Measurement and analysis of the in-cylinder process of a LNG engine under transient operations

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Abstract. Parameters reflecting the in-cylinder process in a turbocharged, LNG fuelled, spark ignited engine, such as the intake manifold pressure, air/fuel ratio (AFR), fuel mass flow-rate, as well as the dynamic in-cylinder pressure traces, are measured cycle-by-cycle for a series of transient operations via external air injection into the engine intake system. During the transient load step process, the engine experiences not only abrupt change of boost pressure and AFR, but also rapid engine load change. The changes in the in-cylinder combustion process during the transient operations, which were rarely revealed before, are illustrated with the rate of heat release (ROHR) characteristics cycle-by-cycle. The influences on the combustion characteristics under the transient operation conditions, of the operational parameters, such as AFR, air and fuel flow, boost pressure, etc. are examined and discussed. Existence of abnormal combustion such as late combustion, misfiring and severe variation of IMEP, etc. are pointed out. The possible causes of those abnormal combustion phenomena are also analyzed. The in-cylinder process detection method presented in this paper is universal and can be extended to other similar applications.

1. Introduction
Automotive engines generally work under transient operating conditions: i.e. the engine speed, load, AFR, turbo boosts pressure, spark/injection timings, etc. all change simultaneously. On the other hand, engine operating parameters are often calibrated on engine dynamometer under steady-state operating conditions. Engine operating and controlling parameters for transient operating conditions are often determined through interpolation among the stored calibration maps. Due to the difficult nature of a transient operating condition, and also lack of appropriate transducers or approaches, the current research mostly focuses on the engine performance under steady-state conditions[1-3] and the knowledge to transient characteristics for the engine is still rare[4-5].

LNG, as an alternative fuel, is now referred as a "clean" fuel and is predicted to have more and more application in the near future because LNG has not only higher octane number than that of gasoline but also its lower carbon content. Also, when LNG is injected into the cylinder, it is vaporized into the gaseous state, which promotes the mixing of air and fuel. Moreover, it produces much less PM emissions compared to diesel, less HC and CO emissions compared to gasoline, and also higher compression ratio could be adapted for higher thermal efficiency. However, in spite of its
many advantages over the gasoline or diesel counterparts, in the past LNG engines have mostly been modified from either their gasoline or diesel cousins, very few have come from a clean sheet design.

Facing with more and more stringent regulations, some technologies are applied in LNG engine, such as lean burn [6-7], hydrogen addition [8-9] and exhaust gas recirculation (EGR) [10-11]. In recent years, many investigations have been conducted on their effects on engine performance. Navarro et al. [12] compared the emission and performance of engine fueled by pure natural gas, pure hydrogen and their blends with different hydrogen fraction. It is found that hydrogen addition can decrease the CO2 emission and extend the lean burn limit. Korb et al. [9] studied the effects of hydrogen addition on the combustion process in a lean burn natural gas engine. The results show that hydrogen addition could enhance combustion and avoid misfire. Moreover, hydrogen addition increases NOx emission while leaner burn could compensate the increase of NOx. Duan et al. [13] investigated the effects of four kinds of EGR system on the performance of the natural gas engine with hydrogen addition. It is found that high-pressure EGR leads to the lowest NOx emission and higher EGR ratio causes lower peak combustion pressure except the internal EGR system.

As mentioned above, a lot of findings on the effects of different parameters on the performance of LNG engine under steady-state conditions have been accumulated in previous investigations. According to the previous studies in gasoline engines [14-15], there exist differences for performance of gasoline engines under steady-state and transient conditions. However, less research and study is dedicated understanding the performance and controlling parameters for LNG engine under transient conditions. The knowledge to the combustion process and engine performances under transient conditions is far behind those in gasoline or diesel engines.

Presented in this paper are the results of a research program to improve the dynamic load response performance of a LNG fueled heavy duty engine.

Through adding compressed air into the intake system, the engine would undergo an abrupt increase of intake boost pressure. Depending on where the intake pressure boosting device is located, the engine's fuel supply rate may or may not follow the air flow-rate increase induced by the boost air, i.e. the instant air fuel ratio may experience an increase (if the fuel flow does not follow the air flow), decrease (if the fuel flow increases at a higher rate than the air flow), or does not change (if the change of the fuel flow matches with that of the air flow). Engine operating and performance parameters are recorded during the load step transient process, together with the in-cylinder pressure traces. The in-cylinder combustion process, represented by the typical characteristic parameters such as the combustion duration (which is typically represented by the duration between the time when 10% fuel is burned and 90% fuel is burned), the 50% burning location, the ROHR curve, etc. are processed from the measured in-cylinder pressure traces, and analyzed together with the other parameters. The purpose of this study is focused in the following two aspects:

1: To understand why the air boosting location upstream of the air/fuel mixture is superior to the downstream location;
2: How the in-cylinder combustion characteristics of a LNG fuelled engine change with engine operating parameters, and how to use those to setup engine performance simulation models.

2. Experimental investigations
Both steady state and transient tests are conducted on an electric engine dynamometer. The steady state dynamometer tests are focused on the influences on engine performance of the major operating parameters such as speed, load, intake boost, relative AFR, etc. while the transient load step tests, due to the difficulty in measurement of many parameters, are carried out to examine whether the relationship maintains as the steady state operations. If so, the conclusions drawn from the steady state tests could be applied.

3. Engine tests
Depicted in Figure 1 is the schematic of an intake air boosting system attached to a LNG fuelled heavy duty engine. A separated air loop is used; one end is connected to the compressed air stored in an air

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tank of 0.3 m$^3$ volume. The air pressure in the tank is pre-charged to approximately 7 bar abs. The exit end of the air tank is connected to the engine intake system through a pressure regulator and an electric control valve. The pressure regulator controls the air pressure fed to the engine, and the electric control valve can be opened or closed at any desired time and duration to feed the compressed air to the engine intake system to boost its intake pressure. Several entry locations of the air boosting device to the engine intake system are screened, including locations of upstream and downstream of the air/fuel mixer. It should be noted that when the air booster pipe entry is located upstream (the green circle in Figure 1) of the air/fuel mixer, the added air will flow through the air/fuel mixer before it enters the engine's intake manifold, while if the air booster pipe entry is located downstream (the red circle in Figure 1) of the air/fuel mixer, the pressure wave and the airflow it generates could go in both directions: i.e. 1) it goes into the intake plenum then to the intake port, and 2) it passes the air/fuel mixer towards the engine's compressor/air entry, which is in the direction of against the normal air flow. As shown later, the location of the air booster entry pipe (upstream or downstream of the air/fuel mixer), would lead to significant differences in the AFR profile during the transient load step operation.

The engine used in this study is a heavy duty LNG engine with the major parameters specified in Table 1.

![Figure 1. Schematics of an intake air boosting system.](image)

| Engine type | Bore (mm) | Stroke (mm) | Connecting rod length (mm) | $\varepsilon^a$ (-) | Rated power (kW) | Maximum torque (N m) | Fuel system |
|-------------|-----------|-------------|-----------------------------|------------------|------------------|----------------------|-------------|
| In-line 6 cylinder spark ignited, Turbocharged, LNG engine | 126 | 130 | 219 | 11:1 | 250 | 1350 | Single point injection |

*a*The compression ratio.

Dynamic pressure transducers are mounted to the cylinder #1 and #4. Fast response static pressure and temperature sensors are mounted at the interested locations along the engine intake/exhaust runner and manifold systems. The engine speed, torque, fuel flow-rate, engine intake/exhaust pressure and temperature at interested locations are sampled with a frequency of 10Hz, while the in-cylinder pressure traces are measured with a 0.2° CA interval.
4. Results and discussions

4.1. Steady state results

The engine was tested for steady state operation conditions first, in which the engine load and speed were varied in a systematic manner: the engine speeds were tested from 600 rpm (idle speed) to 2200 rpm (rated speed) with an interval of 200 rpm, while the engine loads were tested with a 1.0 bar brake mean effective pressure (BMEP) interval in the range of 0.0 to 6.0 bar, and 2.0 bar interval for the range of 6.0 bar to full load. The processed engine burning characteristics data from the measured cylinder traces are depicted in Figure 2. It should be noted that in order to eliminate the effect of pump gas loss, the abscissa in the figure is expressed by net mean effective pressure (NMEP). So it could be concluded:

- Unless at very low load, the 10-90% burning duration is very insensitive to the engine load, although it shows a slightly increasing trend with engine speed, it is only marginal.
- Similarly, the 50% burning location is also insensitive to the engine load. Because the knock tendency decreases with increasing engine speed, the spark timing is advanced with the increase of engine speed, therefore with increased engine speed the 50% burn location is closer to the top dead center (TDC).
- The positions of the peak cylinder pressure (PCP) and the 50% burning location have an almost linear relationship, which suggests that the PCP location is controlled by the 50% fuel burnt location. It should be mentioned and also important to understand that the indicated thermal efficiency is strongly influenced by the PCP location, or the 50% burning location because the latter determines the effective expansion ratio remaining for the combustion heat released to be converted into mechanical work.

Depicted in Figure 3 is the measured 10-90% burning duration as a function of relative AFR at various speeds. As demonstrated, the combustion duration increases with the relative AFR. However, under the same AFR, the combustion duration only experiences a marginal increase with engine speed.

![Figure 2. In-cylinder combustion characteristics as a function of engine load and speed, steady state conditions.](image-url)
The data depicted in Figure 2 and 3 provide very useful guidelines on how to set up combustion models when conducting engine performance simulations for various engine load and speed conditions, which could only be acquired from experimental work.

![Figure 3](image.png)

**Figure 3.** 10-90% burn duration as a function of relative AFR, steady state conditions.

### 4.2. Transient testing results

The transient load step tests were conducted for the engine speeds of 800 rpm, 900 rpm and 1000 rpm and full load conditions. Those lower engine speeds were chosen because the engine’s turbocharger could not provide preferred boost and additional air boosting would significantly improve engine’s performance. The tests were first conducted with the air boost pipe entrance located at the upstream of the air/fuel mixer. To simply data analysis, all the tests were conducted under the "constant speed mode", i.e. the dynamometer is set up in the way that during a test the engine was kept at an almost constant speed.

Depicted in Figure 4 are the transient load step results at a typical engine speed of 900 rpm. In this case, the engine intake pressure is almost instantly boosted from 1.3 bar to 1.6 bar abs. Since the air mass flow-rate is directly proportional to the intake plenum pressure [8], the air mass flow-rate through the intake port is estimated to increase by approximately 25%, while the fuel flow-rate increases by about 30% from approx. 19 kg/h to 24.6 kg/h. The increase as a percentage of the fuel flow-rate is greater than the air flow-rate; therefore the relative AFR experiences a slight decrease.

Along the transient process, the indicated mean effective pressure (IMEP) of the engine increases by about 30%, which corresponds to the fuel flow-rate change. This suggests that the indicated thermal efficiency is kept unchanged. Indeed, the combustion duration and the location of the 50% burn change nominally. A closer look at the 10-90% burning duration will reveal a slight reduction with the decrease of the relative AFR during the transition, together with a very slight advance of the 50% burn location. Since the relative AFR, 10-90% burning duration, 50% burn location all experience little change, the indicated efficiency is also kept almost unchanged.

In contrast to Figure 4, when the air booster pipe entrance is placed at the intake plenum, which is downstream of the air/fuel mixer, a quite different phenomenon is observed, as demonstrated in Figure 5. Unlike the results shown in Figure 4, the fuel flow-rate decreases first when the intake air pressure is boosted, as highlighted by the broken line formed box. The exact reason is unknown with regard to why the fuel mass flow-rate decreases initially at the beginning of this transient process. One possible theory is that since the air pressure booster is located downstream of the air/fuel mixer, the pressure wave generated by the compressed air may somehow interfere with the fuel mass flow-rate through the air/fuel mixer. As time progresses, the fuel mass flow-rate will eventually pick up and reach about 4% of increase over the value before the boost. In this case since the air flow significantly increases while the fuel flow only experiences a marginal increase, the relative AFR increases proportional to the air flow increase. As shown in Figure 5, between the Cycle No. 20-40, the relative AFR first increases from about 0.98 to 1.34, then settles at a value of approximately 1.2.
Figure 4. Results under transient operation, air booster located upstream of the air/fuel mixer, 900rpm.

Figure 5. Results under transient operation, air booster located downstream of the air/fuel mixer, 900rpm.

Corresponding to the changes in the relative AFR, the combustion characteristics experience significant changes: 1) the 10-90% burning duration increases significantly with the increase of AFR; 2) the 50% of burn location is significantly delayed. As a result, the in-cylinder thermal to work conversion efficiency deteriorates, as indicated by the indicated thermal efficiency which is also shown in Figure 5.

The changes in the in-cylinder burning characteristics and the resulting cylinder pressure traces corresponding to the variations in Figure 5 are demonstrated in Figure 6 with individual cycles. In the
cases of before cycle No. 20, the change in boosting pressure does not start yet, the in-cylinder process demonstrates quite a stable nature, some cycle to cycle variations of the cylinder pressure traces are still seen, which is quite normal since those are individual cycles. In the Cycle No. 21-30 plot, when the boosting process starts, the in-cylinder AFR increases significantly, the mixture gets considerably leaner, and the combustion process becomes slower. This is accompanied with the later and later PCP location and the lower PCP value, as indicated by the direction of the arrow in the plot, which are caused by the longer 10-90% burning duration and the later 50% burning location.

In the case shown by the Cycle No. 31-40 plot in Figure 6, the fuel flow-rate starts to increase in Figure 5, and the in-cylinder AFR starts to drop. The 10-90% fuel combustion duration then reduces and the 50% burn location starts to advance, together with the partial recovery of the in-cylinder IMEP. However, since the AFR is still higher than before, the 10-90% burning duration is longer, with later 50% burning location and lower IMEP(even with higher fuel flow), therefore a lower in-cylinder thermal efficiency is produced, Figure 5.

**Figure 6.** Results under transient operation, air booster located downstream of the air/fuel mixer, 900rpm.  
**Figure 7.** Results under transient operation, air booster located downstream of the air/fuel mixer, 1000rpm.
Figure 8. Results under transient operation, air booster located downstream of the air/fuel mixer, 1000rpm.

An extremely lean combustion case and the associated parameters are shown in Figure 7 and 8. In this run the air/fuel mixture gets very lean with the relative AFR reaches as high as 1.5. The elevated AFR significantly slows down the combustion process, with significantly longer 10-90% burn duration, and later 50% burn location. What could also be observed in Figure 8 is that the deviation of the IMEP values from its mean value significantly increases with the increased AFR, which indicates a severe instability of the combustion process, accompanied by the significant variation of the IMEP. This is confirmed by the burning rate profiles and cylinder pressure traces shown in Figure 7. The Cycle No. 2-20 plot represents a steady state operation, where all the operating parameters are fairly stable. The Cycle No. 26-35 plot demonstrated how the in-cylinder burning process deteriorates with leaner and leaner mixture, and how the cylinder pressure traces follow the continuous change of the relative AFR cycle-by-cycle. The Cycle No. 61-70 plot demonstrates that, although the transition is more or less settled in terms of intake pressure, fuel flow-rate and AFR, the in-cylinder combustion process are rather unstable due to the very lean mixture.
5. Conclusions
The following conclusions could be drawn from this study:

a) The in-cylinder process during a transient load step could be better understood by the cycle-to-cycle monitoring of cylinder pressure traces and ROHR curves.

b) Similar to the gasoline combustion process, the in-cylinder burning process of the air/LNG mixture is also mainly controlled by the relative AFR. Other parameters, such as intake pressure, speed, load, are of secondary influence.

c) With regard to the relationship between the combustion characteristics and the AFR of the mixture, steady state and transient operations make no difference, which means the knowledge gained at steady state conditions also applies for the transient process, and the instant AFR is the key parameter to command for.

d) The general trends and data summarized in this study could be utilized to set up combustion heat release models for LNG engine performance predictions under different speed and load conditions.

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Definitions/Abbreviations

NMEP Net mean effective pressure (net)
IMEP Indicated mean effective pressure (gross)
PMEP Pumping mean effective pressure
BMEP Brake mean effective pressure
RGF Residual gas fraction
ROHR Rate of heat release
LNG Liquefied natural gas
AFR Air/fuel ratio of the in-cylinder mixture
BDUR Burning duration
CA Crank angle
TDC Piston’s top dead center
Mfuel Mass flowrate of fuel
PCP Peak cylinder pressure
Pcyl Pressure of cylinder