Simulation-based investigation of a marine dual fuel engine

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Abstract

Recent developments have rendered the Dual Fuel (DF) engines an attractive alternative solution for achieving cost-efficient compliance to environmental regulations. The present study focuses on the safety investigation of a marine DF engine in order to identify potential safety implications. This investigation is based on an integrated engine model, which was developed in GT-ISE™ software and is capable of predicting both the engine steady-state behaviour and transient response. The model includes the engine thermodynamic simulation module as well as the engine control system functional module; the latter is responsible for implementing the ordered load changes and the operating mode switching. The developed model is first validated against available published data and subsequently used to simulate several test cases with fuel changes, from gas to diesel and diesel to gas with rapid and with delayed wastegate valve operation. The derived simulation results are used to investigate the potential safety implications that can arise during the engine operation. The results demonstrate that the engine–turbocharger matching as well as the wastegate control are critical parameters for ensuring the compressor surge free operation during gas to diesel modes transition.

Keywords: marine dual-fuel four-stroke engine; engine and control system modelling; safety investigation

1. Introduction

In the last decade, the maritime industry has been pursuing the reduction of gas emissions driven by the environmental legislation introduced by international and national regulatory bodies. Specific areas have been designated, the so-called Emission Control Areas (ECAs) where stringent limits for NOx and SOx emissions are applied (IMO 2009). At the same time, considerable CO_2 emissions reductions must be achieved by newbuilt ships from 2020 and 2025 (IMO 2011). This renders attractive the use of alternative fuels and propulsions systems including Dual Fuel (DF) engines (Livanos et al. 2014; Trivyza et al. 2018) and batteries (Geertsma et al. 2017), which can be used to meet the regulatory requirements in a cost-effective way.

Considering that the DF engines are currently becoming the industry standard not only for Liquefied Natural Gas (LNG) carriers but also for other vessels (WIN G&D 2017), their continuous design development and optimisation are essential to the marine industry. In this respect, a number of techniques are employed in the engine design phase including experimentation, design, prototyping and engine mathematical modelling (Kyrtatos et al. 2016). Modelling and simulation are considered the most cost effective methods for obtaining a better understanding of the engine operation and the involved interactions between the engine subsystems and components as well as predicting the engine performance and emission characteristics (Geertsma et al. 2018; Stoumpos et al. 2018).

The marine engines typically run under steady state conditions using the same fuel type, although relatively slight power demand fluctuations may occur due to changes in the environmental conditions (air temperature, wind and wave). More considerable power demand changes may occur due to variations

in the operating conditions/modes (e.g. normal sailing vs manoeuvring or berthing), interactions with other power systems on-board or at the user/operator commands. Switching to a different fuel mode needs to be implemented either when the vessel approaches or leaves ECAs or when a failure is present in the fuel systems and their components, i.e. pressure loss of the natural gas fuel supply (Wärtsilä 2015). In this respect, it is essential to assess the behaviour of the engine and understand the interactions between the engine components during both steady and transient conditions, including operating modes changes (Stoumpos et al. 2018).

Apart from ensuring that the DF engines will operate with the greatest possible efficiency and the lowest emissions, it is also necessary to ensure the engine safe operation. Safety is defined as the state where a system operates without causing any harm to humans, environment and assets (Vincoli 2014). As every other system, a DF engine operates with inherent hazards; misfiring, knocking and turbocharger compressor surging may lead to considerable damage to the engine and its components/subsystems (Theotokatos and Kyrtatos 2003; Mavrelos and Theotokatos 2018). In addition, deviations (from the expected ranges) in the engine performance parameters may trigger the engine safety functions, leaving the engine temporarily unavailable. This may lead to system-level hazardous conditions, resulting potentially to a loss of position for the ship or a total blackout; both are associated with a higher risk for a collision, contact or grounding accidents (Bolbot et al. 2018). The use of natural gas increases the potential for fire and explosion accidents (Jeong et al. 2017). Oil mist explosions can lead to hazards such as engine room fires, whilst they may also cause occupational accidents if they occur in close proximity to the vessel operating or maintenance personnel (Cicek and Celik 2013). The NOx and particulate matter (PM) emissions, generated during the combustion process are considered equally harmful for the human health, thus increasing the potential for human deceases in the area of operation (International Agency for Research on Cancer 2012).

The literature addressing the investigation of marine four-stroke and two-stroke DF engines using simulation tools or experiments is limited. Stoumpos et al. (2018) investigated the steady state performance and emissions of a marine four-stroke engine and reported the optimisation of the engine setting for a simultatneous reductions of the CO_2 and NOx emissions. Mavrelos and Theotokatos (2017) presented the investigation of the performance and emissions for a marine two-stroke DF engine of the premixed combustion concept. Georgescu et al. (2016) investigated the transient response of DF and gas engines and discussed the engine operational limitations by using the derived simulation results. Benvenuto et al. (2017) investigated the impact of variable turbocharger technology on a DF engine performance at steady state conditions. (Mayr et al. 2017) addressed the development and implementation of a methodology to simulate the large two-stage turbocharged gas engines at transient conditions. Wang et al. (2015) reported the fuels control system design of a marine DF engine by employing a mean value model for accommodating the effective fuels transitions.

In addition, a limited number of studies with their focus on automotive and heavy duty DF engines can be found in the pertinent literature. Aldawood et al. (2012) investigated an approach to control

combustion in a Homogeneous Charge Compression Ignition (HCCI) engine and presented an applicable and comprehensive control strategy of an automotive natural gas/diesel engine. Doppelbauer et al. (2013) examined the potential of burning compressed natural gas and diesel fuel by performing tests on a four cylinder, commercial vehicle engine and developed a control unit by employing an in-house software. Barroso et al. (2013) modelled a heavy-duty DF compression ignition engine by employing an one-dimensional (1D) model in GT-Power; the model was calibrated by using experimental data and used in both steady state and transient conditions. Xu et al. (2014) developed an 1D model for an automotive four-stroke DF engine in the GT-ISE™ software to study and improve the engine transient response by optimising the engine fuel injection. Fathi et al. (2017) introduced a homogeneous charge compression ignition engine control structure in order for the engine to exhibit acceptable performance and emissions characteristics at steady state and transient conditions.

The preceding literature review reveals a number of research gaps in pertaining literature. More specifically: (a) very few studies have been published focusing on the modelling/simulation of marine DF engines especially investigation of the transient operating conditions; (b) the safety implications analysis in DF engines during transient operation has not been presented in the previous studies, and; (c) the investigation of the control system design parameters on the DF engine safety has not been addressed. An initial study on the investigation of a DF engine safety implications during operating modes switching through simulation and identifying the potential safety implications based on the derived simulation results was presented by the authors in (Theotokatos et al. 2018). The present study is an extension of this paper and includes additional cases investigating the effect of the exhaust gas waste gate response on the engine operation and the occurrence of the turbocharger compressor surging, thus providing additional insights in the engine operation at extreme operating scenarios.

The unique contributions of the present work include: (a) the model-based investigation of the engine operating scenarios that may result in safety issues for the engine and its subsystems/components; (b) the analysis of the DF engine operation that leads to turbocharger compressor surging; (c) the identification of hazards and their causes, and; (d) provision of safety recommendations for the engine design by considering the derived simulation results.

The remaining of this article is organised as follows: The investigated engine description, the model description and its validation are presented in Section 2. The case studies, the derived results and their discussion as well as the safety implications analysis are reported in Section 3. The summary of main findings along with recommendations for the improvement of the system design and suggestions for further analysis are provided in the Section 4.

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2. Engine modelling

2.1. *Investigated engine*

In the present study, the four-stroke, non-reversible, turbocharged and intercooled Wärtsilä 9L50DF engine was investigated (Wärtsilä 2015). The engine is capable of operating in two distinct modes, i.e. (a) the gas mode where the engine runs on natural gas with low pressure gas injection and light fuel oil (LFO) (which is used as pilot fuel for initiating combustion), and; (b) the diesel mode, in which either heavy fuel oil (HFO) or LFO can be used as the main fuel. This engine is an attractive solution for either power generation or ship propulsion (Livanos et al. 2014) due to its high power output, fuel flexibility, reduced emissions, high efficiency and reliability, reduced thermal loading as well as the compliance with the IMO Tier III limits for NOx emissions (in gas mode). In this study, however the examined engine was considered part of a generator set operating at a constant speed of 514 r/min.

The main engine characteristics are illustrated in

Table 1, whilst the engine layout and components are presented in Figure 1. The Engine Control System (ECS) is responsible for the smoothly load changing and operating modes switching. The engine cylinders air−fuel ratio is adjusted via an electronically controlled exhaust gas waste gate (WG), which bypasses a part of the exhaust gas along the turbocharger (TC) turbine (Christen and Brand 2013; Wärtsilä 2015). Each engine cylinder includes a combined diesel and a pilot fuels injector. The gas fuel is injected at each cylinder inlet port (upstream the intake valves) during the engine induction process by using solenoid valves. The gas fuel admission valves as well as the diesel fuel injectors are electronically controlled (in the gas and diesel operating modes, respectively) to regulate the engine power with a target of keeping the engine speed constant. The amount of the injected pilot fuel is also controlled depending
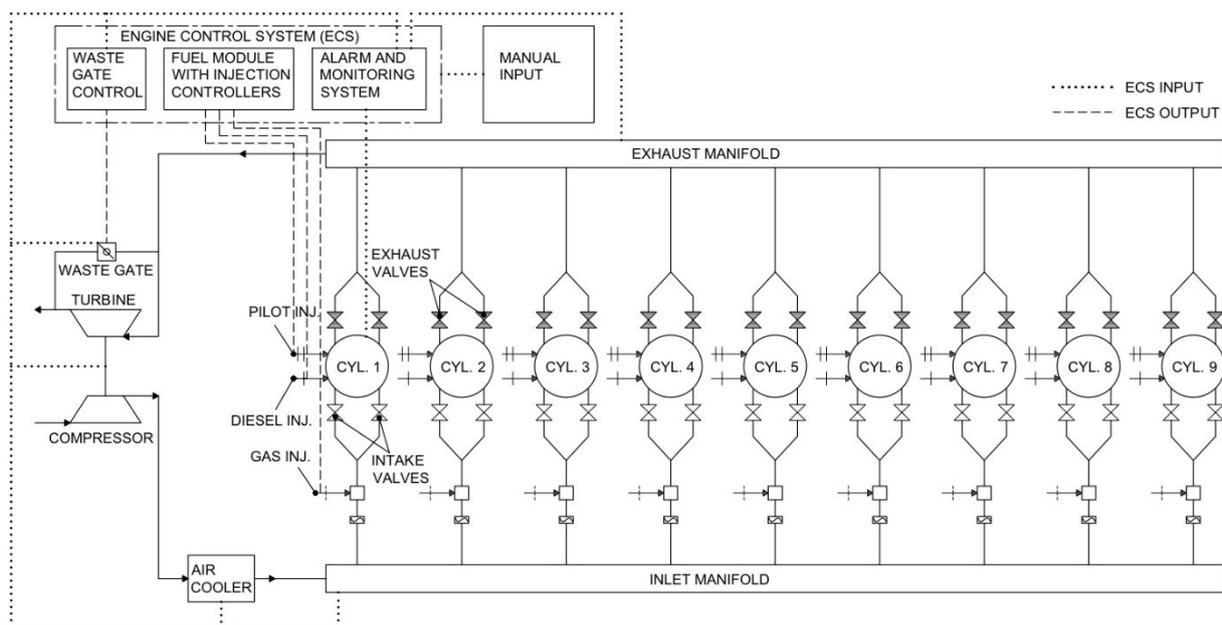

Figure 1 Wärtsilä 9L50DF engine layout

on the engine operating mode and load.

Table 1 Engine main characteristics

Maximum Continuous Rating (MCR) power	kW	8775
MCR speed	r/min	514
BMEP at MCR	bar	20
Break Specific Fuel Consumption at MCR (Diesel mode)	g/kWh	190
Break Specific Fuel Consumption at MCR (Gas mode)	kJ/kWh	7300
Bore	mm	500
Stroke	mm	580
No. of cylinders	-	9
No. of turbochargers	-	1

2.2. *Model description*

The engine model was developed by employing the GT-ISE™ software (Gamma Technologies 2016). The 0D/1D engine model of the investigated engine had previously been developed for steady state conditions as described in Stoumpos et al. (2018). However, the existing model had to be extended to incorporate the modelling of the engine operation at transient conditions with load changes and/or operating mode switching. In addition, the ECS model was incorporated. The integrated engine model was developed by employing the following GT-ISE sub-assemblies: (a) the 0D/1D engine model assembly where the engine thermodynamic modelling is realised; (b) the user input block; (c) the ECS assembly, and; (d) the engine monitors and alarms assembly. For any simulation run, the user needs to specify the engine load time variation, the operating mode (gas or diesel), and the operating modes switching.

A zero-dimensional approach, considering the energy and mass conservation equations, was employed to model the engine cylinders. Two-zones were used for modelling the combustion and expansion processes (the first zone is assumed to contain the unburnt mixture, whereas the second zone is assumed to contain the combustion products), whereas a single zone is considered for the other cycle processes (Merker et al. 2005). The unburned gas zone is assumed to contain the air charge as well as the combustion products from the previous cycle. The burned gas zone is generated following the ignition (Gamma Technologies 2016). Various well established models were employed for modelling the cylinder heat transfer, combustion and friction (Gamma Technologies 2016). The Woschni gas to wall heat transfer model is employed for calculating the respective heat transfer coefficient (Woschni 1967), whereas the engine friction mean effective pressure is estimated according to the Chen-Flynn model (Rakopoulos and Giakoumis 2007). The NOx emissions are calculated by employing the extended Zeldovich mechanism as reported in (Lavoie et al. 1970) and (Hanson and Salimian 1984).

For modelling the engine manifolds pipes and junction elements, an 1D approach was employed, considering the momentum, the mass and the energy conservation equations. For modelling the engine turbocharger compressor and turbine, the respective digitised maps were used. The quasi-steady adiabatic flow equation along with the inlet and exhaust valves profiles (equivalent area versus crank angle) were

employed for calculating the respective mass flow rates. The engine mechanical elements (engine crank shaft and turbocharger shaft) are modelled by employing the respective angular momentum conservation equations for calculating the corresponding rotational speeds.

For the diesel operating mode, the combustion is modelled by using single Wiebe functions, whereas the ignition delay is estimated by employing the Sitkey equation (Merker et al. 2005). Triple Wiebe functions are used for modelling the gas operating mode. Each function is assumed to represent one of the three consecutive combustion phases; in specific: (a) the premixed combustion of a portion of the pilot fuel; (b) the diffusive combustion of the remaining pilot fuel and the rapid burning of the gaseous fuel, and; (c) the cylinder residuals tail combustion (Karim 2015). The ignition delay is approximated by using the data reported in (Sixel et al. 2016) and (Christen and Brand 2013).

The employed Wiebe functions parameters include the fraction, the combustion start, the combustion duration and the shape factors. As part of the mode calibration process, these parameters were tuned at each operating point (25%, 50%, 75% and 100% loads for both the engine operating modes) to match the experimentally measured engine performance parameters. In subsequence, a database was developed to store the calibrated Wiebe functions parameters values, which are considered the controlled parameters for the ECD model. The controlling parameters of the ECS model include: (a) the engine load; and (b) the engine operating mode (diesel or gas).

The developed ECS model assembly, which is shown in Figure 2, includes the gas fuel and diesel fuel control systems as well as the exhaust gas waste gate controller. Figure 2 also illustrates the interconnections between the controlling and the controlled elements. The flowcharts of the control system functions to implement the gas to diesel (GTD) and the diesel to gas (DTG) modes transitions are provided in Figure 3 ((a) and (b) respectively).

The developed ECS model implements the following functionalities:
- The developed ECS model controls the gas, the diesel and the pilot fuels as well as the exhaust gas waste gate valve by employing the respective controllers, thus allowing the engine operation at the following conditions: (a) steady state operation at the diesel or the gas operating modes; (b) transient operation with load changes at the diesel or the gas operating modes; and (c) transient operation with operating modes switching (DTG or GTD).
- According to the engine manufacturer guidelines, the operating modes switching from diesel to gas is not permitted above 80% load due to engine operational limitations.
- The developed ECS model can identify step-wise load changes in the gas mode that exceed the maximum allowed load change (for instance, gas mode load reduction from 100% to 0%). In such scenarios, the engine operation is immediately switched to the diesel mode via fast-acting signals, which control the gas, diesel and pilot injectors as well as the engine waste gate valve.

- The WG valve control is realised by employing a simplified PI controller for adjusting the WG valve area using the engine boost pressure as the controlling variable. The proportional and integral constants of this controller were calibrated by using the Ziegler-Nichols method. The actual ECS controls the engine functions via an advanced automation system that allows optimal running conditions to be set independently of the ambient conditions and the fuel properties (methane number) (Portin 2010). However, it must be noted that in the developed functional control model, the fuel quality and properties were not taken into account.
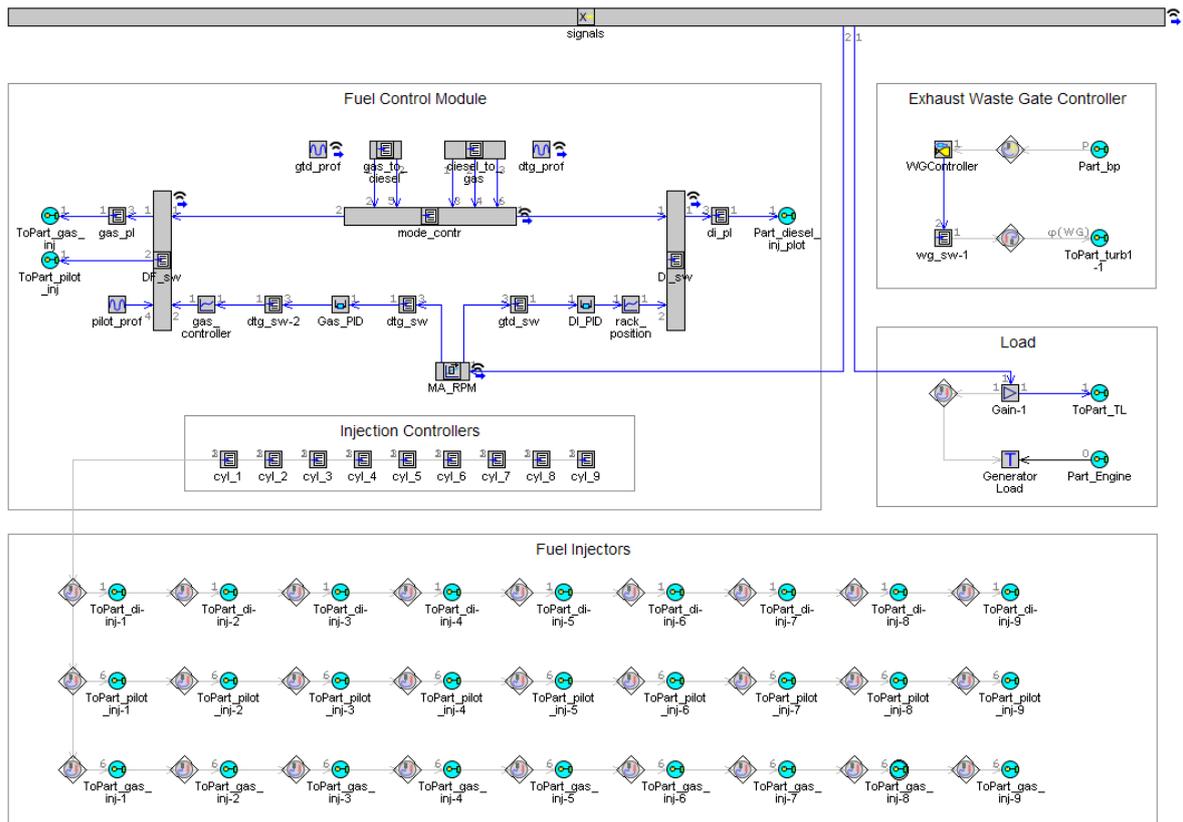

Figure 2 ECS model layout *Abbreviations*; gtd_prof: gas to diesel fuel change transient operation profile; dtg_prof: diesel to gas fuel transient operation profile; DF_sw: logic control switch for gas fuel injection; DI_sw: logic control switch for diesel fuel injection; gtd_sw: gas to diesel feedback switch ; dtg_sw: diesel to gas feedback switch; Gas_PID: PID feedback controller for the gas admission valve; DI_PID: PID feedback controller for diesel governor; pilot_prof: pilot fuel profile; mode_contr: operation mode controller; cyl_n: Engine cylinder injection controller of the nth cylinder; WG Controller: waste gate valve controller; wg_sw: waste gate valve switch; MA_RPM: engine shaft rotational speed
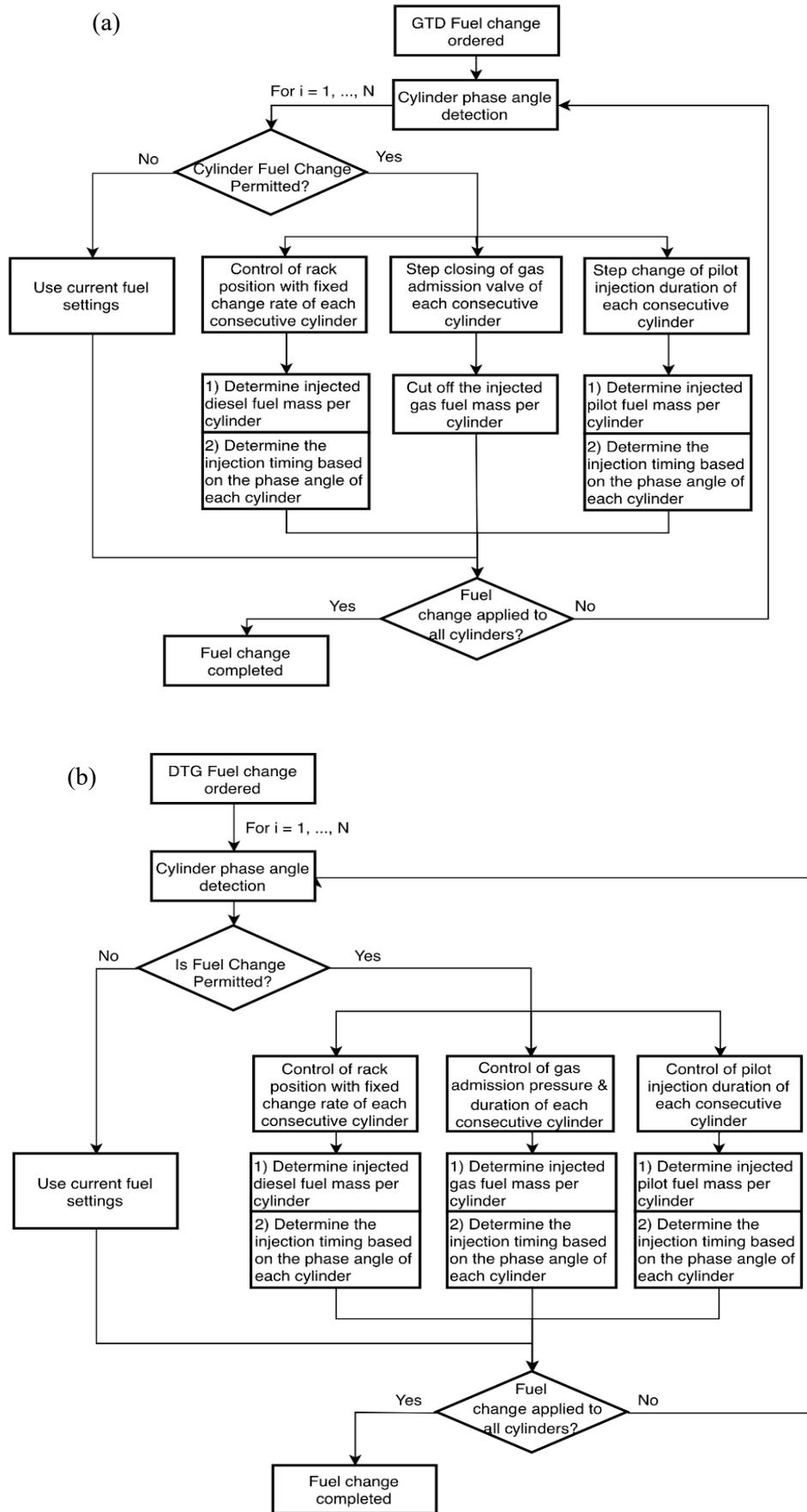
(a)

GTD Fuel change ordered

For i = 1, ..., N

Cylinder phase angle detection

Cylinder Fuel Change Permitted?

No → Use current fuel settings

Yes →

Control of rack position with fixed change rate of each consecutive cylinder

1) Determine injected diesel fuel mass per cylinder
2) Determine the injection timing based on the phase angle of each cylinder

Step closing of gas admission valve of each consecutive cylinder

Cut off the injected gas fuel mass per cylinder

Step change of pilot injection duration of each consecutive cylinder

1) Determine injected pilot fuel mass per cylinder
2) Determine the injection timing based on the phase angle of each cylinder

Fuel change applied to all cylinders?

Yes → Fuel change completed

No →

(b)

DTG Fuel change ordered

For i = 1, ..., N

Cylinder phase angle detection

Is Fuel Change Permitted?

No → Use current fuel settings

Yes →

Control of rack position with fixed change rate of each consecutive cylinder

1) Determine injected diesel fuel mass per cylinder
2) Determine the injection timing based on the phase angle of each cylinder

Control of gas admission pressure & duration of each consecutive cylinder

1) Determine injected gas fuel mass per cylinder
2) Determine the injection timing based on the phase angle of each cylinder

Control of pilot injection duration of each consecutive cylinder

1) Determine injected pilot fuel mass per cylinder
2) Determine the injection timing based on the phase angle of each cylinder

Fuel change applied to all cylinders?

Yes → Fuel change completed

No →

Figure 3 ESC model functional flowchart; (a) GTD fuel change; (b) DTG fuel change.

2.3. Model validation

The engine operation was simulated at steady state conditions for a number of operating points (25%, 50%, 75% and 100% load for both the diesel and the gas operating modes). The performance and emission parameters were calculated and compared with the respective data experimentally obtained from the engine shop tests. The percentage errors between the measured and the predicted parameters are provided in Table 2. From the presented data, it can be inferred that the maximum percentage error is less than 3.5%, thus indicating that the model provides adequate accuracy in all the investigated steady state operating points.

Table 2: Percentage error between the measured and the predicted values at steady state conditions.

	Diesel mode (%)					Gas mode (%)				
Load (% MCR)	100	85	75	50	25	100	85	75	50	25
Brake power	2.6	2.36	1.88	1.14	1.22	-0.42	-1.15	-0.41	1.7	1.34
Maximum cylinder pressure	0.16	-0.6	0.19	0.42	1.77	0.37	0.33	0.51	0.42	0.6
Turbocharger speed	0.04	-0.02	-0.06	-0.79	0.02	0.75	-0.32	-0.9	-0.27	1.14
Brake efficiency	-3.11	-2.9	-2.43	-1.64	-2.22	2.49	3.43	2.32	-1.16	-0.9

For validating the model response at transient conditions, two cases, for which experimental data for the engine load, speed, boost pressure and diesel fuel variations are available in (Ölander 2006), were investigated. More specifically: (a) the engine operation at 100% load at the gas mode and a GTD modes switching (Case 1), and; (b) engine operation at 80% load at the diesel mode and a DTG modes switching (Case 2). The derived time variations of the engine parameters including the normalised rotational speed, the engine load and the normalised fuels amount (for the gas and diesel fuel) along with the respective experimentally measured parameters variations (data for the engine load, the rotational speed, the injected fuel and the boost pressure were only available) for the two investigated cases are presented in Figure 4 and Figure 5. The maximum percentage errors for the engine parameters with available experimental data are provided in Table 3. As it can be inferred from the presented results in these Figures and Table 3, the developed model can predict the engine parameters response with an adequate accuracy; the derived errors were below 2.5% for the majority of the measured parameters apart from the boost pressure, for which deviations 5% and 8% were obtained for the two investigated transitions respectively, and the injected diesel fuel amount only for the diesel to gas transition, for which a maximum deviation of 12.5% was observed. However, as these maximum errors were obtained during the transition period and for only a short time, it can be deduced that the developed model accuracy is sufficient. As all the engine components and processes were meticulously modelled, the prediction of the engine performance and emissions parameters is deemed sufficient for the investigations discussed in the following sections.

Table 3 Maximum percentage of error between the measured and the predicted values at transient conditions.

Parameter	Maximum error [%]	
	Case 1 (GTD at 100% load)	Case 2 (DTG at 80% load)
Brake power	2	2.2
Rotational speed	2.5	2.1
Injected diesel fuel	<1	12.5
Boost pressure	5	8

3. Results & Discussion

3.1. Case Studies Overview

For the investigation of the potential engine safety implications, the following case studies were simulated: (a) gas to diesel (GTD) modes switching at 100% load (Case 1); (b) diesel to gas (DTG) modes switching at 80% load (Case 2). Case 1 and Case 2 were also used for the developed model validation at transient conditions, as discussed in the previous section.

As the results from these two simulation cases indicated that the WG valve control seems to be critical for the engine response, two additional cases were simulated to further investigate the potential engine safety implications as follows: (a) Case 1 with delayed WG valve control (Case 3); (b) Case 2 with delayed WG valve control (Case 4). The simulated case studies characteristics are summarised in Table 4.

For all the investigated cases, a set of the derived simulation parameters time variations including the normalised speed and load, the diesel and gas fuel amount, the boost pressure, the maximum cylinder pressure, the exhaust gas temperature after turbine, the turbocharger (TC) speed, the WG valve opening, the air-fuel equivalence ratio (λ), the engine CO_2 and NOx emissions as well as the compressor operating points trajectory superimposed on the compressor map are presented in Figures 4 and 5. In these Figures, the top two plots correspond to both the presented investigated cases (Case 1 and Case 3, Case 2 and Case 4, respectively), as the WG valve control does not affect the engine speed and fuel controls. The alarm limits for a number of parameters are also indicated in the respective plots in these figures in order to identify responses that lead to potential safety implications for the engine.

Table 4 Investigated fuel change cases.

Simulated case	Description
Case 1	Gas to diesel (GTD) modes switching at 100% load
Case 2	Diesel to gas (DTG) modes switching at 80% load
Case 3	Gas to diesel (GTD) modes switching at 100% load with delayed WG valve operation
Case 4	Diesel to gas (DTG) modes switching at 80% load with delayed WG valve operation

3.2. *Gas to diesel Modes Switching*

For the first investigated case (Case 1), it can be observed from the results presented in Figure 4 that following the GTD modes switching order at the 10.5^{th} s of the simulation run, the gas fuel was immediately cut off, whereas the diesel fuel rack position response, although rapidly increased the injected fuel amount, was not instant. This resulted in the under-powering of a number of engine cylinders for a number of engine cycles, which led to a temporary loss of the engine power (obtaining its minimum value at the 11th s) and the engine speed reduction between the 10.5^{th} s and 11th s. This period is associated with the considerable increase of the air−fuel equivalence ratio (reaching 3.85 at the 10.8^{th} s) and the reduction of the exhaust gas temperature and the maximum cylinder pressure. The exhaust gas temperature drop resulted in a reduction of the turbine inlet exhaust gas energy, slightly reducing the TC speed, which in turn temporarily decreased the compressor air mass flow rate and the engine boost pressure, thus instantaneously driving the compressor operating point towards the compressor surge region. The captured inlet boost pressure drop was also observed in the experimental results. The WG valve quickly closed at the 10th s, immediately after the modes switching order. This resulted in avoiding the compressor surging, which can induce hazardous conditions as it causes TC shaft torsional vibrations of large amplitude and may lead to a potential catastrophic failure of the turbocharger and other engine components.

The fast diesel fuel increase (due to the ECS response to the engine speed drop) resulted in the recovery of the engine power (within 1 s after the time of its minimum value) and caused a notable decrease of the air−fuel equivalence ratio (down to 1.5 at the 11.5^{th} s), which, in turn, led to an overshoot in the exhaust gas temperature (which slightly exceeds the respective manufacturer limit). The observed responses for the air−fuel equivalence ratio and exhaust gas temperature were also affected by the fact that the WG valve was open at the gas mode, and therefore, the air mass flow rate was lower than what was required at the diesel mode. The maximum cylinder pressure reaches its maximum value at the 11th s, which however is below the manufacturer permitted limit, and subsequently quickly restores to the respective steady conditions value.

Exceeding the exhaust gas temperature limit (although for a very short period in the investigated case) is a risky situation that leads to a greater engine components thermal loading and may cause the engine trip, especially when other engine components degradation occurs, such as the turbocharger components fouling. An increased cylinder pressure may lead to extra mechanical loading of the cylinder components and the piston rings, resulting in increased wear and potential blow-by, which in turn causes exhaust gases and unburned hydrocarbons mainly from the cylinder lubricating oil flowing into the engine crankcase, thus increasing the risk for a crankcase explosion (Cicek and Celik 2013). The exhibited λ reduction can lead to potential smoke generation, which may lead to increased soot deposits on the engine components. This in turn may lead to faster components degradation and higher maintenance requirements increasing the engine unavailability.

Following the GTD modes switching, a number of cylinders initially are operating at the gas mode, there is under-powering of a number of cylinders due to the non-instant response of the diesel fuel system, whereas over-fuelling (with diesel fuel) of the engine cylinders occurs for recovering the temporary lost power and speed. The above along with the fact that the air within the cylinders does not match the diesel mode requirements (as the WG valve was open in the gas mode) result in the peak in the NOx emission observed during this switching. In addition, the under-powering of a cylinders number (due to the slower increase of the injected fuel amount) results in the initial reduction of the CO_2 emissions between the 10.5^{th} s and the 11th s of the simulation run. In the subsequent period (between the 11th s and the 12th s), the engine cylinders do not operate with optimal combustion conditions as the fuel and air amounts are varying. This results in the reduction of the engine efficiency the corresponding increase in the CO_2 emissions. As the engine operating conditions (fuelling and air within the engine cylinders) are restored to their respective steady state values, the engine in-cylinder parameters and efficiency are stabilised, and as a result the CO_2 and NOx emissions reach an equilibrium after the 13th s.

As it is observed in Figure 4, the maximum drops in engine speed and power during the GTD mode switching are lower than 5%, whereas the engine speed reaches stable conditions in around 3 seconds after the switching mode order. The engine speed and load responses are in alignment with the requirements set by the classifications societies (DNV GL 2017) according to which, the variation in speed and frequency must be less than 10% and a recovery time must not exceed 5 s. In case of a plant with multiple generator sets, the power output difference between the operating D/G sets is also expected to be less than 15% of their MCR power.

For third investigated case (Case 3) where the WG valve responds in a slower rate, the engine response significantly deteriorates as for few cycles of the engine operation, the compressor surging can be observed. This is a critical engine operation, which apart from the effect discussed in the following paragraph, it also induces significant torsional vibrations of the turbocharger shaft (Theotokatos and Kyrtatos 2003; Leufvén and Eriksson 2013). As it can be observed from Figure 4 (bottom-right plot), the compressor operating point enters the compressor unstable area (left to the compressor surge line) resulting in the occurrence of one compressor surging cycle. The following compressor operating phases are identified: (a) the instant reversal of the compressor air mass flow (from positive flows to negative flows), i.e. air flows from the engine manifold to the ambient via the compressor and the air filter; (b) the mass flow rate increases towards zero flow (the absolute flow reduces); (c) an instant reversal of the flow to positive flows; (d) a restoration period during which the compressor operating point remains within the compressor stable operating area, and; (e) the engine/compressor operation continues till restoring the targeted steady state conditions.

Due to the air flow reversal, the boost pressure further reduces after the mode switching order (in comparison with the respective variation of Case 1), which, in turn, leads to lower λ values (down to 1.25 at 11.5^{th} s) and greater exhaust gas temperature (its peak is observed at 11.7^{th} s). In this λ range, the engine operation at the diesel mode is associated with considerable smoke. The peak in the exhaust gas

temperature also increases considerable exceeding the respective manufacturer limit, which indicates higher thermal loading of the engine components and may cause the engine emergency shout down. This phase of the engine operation is also associated with greater NOx emissions values (its peak was observed at 11.5^{th} s with a value of 2.8 times the NOx emissions of the 100% load engine operation at the diesel mode) and a slightly greater peak value for the CO_2 emissions.

Based on the preceding discussion, it can be concluded that: (a) the GTD modes switching results in a very fast transient, which proves to be challenging for the engine and its systems operation; (b) the engine control systems (components, functions, hardware and software) need to be appropriately designed to satisfy the engine operation requirements, and; (c) the turbocharging system matching as well as the WG valve control, which affect the variation of the engine operating parameters, are crucial for avoiding the turbocharger compressor instabilities and the turbocharger lag effect during engine operation at transient conditions.

3.3. *Diesel to gas Modes Switching*

The derived simulation results for the second investigated case (Case 2) are presented in Figure 5. Following the DTG modes switching order and the quick opening of the WG valve (to match the gas mode conditions), the engine operating parameters exhibited smooth time variations till obtaining their respective steady state values. This is attributed to the fact that this transition takes place within 2 minutes, and therefore, it is much slower in comparison with the GTD modes switching. The TC speed and the engine boost pressure time variations were dependent on the turbocharger system lag and the engine manifold inertias as well as the WG control response (fast opening) resulting in a lower turbine upstream pressure and exhaust gas mass flow rate (i.e. lower turbine gas energy flow), which consequently led to a reduced turbocharger speed. The engine boost pressure variation exhibited a sufficient agreement with the respective measured variation (Ölander 2006) demonstrating that the model sufficiently captures the engine and its components dynamics.

As this engine modes switching is smooth, the derived engine performance parameters (exhaust gas and maximum cylinder pressure) did not exhibit peaks (as occurred in the GTD transition) and their values remained much below the respective manufacturer alarm limits. However, this should be also investigated under the presence of other engine faults such as turbocharger components fouling or cylinders components degradation. Similarly, the cylinder maximum pressure as well as the NOx and CO_2 emissions exhibited a smooth transition towards their steady state values at the gas mode. From these results, it can be inferred that the mechanical and thermal loading of the engine cylinder components is reduced in comparison with the GTD modes transition.
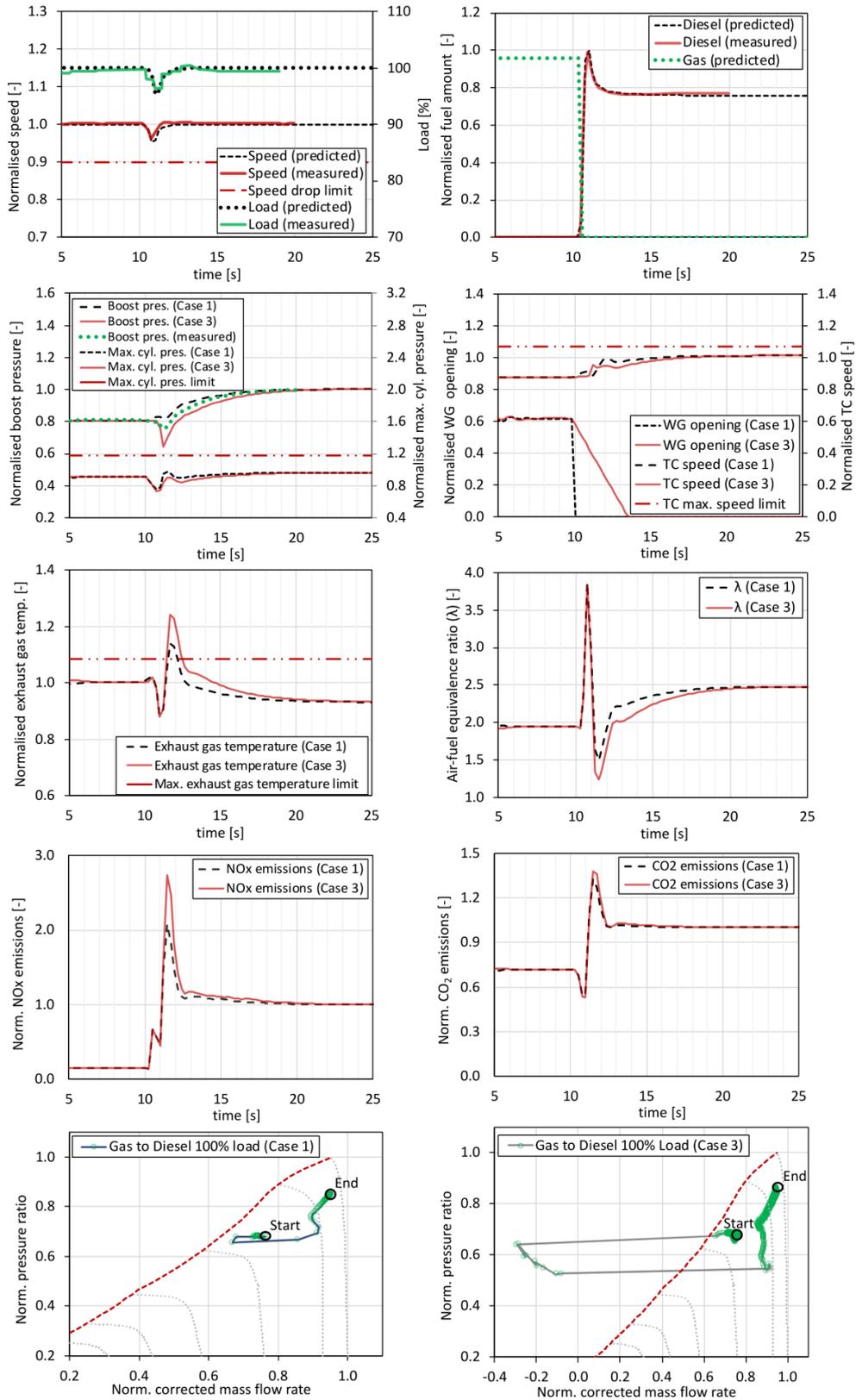
Figure 4 Predicted engine response parameters for GTD modes switching at 100% load with normal and delayed exhaust waste gate valve operation (All the presented parameters except for the equivalence air−fuel ratio are provided in a normalised basis by using the corresponding values of the parameters at 100% load of the diesel mode. The exhaust gas temperature was normalised by using [K]).
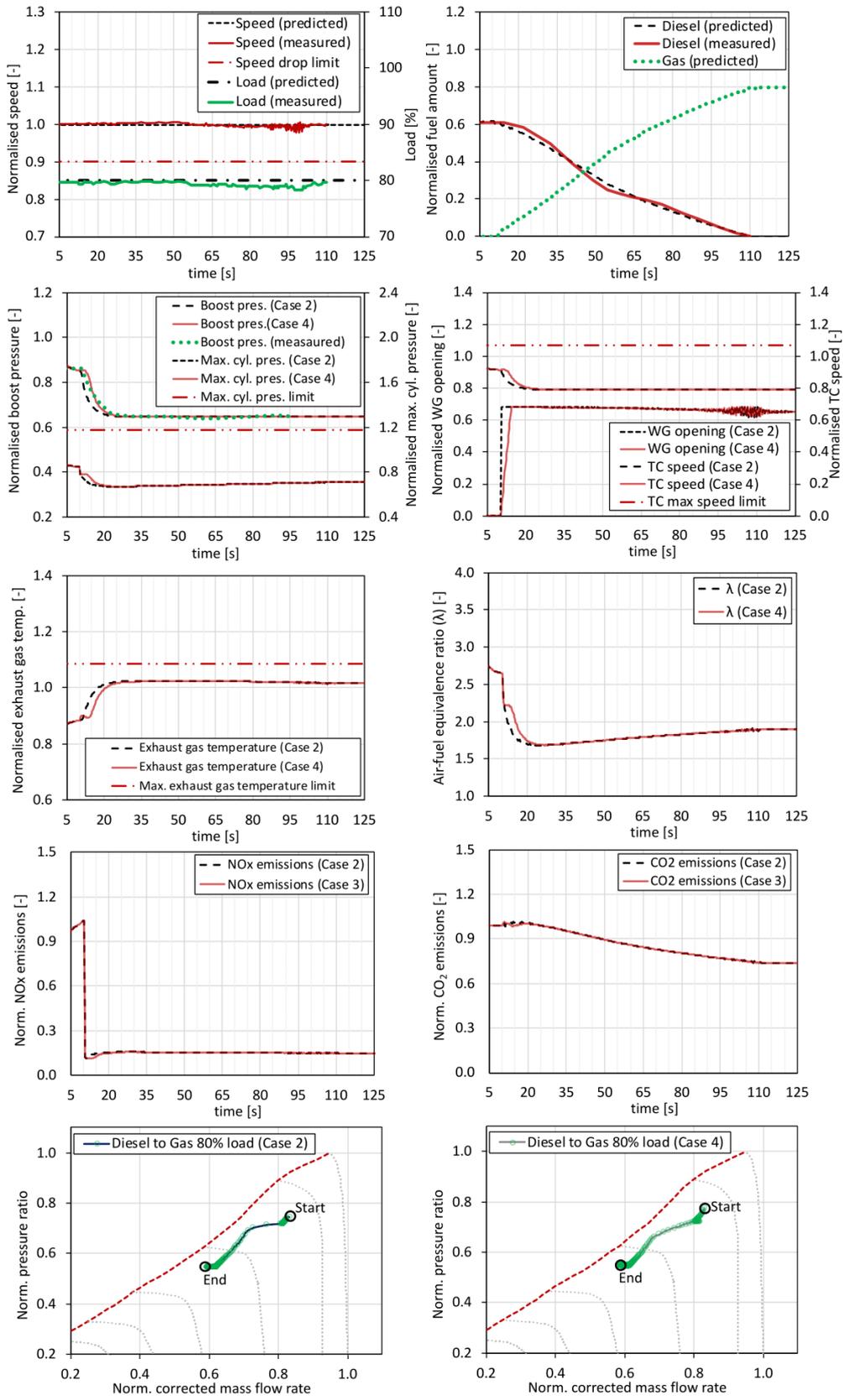
Figure 5 Predicted engine response parameters for DTG modes switching at 80% load with normal and delayed exhaust waste gate valve operation (All the presented parameters except for the equivalence air–fuel ratio are provided in a normalised basis by using the corresponding values of the parameters at 100% load of the diesel mode. The exhaust gas temperature was normalised by using [K]).

As it is shown from the results of Figure 5, slight oscillations occurred in the measured engine load and speed during this transition. These minor oscillations are attributed to the fact that knocking instability occurs in the engine tests; such oscillations are also typical in the gas engines operation. The observed WG valve opening oscillations are not related to the above and are attributed to numerical instabilities of the employed controller; however, these do not affect the derived engine performance parameters as well as the results accuracy.

As the WG control affects the boost pressure time variation, engine cylinders knocking may occur due to low values of the air-fuel equivalence ratio (reaching around 1.7 at the 20th s). According to the information provided by the engine manufacturer (Wärtsilä 2009), λ values lower than 1.8 may result in knocking for the investigated engine load (80%) at the gas mode. Therefore, to avoid such conditions and provide a stable, secure and reliable operation, in terms of engine performance, it is recommended to implement a limiter confining the WG valve maximum opening value during the DTG modes switching.

For the case where the WG valve opened more slowly (Case 4), delayed responses (of around 4 s) were exhibited from the engine and turbocharger performance parameters, which also caused small delays in obtaining the respective parameters steady state values. The slower opening of the WG valve resulted in the slower reduction of the exhaust gas energy entering the turbocharger turbine, thus retaining for a longer period the engine turbocharger speed and consequently the engine boost pressure. However, following the WG valve opening, the exhaust gas energy to the turbine is considerably reduced, causing the drop of the turbocharger speed, the engine boost pressure as well as the observed variations of the other performance and emission parameters. As in Case 2, smooth parameters variations were predicted (after the opening of the WG valve), which indicated that the DTG transition with delay WG valve operation does not result in arising additional safety implications. However, malfunctions in the ECS and actuators may result in incorrect fuel amounts supplied to the engine cylinders leading to uneven cylinders load sharing, which can be exacerbated during transients, and thus should be further examined.

Based on the preceding discussion, it can be inferred that: (a) knocking may occur if λ value is not properly controlled; (b) the DTG modes switching is a smooth relatively stable and other implications to the engine safety are not expected unless other engine component failures are present.

3.4. *Safety implications during GTD and DTG modes switching*

The previous simulation results and their analysis supported the identification of the potential safety implications that can arise during the investigated engine transient runs. The identified implications are presented in Table 4, which follows a format similar to a Preliminary Hazard Analysis (PHA) table (Vincoli 2014). The results demonstrate that during the GTD modes switching, the number of potential hazards is significantly higher than the one in the case of the DTG modes switching. In the former case, the most critical situations is associated with the turbocharger compressor surging as the GTD engine modes switching involves much faster dynamics. During the DTG transition, the potential hazards are associated with low λ values, which can lead to knocking in the engine cylinders.

Table 5: Potential safety implications during transient for two investigated cases.

a/a	Hazard	Caused by	Effect	Recommendation
1st investigated case GTD modes switching	Compressor surging	TC lag / matching at different conditions in the gas and diesel model with WG valve open/close, respectively	Lower air entering the cylinders / lower λ values and greater exhaust gas temperature / smoke and engine thermal loading Fluctuating TC shaft torque / TC shaft torsional vibrations	Proper design of WG valve control Appropriate TC matching Appropriate fuel injection timing
	High smoke emissions	Low λ values	Deposits formation leading to increased wearing of engine components	Proper design of waste-gate valve control Appropriate TC matching Appropriate fuel injection timing
	Oscillatory thermal and mechanical stresses	λ fluctuations	Fatigue problems	Proper control of injection timing
	D/G set Trip	High exhaust gas temperature	Power unavailability leading to ship level hazards	Monitoring of engine parameters Selecting alarm settings ignoring temporal temperature variations
	Piston rings blow-by and potential crank case explosion	High maximum cylinder pressure	Damage to the engine and cylinder components	Proper control of injection timing
2nd investigated case DTG modes switching	Knocking	Low λ values	Damage to the engine cylinders	Appropriate TC matching WG valve opening limiter

4. Conclusions

This study investigated the potential safety implications for a large marine four-stroke DF engine operation with focus on switching between the engine operating modes from diesel to gas (DTG) and from gas to diesel (GTD). A model was employed for the simulation of the engine and its control system capturing the engine steady state operation as well as the engine transient operation with load changes and modes switching. The engine operation during the GTD and DTG modes switching for the cases of normal and delayed WG valve control was simulated and the analysis of the derived simulation results was employed for identifying critical situations along with recommendations for enhancing the engine safety.

The main findings of this study are summarised as follows.

- The developed model adequately captures the engine transient response indicating the effectiveness for the engine thermodynamic modelling as well as the engine control system functional modelling.

Thus, it can be used with fidelity for simulating engine operation scenarios including extreme/fast transient cases.

- The GTD mode switching is a rapid transient that must be completed within 3 s and therefore, has a profound effect on all the engine operational parameters resulting in a number of potential hazards including compressor surging, smoke, fluctuating mechanical and thermal stresses in the various engine components.

- For the GTD mode switching, the turbocharger compressor surging can occur due to delayed response of the WG valve caused by a faulty controller operation or a degraded/faulty performance of the WG valve actuator and/or its electric motor.

- As the DTG mode switching is slower (compared with the GTD transition) taking place within 2 min, the engine operating parameters demonstrated a smooth time variation. However, knocking in various engine cylinders may occur due to the air-fuel ratio variation and limitations of the engine operation within a window.

- Although the WG valve control has only slight influence on the engine operation at the DTG mode switching, the WG valve opening limiter is deemed as essential for avoiding compressor surging.

The following safety recommendations were provided and can be used during the design and operation phases of marine DF engines:

- To avoid problems with compressor surging during the engine modes switching, the WG valve control system must be designed considering the optimal WG valve response rate to accommodate both the GTD and DTG transitions.

- It is important to ensure the appropriate matching of the turbocharger with the engine for both operating modes for minimising the turbocharger lag effect during modes transitions and avoiding the compressor surging and its effects when faults in the WG valve control are present.

- Appropriate monitoring of the engine components health status including the WG valve, the turbocharger fouling and the engine cylinders condition is required to avoid the engine trip and emergency shut down, especially during the GTD modes switching.

- The engine alarm settings must be able to ignore temperature spikes in measurements before tripping the engine during the GTD transition.

- Although faster diesel fuel injection control cannot be achieved due to the limitations of the diesel fuel system mechanical components, the complete engine control may be revisited and improved to reduce the combustion instabilities during the GTD modes switching (in this way, mitigating the effects of the mechanical and thermal stresses fluctuations as well as the deposits formation).

- The engine control system must keep the air−fuel ratio values at adequately high levels during the DTG transition to avoid potential knocking in the engine cylinders.

The developed engine and its control system model proved to be a useful tool for obtaining a better understanding of the engine processes and components interactions as well as for investigating potential engine safety implications. It supported the identification of the critical importance of the waste gate exhaust gas valve control, which is required for enhancing the engine safety. It is expected that approaches that use simulations and detailed models to identify hazardous situations in complex maritime systems will become more popular for future studies.

5. Acknowledgements

6. Abbreviations list

0D	Zero-dimensional
1D	One-dimensional
BMEP	Brake Mean Effective Pressure
CO_2	Carbon Dioxide
DF	Dual Fuel
D/G	Diesel Generator
DTG	Diesel to gas fuel modes switching
ECA	Emission Control Area
ECS	Engine Control System
EEDI	Energy Efficiency Design Index
GTD	Gas to diesel models switching
HFO	Heavy fuel oil
IMO	International Maritime Organization
LNG	Liquefied Natural Gas
LFO	Light fuel oil
MCR	Maximum continuous rating
NOx	Nitrogen Oxides
PHA	Preliminary Hazard Analysis
PI	Proportional - Integral
SOx	Sulphur Oxides
TC	Turbocharger
WG	Waste gate
Λ	air–fuel equivalence ratio

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