COMBUSTION CONTROL IN GASOLINE HCCI ENGINE WITH DIRECT FUEL INJECTION AND EXHAUST GAS TRAPPING

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Abstract

Homogeneous charge compression ignition (HCCI) seems to be the most promising solution for gasoline engines in the light of future emissions regulations. This novel combustion technique allows for significant reduction of fuel consumption and engine-out NO_X emissions at low and medium engine load/speed conditions. High heat release rate enables realization of the Otto cycle close to ideal, increasing thermal efficiency. Among different approaches to invoke an auto-ignition of air-fuel mixture, exhaust gas trapping with the use of a negative valve overlap is under intensive investigations.

The paper presents research results of controlling an auto-ignition and combustion phasing in a single cylinder gasoline engine with direct fuel injection operated in the negative valve overlap mode. The experiments were performed at variable valvetrain settings, providing a control of EGR rate and volumetric efficiency. Additionally, the combustion process was investigated at variable air-fuel ratio. It was found that volumetric efficiency and EGR rate are mainly dependent on exhaust valve timing, while a timing of intake valve determined combustion on-set and its duration. The effects of EGR rate and air-fuel ratio on combustion timing and exhaust gas emissions were isolated. The direct fuel injection showed its benefits versus mixture formation outside the cylinder. The application of variable injection timing provided additional possibility to control the combustion timing and exhaust emissions. However, it was found that the fuel injection strategy should be related to the engine load conditions.

Keywords: *homogeneous charge compression ignition, variable valve timing, negative valve overlap, gasoline, direct injection*

1. Introduction

Homogeneous charge compression ignition (HCCI) in gasoline engines allows for significant reduction of fuel consumption and engine-out NO_X emission at low and medium engine load/speed conditions. However, an auto-ignition of air-fuel mixture requires much higher temperature levels than one which than can be obtained with low compression ratio, typical for spark ignition engines. The in-cylinder temperature at the moment of auto-ignition exceeds 750 K [7]. There are two commonly used methods which allow a realization of HCCI process in the gasoline engines. The first approach is to use elevated compression ratio combined with intake air preheating [1, 2, 9, 10]. The second approach, more feasible for production SI engines, is to provide high enthalpy of the in-cylinder load without change of compression ratio. This can be achieved via a high dilution of a fresh mixture with hot exhaust gases. The most effective way which enables internal exhaust gas re-circulation (EGR) is an exhaust trapping by negative valve overlap (NVO) and reduced lift of the valves [3, 6, 12, 15, 16].

Recent investigations into development of HCCI engines with conventional compression ratio are focused on a control of combustion on-set with the use of variable valve timing (VVT) and air excess ratio (λ) [8]. Volumetric efficiency of the engine is dependent on exhaust valve closing (EVC) timing in much higher extent than on intake valve opening (IVO) [3, 16]. Volumetric efficiency and EGR rate are oppositely related resulting with almost constant in-cylinder mass at operation of the engine with VVT [3]. Variability of IVO angle changes the thermodynamic compression ratio of the engine, thus influencing auto-ignition timing [11]. The engine load control via air excess ratio has an impact on EGR rate, as increase of λ provides drop of exhaust temperature, resulting with higher exhaust density.

An application of direct fuel injection provides possibility to control the combustion process via variable injection timing. Fuel injection during NVO phase results with fuel reformation, which extends the air-fuel mixture flammability region, thus resulting in widening of HCCI operation range. The reformation process modifies the fuel composition at elevated temperatures and in the presence of water vapour as a catalyst [4, 12]. This process produces an exhaust-fuel mixture rich in hydrogen and monoxide. Amount of fuel which is reformed depends on injection timing, in-cylinder temperature and oxidizer content [4]. Thus, fuel injection during the NVO phase has a major impact on combustion characteristics, although fuel is introduced into the cylinder almost one crankshaft revolution before self-ignition [13].

2. Experimental setup

The examinations were carried out using a SB 3.5 single cylinder research engine with a fully variable valvetrain mechanism. All valvetrain parameters could be changed independently for the intake and exhaust valves during engine operation. Variable valve lift was achieved with the use of a hydraulic device, described in details in refs. [3, 5]. The research engine had a bowl shaped combustion chamber located in the engine head. The piston face was protruding on its perimeter and approached the cylinder head closely at TDC, which generated some amount of squish. The main engine parameters were as follows: cylinder bore was 84 mm, piston stroke – 90 mm and compression ratio – 11.7. The engine head was fitted with an in-cylinder pressure sensor.

Solenoid swirl-type injector was used in the study. The injector was positioned tangentially to the swirl generated by the shape of the intake port and inclined by 38° in relation to the cylinder axis. Fuel was provided by a high pressure pump driven by an electrical motor. Fuel pressure was kept at the desired level by an overpressure valve. Due to the design of the engine head with the hydraulic valvetrain system, side mounting of the injector in the combustion chamber was applied.

All engine ancillaries were driven by external devices. Lubricating oil was supplied by an electric oil pump and a set of pressure valves in order to provide oil for slide bearings at the pressure of 4.5 bar and lower pressure to fill the hydraulic valve lift system. The cooling system was equipped with an electrical heater and an external circulation pump, providing accurate control of water and oil temperature.

The research engine was coupled to a DC current dynamometer, which allowed for motored engine operation.

The engine control system was based on a microprocessor timing module governed by a personal computer with real-time software. A dedicated injection and ignition timing module was designed in order to allow accurate and repeatable dosing of fuel and spark discharge generation. So as to avoid control errors resulting from engine rotational speed fluctuations, crankshaft angle domain procedures (injection and ignition timing) were controlled on the base of signals from the crankshaft encoder at the angular resolution of 0.1 °CA. The same timing signal was used for triggering the A/D converter recording of the in-cylinder pressure in the crankshaft angle domain.

The composition of exhaust gases was measured with a FTIR multi-component analytical system.

3. Experimental conditions

The Research was conducted at one crankshaft rotational speed of 1500 rev/min. The temperature of cooling liquid at the engine outlet was constant and set to 87 °C +/- 1. The engine was naturally aspirated and the intake air was heated up by a water jacket around the intake pipe.

Average intake temperature was kept at the level of about 40 °C. The engine was operated in NVO mode and reduced lifts of the valves which resulted in internal EGR. Exhaust valve closing (EVC) angle was varied between 527 to 646°CA, while intake valve opening (IVO) was varied between 77 and 96°CA.

Fuel was injected in single dose with variable timing. Fuel rail pressure was constant and set to 96 bar. The range of applied air excess ratio and injection timings was limited by the misfire boundary, where it was necessary to support ignition with spark discharge.

4. Experimental results

Figure 4.1 presents in-cylinder pressure and heat release rate curves at constant IVO and variable EVC angles. Air excess ratio (λ) was kept constant at level of 1.08. In order to obtain homogeneous charge start of injection (SOI) angle was set to 20°CA after top dead centre (TDC) during NVO phase. At decreasing NVO angle, volumetric efficiency was varied from 0.29 to 0.38, while EGR rate was changed in the range of 0.5-0.38. At the same time IMEP was varied between 0.25 and 0.37 MPa.

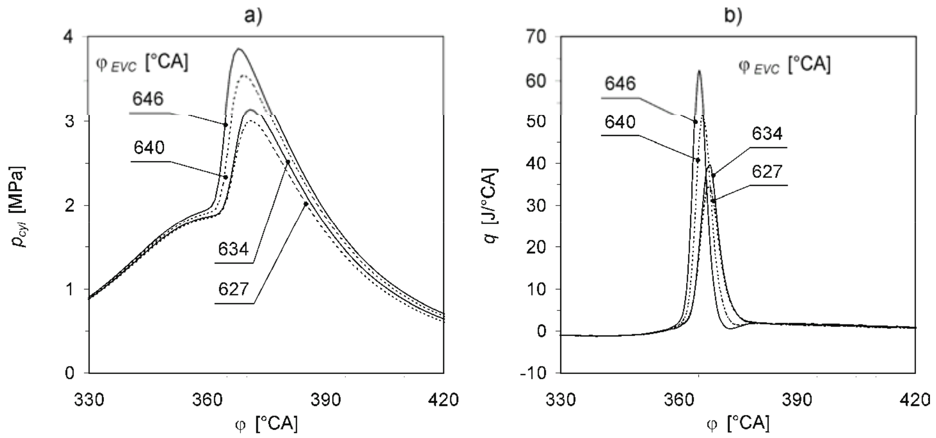

Fig. 1. In-cylinder pressure (a) and heat release rate (b) at variable φ_{EVC}, φ_{IVO} = 89°CA, n = 1500 rev/min, λ = 1.08

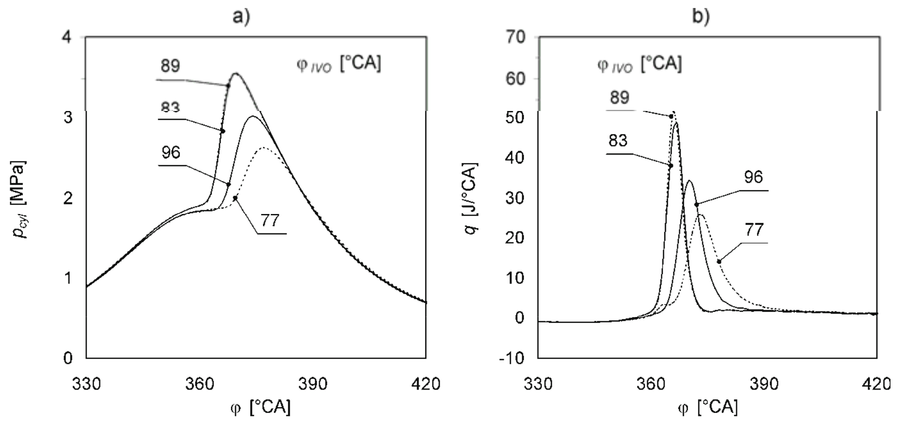

Fig. 2. In-cylinder pressure (a) and heat release rate (b) at variable φ_{IVO}, φ_{EVC} = 640°CA, n = 1500 rev/min, λ = 1.08

The decrease of volumetric efficiency via EVC advance caused also drop of compression pressure (Fig. 1 a). At two extreme EVC settings pressure at the start of combustion (SOC) was varied from 1.79 to 1.9 MPa. However, in-cylinder temperature at SOC was at the same level about 785 K. Also, crankshaft angle at SOC was independent on engine load.

In case of constant exhaust valve timing and variable IVO, its strong influence on combustion course can be observed (Fig. 2). It should be noticed that volumetric efficiency and resulted mass of injected fuel was almost constant at all conditions. At IVO angle 89 and 83°CA combustion courses where the same, and the earliest SOC and the highest maximum heat release rates appeared. Minimum ISFC was observed at this range of intake valve timing, independently of engine load. Both, retard and advance of IVO caused delay of combustion on-set and decrease of heat release rate.

Examinations at variable air-fuel ratio were carried in a whole range of exhaust valve timings from 527 to 646°CA. In each case IVO angle was set to provide the lowest ISFC at $\lambda = 1.08$. Fig. 3 presents the combustion on-set timing (calculated as crankshaft angle at 5% MFB) and the combustion duration at variable air excess ratio and NVO angle.

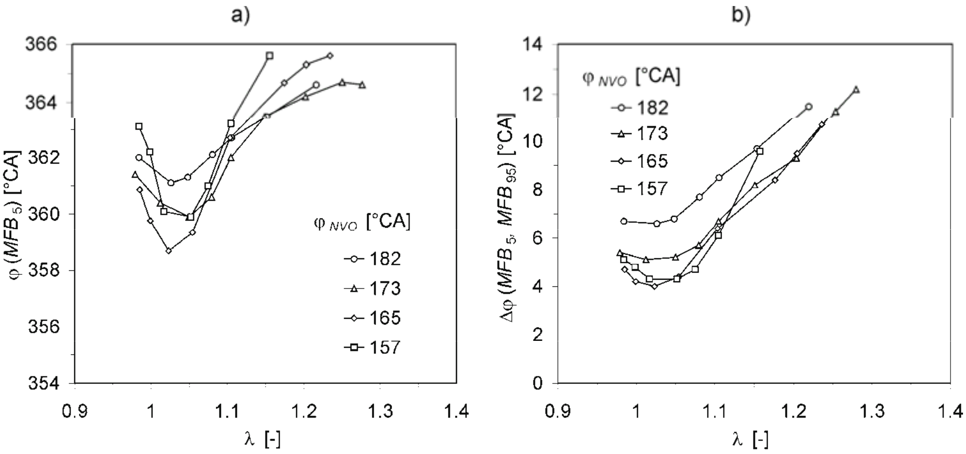

Fig. 3. Auto-ignition timing (CA at 5% MFB) (a) and combustion angle (calculated from 5% to 95% MFB) (b) with respect to air excess ratio (λ) at variable NVO angle, n = 1500 rev/min

The earliest auto-ignition took place at air excess ratio (λ) about 1.05. In this range of mixture composition auto-ignition was delayed with the increase of re-circulated exhaust amount. Both, for leaner and richer mixtures auto-ignition angle increased. It should be noticed that the highest delay of ignition timing was observed for valvetrain settings providing the lowest NVO angle. Relatively low amount of exhaust at lowered temperature due to lean mixture was not able to heat-up the in-cylinder load to the temperature of auto-ignition. This is the reason why at the lowest NVO angle the misfire limit appears for richer mixtures than for lower loads. Similarly, the lowest combustion duration (Fig. 3 b) appeared in the same range of air-fuel ratio ($\lambda = 1.05$), and rapidly increased at leaner mixtures.

Minimum indicated specific fuel consumption (ISFC) was observed at the air excess ratio range from 1.05 to 1.1 for all investigated valvetrain settings (Fig. 4). At stoichiometric mixture ISFC was about 5% higher. At leaner mixtures ISFC was rising rapidly. However, it should be noticed that the increase of λ value causes cooling of exhaust gases which results with the increase of EGR rate and the decrease of volumetric efficiency. In other words, the amount of fuel injected was decreased not only due to the air excess ratio, but also due to the dropping mass of fresh air. Thus, at the largest NVO angle IMEP was varied from 0.25 to 0.18 MPa, while for the lowest NVO angle this value was between 0.35 and 0.29 MPa. It is also noticeable, that at the NVO angle equal 157°CA ISFC was higher than at 165°CA. This is associated with the energy balance in the cylinder, particularly the increase of exhaust enthalpy.

As overall mixture dilution at variable air excess ratio rises much faster than λ coefficient itself, it results substantial drop of temperature levels which are achieved in the cylinder. It allows for dramatic reduction of cylinder-out NO_X emission. At lower loads and lean mixtures NO_X molar fraction in exhaust gases was below 10 ppm (Fig. 5). However, at higher engine loads NOX emission increased. Delay of auto-ignition timing and increase of combustion duration would reduce emission of this component, while at advanced ignition timing at lower loads thermal efficiency would be improved.

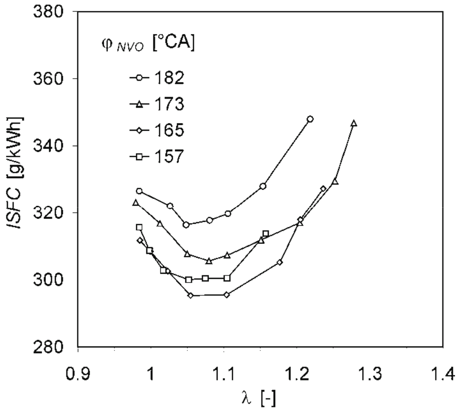

Fig. 4. Indicated specific fuel consumption with respect to air excess ratio (λ) at variable NVO angle, n = 1500 rev/min

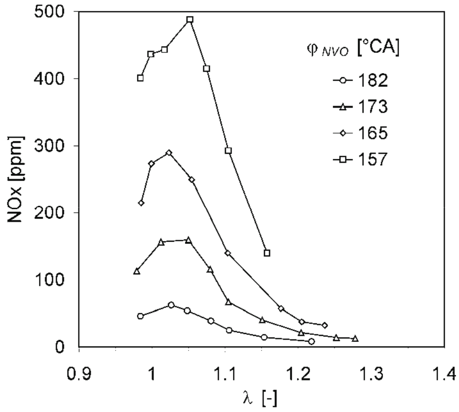

Fig. 5. NOX molar fraction with respect to air excess ratio (λ) at variable NVO angle, n = 1500 rev/min

The use of direct fuel injection provides additional benefits in scope of combustion timing control. Variable injection timing can be used for control of amount of fuel which is reformed in the cylinder during NVO phase. The earlier injection is applied the bigger fraction of fuel is reformed in the cylinder. Moreover, retarded injection provides lack of reformation, while it results with partial mixture stratification.

Examinations at variable injection timing were performed at constant air excess ratio (λ) equal 1.08 and two valvetrain settings. At the larger NVO angle the aim was to reduce ISFC, while at lower NVO angle reduction of NO_X emission was expected.

Figure 6 presents the combustion timing at variable injection timing. For the injection timing sweep between 60 and 40°CA before TDC during exhaust compression, auto-ignition was advanced. In the range of SOI from 30°CA before TDC to 10°CA after TDC ignition delay was observed. It was ascribed to piston impingement by the fuel stream. Further retard of injection resulted in rapid increase of auto-ignition angle, however at SOI later than 40°CA there was no its influence on combustion on-set. The combustion duration characteristic (Fig. 6 b) was similar to one of auto-ignition angle, however less affected by the injection timing. It should be noticed that at retarded injection, combustion duration was rising, while auto-ignition timing was kept almost constant.

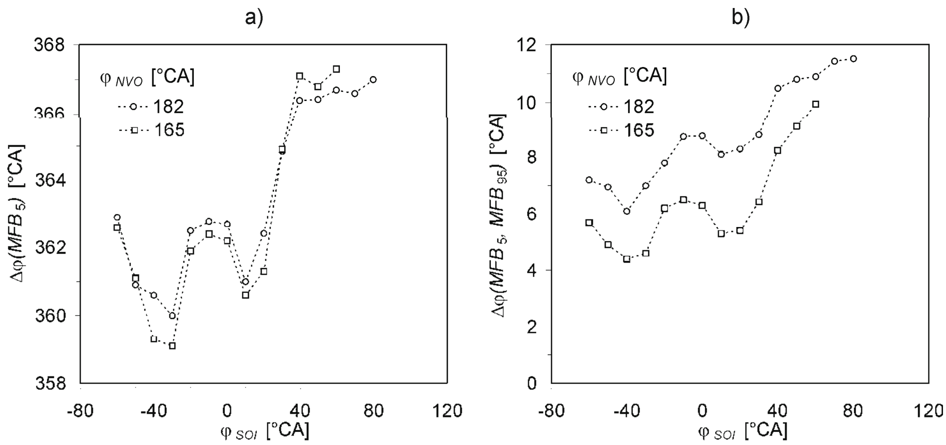

Fig. 6. Auto-ignition timing (CA at 5% MFB) (a) and combustion angle (calculated from 5% to 95% MFB) (b) with respect to start of injection timing (φ_{SOI}) at variable NVO angle, n = 1500 rev/min, λ = 1.08

Advance of auto-ignition at injection before TDC resulted in increase of ISFC for both loads of the engine (Fig. 7). Fuel injection 20°CA after TDC resulted with the same level of ISFC as for early injection. However, further delay of injection resulted in increase of fuel consumption in case of higher internal EGR rates, while for higher loads it allowed for reduction of fuel consumption.

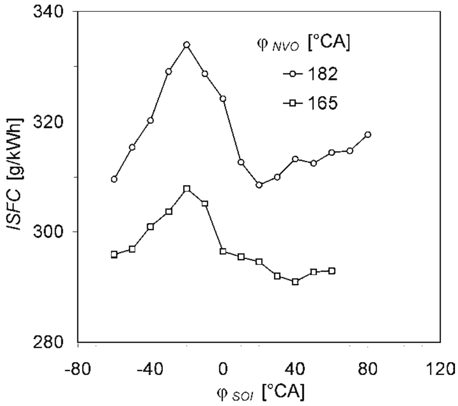

Fig. 7. Indicated specific fuel consumption with respect to start of injection timing (φ_{SOI}) at variable NVO angle, n = 1500 rev/min, λ = 1.08

Variable injection timing allows for reduction of NO_X emission at higher loads of the engine. Retard of SOI from 20°CA after TDC (which was the baseline level) to 40°CA resulted in decrease of NO_X molar fraction from 190 to 120 ppm (Fig. 8 a), while specific fuel consumption was reduced from 295 to 290 g/kWh. In case of lower load, the delay of injection provided smaller benefit in NO_X reduction, additionally decreasing thermal efficiency.

Carbon monoxide and unburned hydrocarbons emissions are not affected by engine load (Fig. 8 b and c). HC molar fraction increases with injection retard, while CO fraction slightly decreases. Dramatic increase of emissions of both components for injection angles between –30°CA and TDC was a result of mentioned before piston impingement by the fuel stream, and is not applicable from the point of view of the engine control strategy.

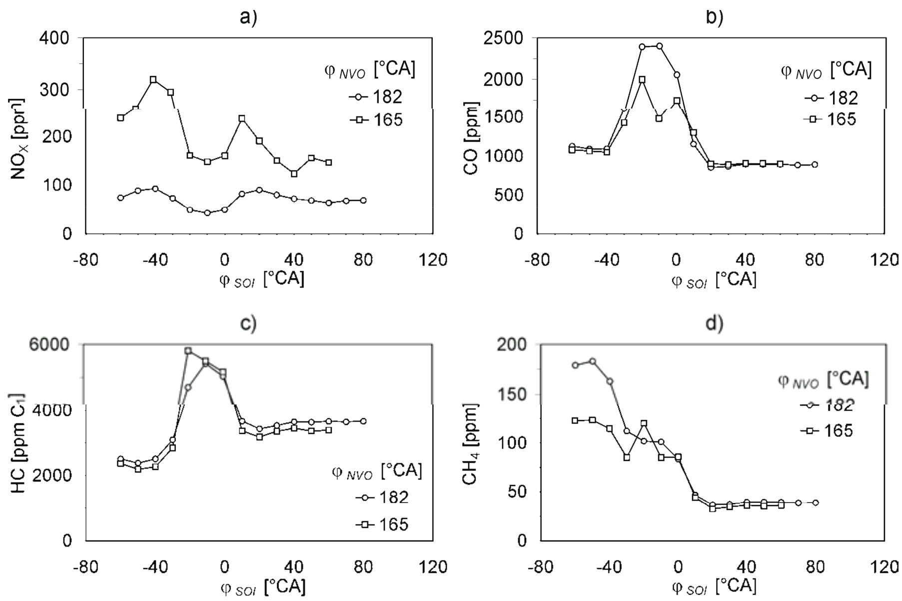

Fig. 8. Molar fractions of exhaust toxic components with respect to start of injection timing (φ_{SOI}) at variable NVO angle, n = 1500 rev/min, λ = 1.08

At early injections, where in-cylinder fuel reformation was expected, elevated levels of methane content was observed (Fig. 8 d). However, at the same time THC emission was lower. Considering the fact that CH_4 is one of the fuel reformation products, it proves that this process takes place during negative valves overlap phase. At smaller amount of re-circulated exhaust methane content was smaller too. In practice, reformation reactions use free oxygen present in exhaust rather than reduce carbon dioxide [4]. It suggests that reformation intensity would be affected by air excess ratio as well as amount of re-circulated exhaust.

5. Conclusions

Gasoline homogeneous charge compression ignition (HCCI) engine with a fully variable valvetrain mechanism and direct fuel injection was used for the evaluation of parameters which affect auto-ignition timing and combustion duration and their relation to fuel consumption and exhaust emissions. Auto-ignition was achieved with the use of internal EGR provided by negative valve overlap (NVO). The investigation was carried out for variable valve timings and variable air excess ratio. Additionally, variable injection timing was applied in order to take advantage of in-cylinder fuel reformation. The findings of this study are summarised below.

At the adjustment of the engine load via variable NVO angle it was possible to control the engine load in the range of indicated mean effective pressure (IMEP) from 0.37 to 0.25 MPa at the air excess ratio (λ) equal 1.08, which provide the lowest fuel consumption. The range of the engine load was extended by variable air excess ratio. At leaner mixtures IMEP was reduced to 0.18 MPa.

The combustion on-set is affected mainly by air excess ratio. The influence of exhaust gas re-circulation (EGR) rate and resulting volumetric efficiency is much smaller. However the combustion duration is determined by all of the parameters.

The advance of injection timing results with fuel reformation, which was identified via substantial increase of methane content in the exhaust gases. The fuel reformation accelerates auto-ignition of the mixture. However, in the examined range of air excess ratio reformation leads to increase of fuel consumption. It is expected that this chemical process would extend the range of the applicable lambda values. Delay of injection timing at higher loads benefited with increase of the engine thermal efficiency and reduction of NO_X emission.

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