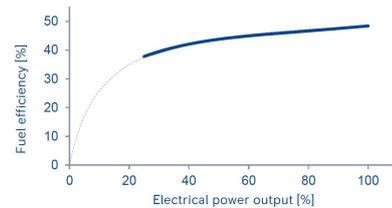


Figure 2. Fuel efficiency for and LNG engine. (Courtesy: Bergen Engine AS)

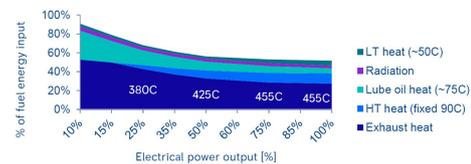


Figure 3. Distribution of losses for an LNG gas engine. (Courtesy: Bergen Engine AS)

complex logic, which takes into account more of the information available, is constantly developed. This has resulted in a large increase in parameters and static maps that interact with each other, which makes the engine tuning phase a complex and time-consuming job.

It is of great interest to Bergen Engines AS to use advanced model based controller such as a model predictive controller (MPC) for generating optimal set points based on measured states of the engine and known disturbances. This is set to be the first step towards a more data driven, self-optimizing algorithms that can use the large amount of data produced. The ultimate goal here is to use a reduced set of parameters which can be used to prioritize different possibilities such that various requirements are reached. For instance one of the requirements for a project can be to reach a given NO_x set point and secondary fuel efficiency, while for another case, it can be the fuel economy as the most important requirement while keeping the NO_x within given constraints.

The main engine controller currently used is an embedded controller from Woodward Inc. The LECM (Large Engine Control Module) is a purpose-built controller with suitable hardware for interfacing large industrial engines. The software for the controller is developed and built in-house at Bergen Engines and hence gives a large flexibility in custom made control algorithms. The control software used in the LECM is developed in MATLAB Simulink with a proprietary library for hardware access to the ac-

tual controller from MotoHawk. The MotoHawk library is a rapid programming development tool which allows engineers to quickly develop control software in Simulink to run on a MotoHawk enabled control module; like the LECM. Even though MotoHawk as many pre-made algorithms available, it does not have any available MPC control structures and hence this must be developed using standard MATLAB and Simulink functions. Thus the main goal of this project work is to add MPC control structure to this existing controller for engine fuel optimization.

MPC has been proven in use for instance in steam generators and servos as illustrated in Richalet (1993). In Koli (1981) an MPC for a turbocharged gasoline engine with EGR (Exhaust Gas Recirculation) has been developed where the dynamics of the model is defined by simple feedforward neural network. It showed that the simple black-box model is sufficient for using with MPC. A paper by Luther (2002) compares neural-net-based modelling to Adiabatic Mean Value Engine Modelling. Some system parts were modeled with good accuracy, but others with large deviations. A neural net based MPC has been used for the non-linear MIMO system with great performance. In Lu et al. (2015) support vector machine for non-linear system identification has been used. The engine model shows the rotation speed of the engine as a function of fuel flow, and an MPC is designed in Simulink.

2 Operational Philosophy

The B36:45 engine family is a medium speed lean-burn single fuel spark ignited internal combustion engine. It mainly uses LNG as fuel source and a run at 720/750 rpm for 60- and 50 Hz applications respectively. It is turbocharged and has a 2-stage water cooled charge air cooler. In most land-based power plants it connected to a generator which is often connected to either a small local grid or a large national grid. The engines nominal power output is 600 kW pr cylinder mechanically and comes in both in inline and Vee configurations. The smallest is an inline 6-cylinder engine and the largest is a Vee 20-cylinder engine.

2.1 Lean burn gas engine - Otto Cycle

A lean-burn gas engine runs with a high air to fuel ratio compared to the required air for a stoichiometric combustion. This lowers the combustion temperature and hence reduces NO_x emissions. The B36:45 engine is a lambda 2 engine, indicating that it runs with twice the required amount of air for a stoichiometric combustion. This lean mixture is difficult to ignite and hence a pre-combustion chamber is mounted in the cylinder head. A rich mixture is here ignited by a spark plug and the resulting flames will propagate out and into the main chamber where it will ignite the lean mixture. Figure 4 shows an illustration of the engine process and some control loops. The engine is a 4-stroke (also known as the Otto cycle) which means that there are 4 distinct phases for the combustion process.

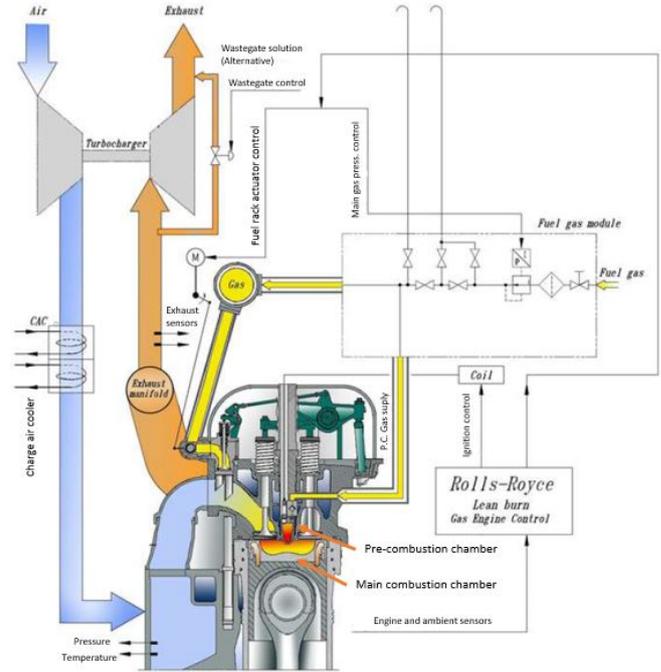


Figure 4. Illustration of the process with some control loops. (Courtesy: Bergen Engine AS)

2.2 Main control loops

This section shortly describes the most common control loops for the combustion process which are controlled by the engine control system today. There some other control loops as well, but the major once are described here.

2.2.1 Speed Control

It is a PID controller which controls the flow of fuel admission in order to keep the engine speed at a desired set point. When connected to a large electricity grid with fixed frequency, speed control loop is used to control the engine power output to a given set point. Increasing the set point for speed will make the speed controller to increase the fuel admission by increasing the fuel flow which will result in an increase of engine power output as the grid frequency cannot change.

2.2.2 Air pressure/AFR control

The AFR, or air pressure control, is a control loop whose main purpose is to control the charge air pressure in the air receiver to a given set point. The set point is based on a map with engine power output and engine speed as inputs. The set point map is derived based on numerous of test runs at the test bed by skilled engineers. Since this is a static map the set point must be biased to a certain degree based on operational conditions. The air pressure control is the most active and influents of all control loops. It dictates most of the engine behaviour as it directly controls the air/fuel ratio under all operational conditions. The output from the air pressure control is a position control signal to a waste gate actuator. The waste gate actuator will control the amount of exhaust by-passing the turbine

part of the turbocharger(s) and hence the energy used to increase the air pressure. The feedback to the air pressure control is the measured air pressure in the air receiver which then forms a closed loop control system. One of the goals of this project is to find the optimum charge air pressure to maximize engine efficiency.

2.2.3 Air temperature control

In order to further control the air/fuel ratio the temperature of the combustion air should be kept at given values. The air temperature is often controlled by a low level PID controller to a given set point based on operational ambient conditions. The control signal from the PID controller is used to control a 3-way valve which will direct, or bypass, water to a 2-stage charge air cooler. By increasing the amount of water going through cooler the temperature can be reduced, and vice-versa. An increase in temperature will lead to an increase in NO_x due to lower air mass added to the combustion process and due to increase in temperature. This will also result in the engine operating closed to the ignition knocking limit.

2.2.4 NO_x control

There are dual NO_x sensors at the exhaust outlets to measure NO_x [ppm] and Oxygen [%]. A PID controller control the NO_x level to a given setpoint. However, the NO_x controller will bias the air pressure controller's base set point between +10% and -5%, so there is direct interaction between these controllers.

2.2.5 Global ignition timing control

The ignition timing is the time in crank angle (CA) degrees at which the cylinder individual spark plug is ignited in the pre-combustion chamber. The base timing setpoints is created as a map based on various testing on the testbed by engineers. This base timing is adjusted such that a good margin to ignition knocking while maintaining high level of efficiency is achieved. The global timing is adjusted so that the maximum pressure in the cylinder is occurring around 13-15 [degCA] after TDC (Top Dead Center). This will give rise to best performance. The control of the timing location is complex and difficult to maintain.

In this project, this complexity is reduced by allowing an advanced model based controller to find the optimum timing set points given a set of constraints to protect the engine from running into dangerous operational points.

2.3 Global ignition timing and efficiency

In Figure 5, an indication of the relationship between global ignition timing and peak pressure and engine efficiency is shown. These curves are based on data from tests performed on the previous version of the Bergen LNG gas engine, the B35:40. This engine operates at lower brake mean effective pressure (BMEP) than the B36:45 engine and with lower peak pressures. The indicated relationship

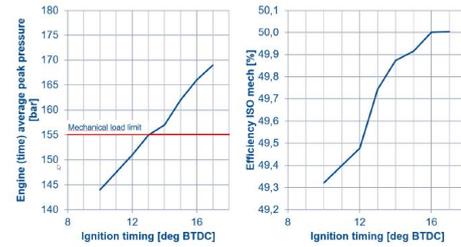


Figure 5. Global timing influence on efficiency and cylinder pressure.

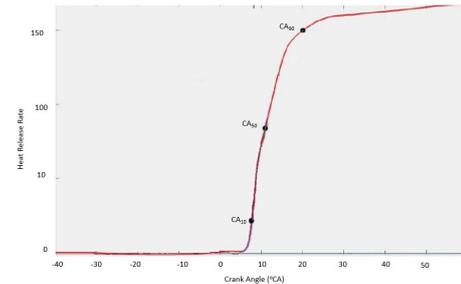


Figure 6. Heat release curve.

however shows similar behaviour also for the B36:45 engine. Earlier ignition timing, i.e. ignition before top dead centre (BTDC) will increase the peak pressure in the cylinder, but it will also increase the efficiency of the engine. There are however, as indicated, a mechanical limit in the construction of the engine on how high the peak pressure can become before there is a risk of mechanical breakdown. The engine control system therefore monitors the peak pressure for all cylinders and in case of too high pressure the engine will shut down. For the MPC, this pressure will be used as a constraint to avoid too high pressure.

2.4 Global ignition timing and heat rate

The accumulated heat release average over a number of combustion cycles are used together with the measured engine load to calculate the heat rate of the engine. The smaller the heat rate of the engine, the less is the amount of fuel used per unit power output. In other words, minimized heat rate will maximize engine efficiency. A typical heat release curve for different ignition timing (crank angle) is shown in Figure 6. The heat release curve is shown with the locations for CA10, CA50 and CA90. These are the locations at which 10%, 50% and 90% of the fuel have been burned in the combustion chamber.

3 Process modelling and description

Operational data collected from a commercial B36:45 engine operating in a power plant in the city of Tabor in Czech Republic was used to develop a data-driven model. The process data is captured at 10 Hz sampling rate by a local data logger. This logger then pushes the data to the cloud every 30 minutes. This data is used both for modelling, fault detection, machine learning and support

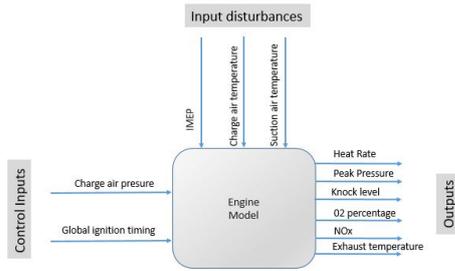


Figure 7. Process functional diagram.

at Bergen Engine AS. A functional diagram of the process showing the control input signals, output signals and the input disturbances is shown in Figure 7.

The two control inputs are the *charge air pressure* and *Global ignition timing*. There are six measured outputs from the system namely *heat rate*, *peak pressure*, *knock level*, *oxygen (O₂) percentage*, *NO_x* and *Exhaust temperature*. In addition there are three measured input disturbances namely *IMEP (Indicated Mean Efficiency Pressure)*, *charge air temperature* and *suction air temperature*.

3.1 Charge air pressure

The charge air pressure is the pressure of the combustion air entering the combustion chamber from the air receiver. This pressure is controlled by adjusting the waste gate bypass valve such that the pressure is according to set point. Traditionally these set points are found based on testing on a real engine where the emissions are measured and the distance to the knocking limit is observed. In addition, the pressure is mapped towards the turbocharger to prevent any stalling or crossing the surge limit. The charge air pressure set point is usually a map where the set point is based on the engine speed and the power output but biased from several sources to make it adaptable to ambient conditions and ageing. This charge air pressure set point is traditionally highly driven by the engine power output in an almost linear relation. Nominal charge air pressure at 100% power output is approximately 4.2 barg.

3.2 Global Ignition timing

The global ignition timing command is used to set the base ignition timing for the engine. Each individual cylinder will adjust this base timing withing a window of ± 3 [degCA] to balance the peak pressure off the cylinders. The ignition base timing is traditionally found during testing and running the engine close to the knocking limit during controlled environments. It is however not given that the same conditions will be applicable on every project and hence margin must be added on the set point to take into account different fuel compositions and ambient conditions. Ageing is also a factor here. To counter act these changing conditions several set point modifiers are in place which will bias the set point if a change in ignition knocking is detected or if exhaust temperature is increased. In addition, the location of the centre of combus-

tion is measured based on the heat release curve from the combustion process. This location is used as a set point on a second level PID controller which biases the base set point so that this location is kept on set point as well. But none of these measures are there to optimize the fuel consumption over time and to take into account all these constraints as an MPC controller can do.

3.3 Suction air temperature

This is slow varying input disturbance to the system which has the least impact. The suction air temperature is the air temperature measured at the inlet of the compressor part of the turbocharger. This disturbance will inform the system about the ambient conditions under which the engine is currently operating. The ambient temperature, and hence the suction air temperature will vary over a year for a given installation location. This variation might be small or large depending on the location. It might therefore have an impact in some cases and hence it is included here.

3.4 Charge air temperature

The charge air temperature is measured in the air receiver and is the temperature of the combustion air fed into the combustion chamber during the opening time of the inlet valve. This temperature is in some cases actively regulated by a PID controller, while it in some installations are mechanically adjusted at max power output to give a certain temperature. Normal operational temperature here is around 50-55 °C. This might however change if the humid conditions are present such that condensation might occur at this temperature. The temperature of the charge air influences the air mass which is available to the combustion process and hence any change here will impact both NO_x emissions and the resilience towards knocking.

3.5 IMEP

IMEP (Indicated Mean Effective Pressure) is a measure of produced work of the engine including the friction work, i.e. the actual work done by the engine independent of the engine displacement. It is a measure of the average pressure in the combustion chamber of the engine cycle. IMEP is measured directly by the cylinder pressure sensors. The highest and lowest values are removed and the average over the number of cylinders is taken and fed into a moving average filter over 100 cycles. This final value indicates the current loading (power output) of the engine.

3.6 Heat rate

This is the variable that is to be minimized. It is indicative of the relation between the power output and the fuel consumption estimation. The heat rate is given as the relation between the IMEP and the total heat release. The IMEP is measured by the cylinder pressure sensors as well as the total cumulative heat release. The heat release is given as [kJ/cycle] and is estimated based on the pressure rise curve measured by the cylinder pressure sensors.

3.7 Knock level

Knock level is also known as engine detonation and is when the combustion takes place prematurely in part of the compressed air fuel mixture in the cylinder. This knocking can cause severe damage to the engine if not responded to early because of high frequency pressure waves causing very high cylinder pressures potentially above the design limit of the engine. The engines are constantly pushed towards the knocking limit as this area produces better fuel efficiency at higher power outputs. Knocking might occur if the air fuel mixture is not correct or substances such as oil leaks into the combustion chamber causing changes to the burn rate of the air/fuel mixture. Each cylinder is monitored for knock level and any increase in knocking results in that cylinder ignition timing being retarded for some time. If several cylinders experience knocking the global timing point is usually retarded to avoid any further increase into non-operational areas. The knock value is measured by looking at the ripples on the cylinder pressure curve after the ignition location. This value indicates the level of knocking for each cylinder but is averaged for all cylinders here. This will in general only pickup up globally severe knocking. Knocking can also be reduced by lowering engine power output or increasing the amount of air in the air/fuel mixture resulting in a leaner mixture. That will however impact efficiency.

3.8 Peak pressure

The peak pressures are measured from cycle to cycle and is the highest measured pressure in the combustion chamber of the combustion cycle. The peak pressure is a value which must be limited as there are design limitations on the engine for how high pressures the internal components can withstand without damage.

3.9 NO_x

The NO_x emissions are measured in the exhaust outlet after the turbocharger. The emissions are measured with sensor from Continental most commonly used on trucks and cars. This sensor gives the wet NO_x values in ppm directly and is used in a closed loop regulation for controlling the level of NO_x to a given set point. The NO_x values are good indications of how rich or lean the fuel mixture in the combustion is. A high NO_x value indicates a rich mixture and vice versa. The NO_x value is very sensitive to these variations and will rapidly increase in case of the charge air pressure is reduced. It should however be noted that the NO_x values should rarely be seen drifting high during steady operation. During transients a change in NO_x value is expected as the engine increases the air pressure during the transient to get better margins to the knock limit.

3.10 O₂

The same sensors that measures the NO_x level in exhaust will also measure the O₂ level. In the traditional engine

controller, the O₂ percentage is used actively for engine limitation. That is if the O₂ level becomes too low, which indicates a too rich mixture, the engine will limit the fuel admittance and hence reduce power output.

3.11 Exhaust temperature

Traditionally the exhaust temperature outlet from each cylinder has been used to balance the engine power output from each cylinder. Before the cylinder pressure sensor era the only possibility to check how much each cylinder contributed to the power output was by looking at the deviation in exhaust temperature between the cylinders. The exhaust temperature used here is the temperature measured in the collecting pipe just prior to the turbine part of the turbocharger. This exhaust will therefore be an indicative of the all cylinders on that pipe collectively. The temperature will increase in case the power output increases and mixture becomes too rich. It is therefore very dependent on the charge air pressure, but also the ignition timing. In case the ignition timing is retarded the temperature will increase and hence this needs to be handled.

3.12 State space model of engine

In order to find the relationship between the control inputs, the measured outputs and the measured input disturbances, system identification toolbox in MATLAB was utilized to obtain a discrete state space model of the form,

$$\begin{aligned} x_{k+1} &= Ax_k + Bu_k + B_d u_{d,k} \\ y_k &= Cx_k \end{aligned} \quad (1)$$

Here, x is the state vector, u is the vector of control inputs, u_d is the vector of measured disturbances and y is the vector of the measured outputs. The state matrix, input matrix, disturbance matrix and the output matrix are $A \in \mathbb{R}^{25 \times 25}$, $B \in \mathbb{R}^{25 \times 2}$, $B_d \in \mathbb{R}^{25 \times 3}$ and $C \in \mathbb{R}^{6 \times 25}$ respectively. The subscript k denotes the discrete time steps.

4 Optimal control problem formulation

The goal of utilizing an advanced optimal controller is to maximize the engine efficiency. This is achieved by minimizing the heat rate of the engine. To do so, the controller will generate optimal values for the *charge air pressure* and *global ignition timing*. From an optimization point of view, these two are the decision variables. These signals will not directly control the process values but will act as optimal set points for the lower level PID controllers which in turn will adjust accordingly to achieve optimal operation.

The physical constraints for the charge air pressure will be imposed as bounds to the optimizer. By a defined upper limit for what is physical possible and at the same time set a lower bound close to 0 barg. Since the process will not be working on 0 barg air pressure, the lower bound will be set to 0.3 barg and the upper bound to 4.5 barg to give some regulation margin. The charge air pressure

cannot instantly change from one pressure to the next and hence the optimizer is limited based on rate of change of the control value such that it cannot change instantly. Nor can the waste gate valve change instantly and hence such limitations makes sense.

The global base timing will influence the efficiency of the engine but also has an impact on the peak pressures, NO_x generation, knocking and exhaust temperature. By advancing the global timing, the peak pressure increases and this needs to be within the design limit of the engine to prevent mechanical damage to the it. If the ignition is retarded the exhaust temperature increases and the NO_x emissions decreases as the combustion air temperature increases due to longer burn duration. By retarding the ignition timing, the heat rate increases to indicate less efficiency. There are some physical known limitations of the ignition timing that should be obeyed. The global ignition timing is seldom, if at all, below -8.5 [degCA] for a running power plant connected to the grid and producing power at nominal speed. It will also not be possible to advance the timing more than to -20 [degCA]. Nominal global ignition timing is usually in range of -12 [degCA] to -16 [degCA].

The knock level here is included as constraint at it is a limiting factor for advancing the global ignition timing too much or reducing the charge air pressure too much. Typically, the engine is shutdown with a value over 30%.

The nominal peak pressures during 100% power output at nominal speed is usually around 175 [bar]. The control system has alarms and shutdown conditions if sustained operation around 200 bar is experienced and hence the optimizer should avoid operation above 200 bar, and preferably limit operation to 180 bar but with some slack.

The NO_x value would be used as a constraint during the optimization phase such that it can stabilize below at least 150 ppm.

The O₂ is rather critical and hence strict constraints on the low level is to be used. During normal operational conditions the O₂ percentage is somewhere between 8.5% and 12.5%, with nominal condition a approx. 9.5%. The optimization should be rather strict on the lower limit while the upper limit can be broken during given condition. The aim should however be to stay within the limits of 8.5% and 12.5%.

There are limitations from the turbocharger supplier on the max inlet/suction temperature of turbine and hence these needs to be obeyed. The constraints can here be set based on normal operational conditions where the exhaust temperature before the turbocharger turbine should not exceed 600 °C.

Table 1 shows the constraints on the output signals and Table 2 lists the constraints on the control inputs and input disturbances.

In order to minimize the heat rate of the engine while still satisfying the constraints on both the inputs and the output signals, a constrained optimal control problem is

Table 1. Constraints on the outputs.

Signal (symbol)	lower limit	Upper Limit
Peak Pressure (P_p)	0 bar	180 bar
Knock level (K_l)	0%	30%
Heat rate (h_r)		
Exhaust temperature (T_e)	0 °C	600 °C
NO _x (N_{ox})	0 ppm	150 ppm
O ₂ (O_2)	8.5%	12.5%

Table 2. Constraints on the control inputs and disturbances.

Signal (symbol)	lower limit	Upper Limit
Charge air Pressure (P_{ca})	0.3 barg	4.5 barg
Global timing (G_t)	-20 degCA	-8.5 degCA
Charge air pressure rate (ΔP_{ca})	-0.2 barg/s	0.2 barg/s
Global timing rate (ΔG_t)	-0.3 degCA/s	0.5 degCA/s

formulated as,

$$\begin{aligned}
 \min_{P_{ca}, G_t, s_1, s_2} & \frac{1}{2} \sum_{k=1}^N h_{r,k}^T P h_{r,k} + P_{ca,k-1}^T Q_1 P_{ca,k-1} + G_{t,k-1}^T R_1 G_{t,k-1} \\
 & + \Delta P_{ca,k-1}^T Q_2 \Delta P_{ca,k-1} + \Delta G_{t,k-1}^T R_2 \Delta G_{t,k-1} \\
 & + s_{1,k}^T M_1 s_{1,k} + s_{2,k}^T M_2 s_{2,k} \\
 \text{s.t.} & \quad x_{k+1} = Ax_k + Bu_k + Bd u_{d,k} \\
 & \quad y_k = Cx_k \\
 & \quad 0 \leq T_{e,k} \leq 600 \\
 & \quad 0 \leq N_{ox,k} \leq 150 \\
 & \quad 8.5 \leq O_{2,k} \leq 12.5 + s_{1,k} \\
 & \quad 0 \leq K_{l,k} \leq 30 \\
 & \quad 0 \leq P_{p,k} \leq 180 + s_{2,k} \\
 & \quad 0.3 \leq P_{ca,k} \leq 4.5 \\
 & \quad -20 \leq G_{t,k} \leq -8.5 \\
 & \quad -0.2 \leq \Delta P_{ca,k} \leq 0.5 \\
 & \quad -0.3 \leq \Delta G_{t,k} \leq 0.5
 \end{aligned} \tag{2}$$

For the relaxation of the upper bounds on the output constraints O₂ and P_p, two slack variables s₁ and s₂ are used. The slack variables are then added to the list of decision variables so that the relaxation of the output constraints is a gentle as possible. When the output constraints are within their limits, the variables s₁ and s₂ take values as zeros. In equation 2, P, Q₁, R₁, Q₂, R₂, M₁ and M₂ are the weighting matrices of appropriate sizes. The prediction horizon for the MPC is denoted by N. The terms $\Delta P_{ca,k} = P_{ca,k} - P_{ca,k-1}$ and $\Delta G_{t,k} = G_{t,k} - G_{t,k-1}$ denote the rate of change of the control inputs.

To solve this constrained optimization problem, *fmincon* solver in MATLAB has been used. For the receding horizon strategy of MPC, only the first control move is applied and the optimal control problem given by equation 2 is re-solved at every sampling time.

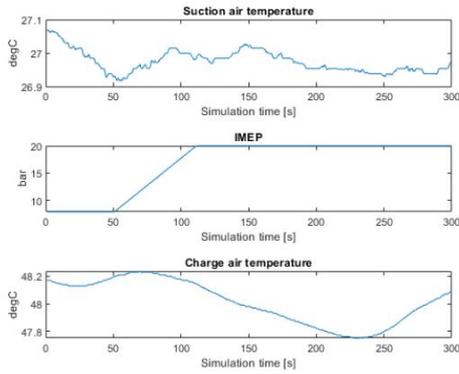


Figure 8. Input disturbances.

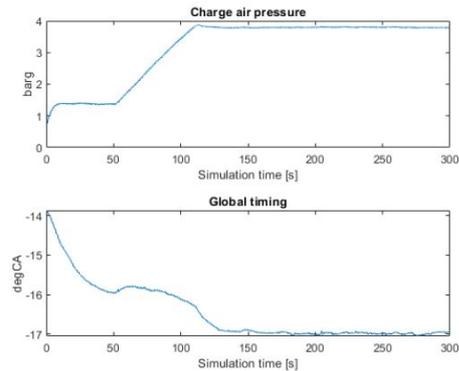


Figure 9. Optimal set points for charge air pressure and global ignition timing under real life disturbances.

5 Results and Discussion

Data from a real power plant operating the B36:45 LNG engine are used for two input disturbances namely for *charge air temperature* and *suction air temperature*. These two input disturbances together with the third disturbance (IMEP) is shown in Figure 8. These input disturbances vary over time and when they act on the system, they can cause the operation of the plant to be far from optimal. In order to compensate for these disturbances, the optimal controller continuously generates new optimal values of the control inputs which are then fed as variable setpoints to the local PID controllers. Figure 9 shows the optimal values of the control inputs (charge air pressure and Global ignition timing) as calculated by the advanced controller.

When these optimal values of the charge air pressure and global ignition timing is applied to the system, the efficiency of the engine is maximized. As already stated above, this can be shown by the minimization of the heat rate as shown in Figure 10.

During the process of minimizing the heat rate, the optimal controller was also able to satisfy the output constraints. The output variables N_{ox} , O_2 and T_e are well within their limits as shown in Figure 11. In order to further test the advanced optimal controller, data from the real power plant containing large variation in the input dis-

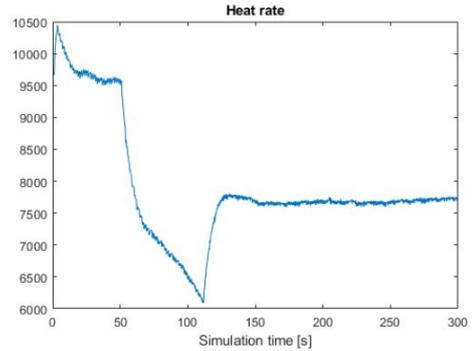


Figure 10. Minimization of the heat rate under real life disturbances.

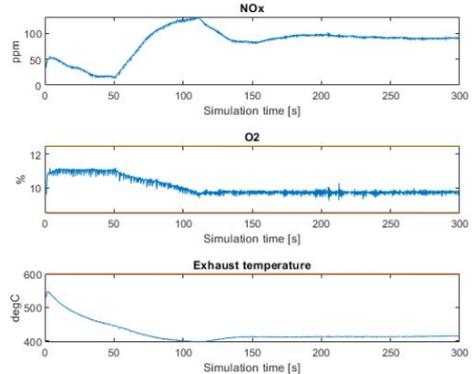


Figure 11. Output variables within their limits under real life disturbances.

turbances was applied. The real life disturbances are given in Figure 12 for all three input disturbances. In particular, the IMEP covers a range of operational windows from lower values to a peak at full nominal power at 1200-1500 seconds before reducing back down to low level again.

Figure 13 shows the simulated optimal values of the charge air pressure and global ignition timing as calculated by the optimal controller (blue line). In addition it also shows the real values of these two variables from the current engine controller operating in the field (red line). Some differences between these two coloured lines,

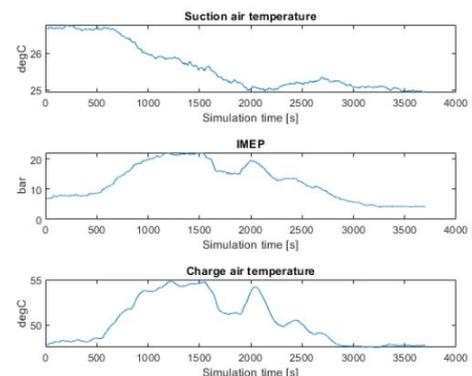


Figure 12. Input disturbances with real life field data.

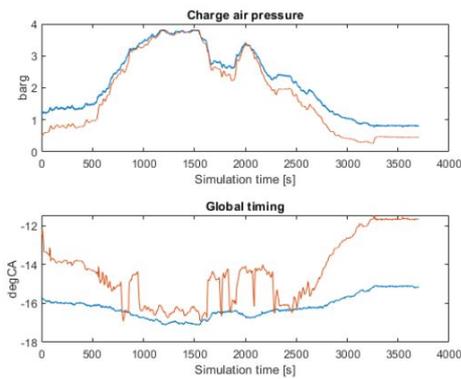


Figure 13. Optimal set points for charge air pressure and global timing for real life field data containing large input disturbance variation. Red line: Measured real values, Blue line: Simulated values.

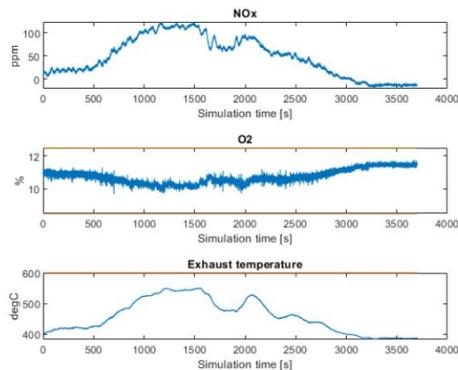


Figure 14. Output variables within their limits for real life field data containing large input disturbance variation.

in particular related to the engine global timing can be observed. The optimal controller utilizes the global ignition timing more than the current controller used in the power plant. This should in principle increase the fuel efficiency. The charge air pressure is also slightly increased to compensate for the increase in burn rate due to the advanced timing generated by the optimal controller. It should be noted that an unlimited possibility of increasing the charge air pressure might not be feasible due to capacity of the turbocharger and a variable upper bound in this value should be established to avoid unrealistic optimal behavior of the optimizer.

At the same time, the output variables N_{ox} , O_2 and T_e are kept within their upper and lower limits in Figure 14 which implies safe operation of the engine. The heat rate output is minimized by the optimizer and the simulated result is shown by blue line in Figure 15. In addition, the measure heat rate from the real field is shown by red line for the same real life input disturbances. It can be noted that values are closely related and share the same form. For the major part of the simulation, the optimized heat rate is also lower than the measured value from the installed engine running traditional control. There is a period around 2250 second mark where the estimated optimize is slightly

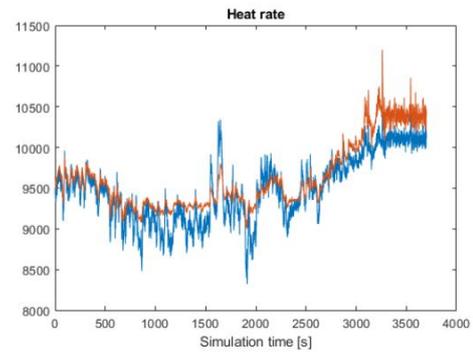


Figure 15. Heat rate minimization for real life field data containing large input disturbance variation. Red line: Measured real values, Blue line: Simulated values.

higher than the measured, but for the majority of the time, the value of the heat rate from the optimal controller is lower. This indicates that the advanced model based controller performs relatively better.

6 Conclusions

The potential of using a model based advanced controller for a B36:45 LNG engine is investigated in this paper. The optimal controller shows promising results in simulations. Compared to the current traditional controller used in the field, the advanced optimal controller could improve the efficiency of the engine. However, the advanced controller requires a dynamic model of the engine, and development of such a model using operational data can be difficult and time consuming. The quality of the results from the optimal controller relies on the quality of the dynamic model. The model used in this paper can be improved in the future and the optimal controller should be tested on a real engine.

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