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ScienceDirect

Energy Procedia 81 (2015) 784 – 793

Energy Procedia

69th Conference of the Italian Thermal Machines Engineering Association, ATI2014

Preliminary Experimental Study on Double Fuel HCCI combustion

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Abstract

This paper regards an experimental study on a particular internal combustion engine process which combines Double Fuel combustion with Homogeneous Charge Compression Ignition (HCCI) using mixtures of natural gas (NG) and gasoline. The tests performed on a CFR engine demonstrate that HCCI combustion can be achieved using NG-gasoline mixtures without knocking occurrence for low to medium engine load varying the proportion between the two fuels. The main advantage of this new combustion process relies on the noticeable higher engine efficiency obtained with respect to conventional spark ignition operation, and on the strong reduction of NO_X emissions.

Peer-review under responsibility of the Scientific Committee of ATI 2014

Keywords: Double Fuel; HCCI; simultaneous combustion; natural gas; gasoline; Nitric Oxide; efficiency;

1. Introduction

Energy efficiency as well as pollutant emissions reduction are the main research and development purposes on modern internal combustion engines design. As a consequence, researchers and manufactures are continuously impelled to develop innovative and affordable solutions. Nowadays engine downsizing and super charging as well as cutting-edge valve train control systems are widely spread in the automotive market, contributing to improve engine performances and efficiency. For similar reasons various hybrid engine systems recently obtained a good commercial success. Moreover, in the last years, the increasing cost of conventional fuels promoted the use of alternative fuels as well as fuels mixtures. Gaseous fuels, such as natural gas (NG) and Liquefied Petroleum Gas (LPG), represent today a concrete alternative to conventional fuels for road vehicles propulsion and stationary

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Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.084

engines, since these are characterized by a relatively low cost and lower environmental impact. For these reasons in the last 20 years, gaseous fuels [1] [2] as well as liquid fuel mixtures [3] have been deeply studied with the aim to experience their compatibility and properties as alternative fuels for spark ignition engines.

The use of fuels mixtures has been also studied by the authors in previous works [4] [5], where the simultaneous combustion of a homogeneous mixture of gasoline and gaseous fuel (NG or LPG) in stoichiometric proportion with air has been experimentally tested, with several proportions between gasoline and gas, thus exploiting the good qualities of both fuels to obtain cleaner and more efficient combustions. In effect, the addition of natural gas (as well as LPG) to gasoline, due to the high knocking resistance increase obtained, allowed to run the engine with "overall stoichiometric" mixtures even at full load and to improve the thermodynamic cycle by advancing the combustion phase. As a result, with respect to the pure gasoline mode, efficiency increments of about 26% were obtained, together with HC and CO reduction in the order of 90% and without noticeable power losses (-4%). This third combustion mode, called by the authors Double Fuel combustion, is quite different from the well-known Dual Fuel, in which the auto-ignition of a small quantity of one of the two fuels acts as igniter to start the flame propagation combustion of the second fuel. As shown in [6][7], due to the considerable difference of knock resistance between gasoline and NG, the mixture auto-ignition property revealed a non-linear function of the natural gas concentration. Afterwards the simultaneous combustion of gas and gasoline has been investigated also by other researchers [8][9][10][11], both in naturally aspirated and supercharged SI engines.

In light of their results, the authors decided to exploit NG-gasoline mixtures to experience a combustion process that combine the Double Fuel and the Homogeneous Charge Compression Ignition (HCCI) combustion, a particular combustion process [12] in which a homogeneous mixture of air and fuel is auto-ignited through compression. Similar to a SI engine, the fuel and air are mixed to obtain a homogenous mixture, which eliminates fuel-rich diffusion combustion thus dramatically reducing the particulate matter emissions, which are usually associated with conventional Compression Ignition (CI) combustion. Moreover, the auto-ignition of a homogenously premixed charge eliminates the high-temperature flames of conventional engine combustion, reducing Nitrogen Oxides (NO_X) emissions respect to conventional CI combustion. Furthermore, similarly to CI engines, HCCI combustion power output is ordinary controlled by unthrottled operation, acting on the air-to-fuel ratio, thus reaching higher thermal efficiency compared to SI engines in part load condition. However, respect to conventional engines that have a direct control on the combustion phase, realized by means of the spark timing in SI engines or by means of fuel injection timing in CI engines, HCCI engines achieve auto-ignition of the homogeneous mixture around the Top Dead Centre (TDC), thus presenting some difficulties in the control of both combustion phase and rate of heat release, in a wide range of operative conditions. In particular, at idle, in-cylinder thermodynamic conditions are not sufficient to auto-ignition while at high loads the rate of heat release results excessive and entails heavy knocking conditions.

As concerns the Double Fuel HCCI combustion, no references or previous works have been found in the scientific literature, hence the experimental study carried out by the authors represents a quite novel research, whose original results are illustrated in the present paper.

2. Experimental setup

The experimental campaign has been carried out using a Cooperative Fuel Research (CFR) engine (Table 1), which is commonly employed for fuel octane rating and, thanks to its robust construction, allows to experience various combustion process such as HCCI, even in knocking condition. The CFR engine is a four-stroke two valve stationary single-cylinder spark-ignition engine, with a particular system that allows to vary quickly and accurately the Compression Ratio (CR) from 4.5 to 16. The CFR engine is connected to an electric synchronous motor that maintains the CFR speed at 900 RPM both in operating and motored condition. The engine is equipped with two electric heater which are connected to two independent PID control systems Omega CN4116 in order to maintain both inlet air temperature (T_{AIR}) and intake air/fuels mixture temperature (T_{IN}) at the required values during the tests.

As regards fuel supplying, a standard CFR engine features an original carburettor system which does not allow the use of gaseous fuels. The authors hence endowed the CFR engine with two independent injection systems (as shown in Figure 1) and a Venturi air flow meter in order to realise each desired NG-gasoline mixture and to accurately control the overall air-fuels ratio (a more detailed description is given in [6] [7]).

Table 1. CFR Engine Specifications.

Manufacturer	Dresser Waukesha
Model	F1/F2 Octane
Compression ratio	4.5 - 16
Bore	82.6 mm
Stroke	114.3 mm
Connecting rod length	254.0 mm
Displacement	611.2 cm³

Gaseous fuel mass flow was measured by means of a Bronkhorst mini CORI-FLOW® Coriolis effect mass flow meter while gasoline mass flow was deduced on the basis of the imposed injection time by means of a precise injector flow chart previously experimentally determined.

A personal computer (n.1 in Figure 1) equipped with a National Instruments DAQCard 6062E was used to manage the two injection systems and perform data acquisition, by means of an expressly designed software developed by the authors in LabVIEW environment.

The in-cylinder pressure, measured by means of a Kistler AG piezoelectric pressure sensor placed on the combustion chamber, and all the relevant quantities (manifold absolute pressure as well as air and fuels mass flows) were acquired by means of the above mentioned DAQCard 6062E using as trigger and scan clock the pulses generated by a 360 pulses per revolution incremental optical encoder connected to the engine crankshaft.

The knock occurrence was monitored using the in-cylinder pressure sensor. To this purpose another personal computer (n.2 in Figure 1) was used to process the in-cylinder pressure signal, acquired by means of a second National Instruments DAQCard 6062E, with a scan rate of 200 kHz.

Finally, in order to measure pollutant emissions, the test bench has been also endowed with an exhaust gas analyzer.

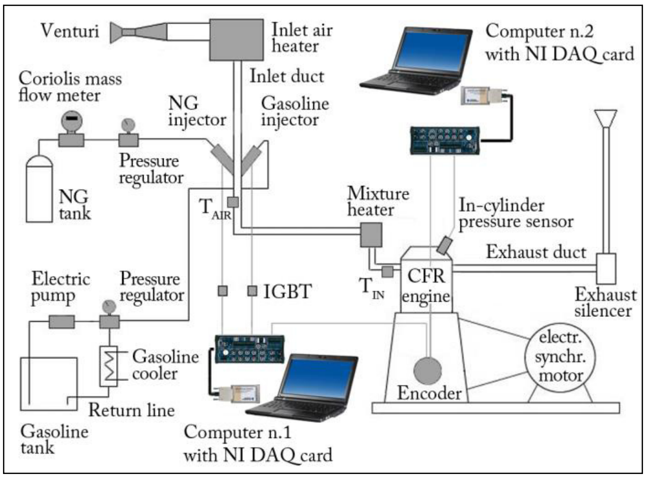

Figure 1 - Experimental system layout.

3. Test methods

In the experimental campaign carried out, NG-gasoline mixtures, under different proportions between the two fuels, were tested in order to define the range of Double Fuel HCCI operating conditions without knocking and to evaluate the corresponding performances in terms of indicated efficiency and pollutant emissions.

Table 2. Double Fuel HCCI test conditions.

Engine speed	900 RPM
Intake temperature	135, 165 °C
Engine coolant temperature	100 °C
Compression ratio	15, 16
NG mass fraction	from 20 to 80 %
Engine load	regulated varying the air to fuel ratio

As resumed in Table 2, the HCCI tests were carried out running the CFR engine at 900 RPM, with two compression ratios (15 and 16) and two intake air-fuels mixture temperatures (135 and 165 °C). By means of preliminary evaluations, it was observed that these parameters allow to experience a wide range of different Double Fuel HCCI operative conditions with NG-gasoline mixtures.

The NG mass fraction, i.e. the ratio between the injected NG mass and the mass of the total amount of fuel injected, was varied from 20 to 80% (or to the maximum value allowing stable HCCI operation without knocking). For each mass fraction, the engine load was regulated to run the engine at the maximum knock free load, acquiring 50 consecutive in-cylinder pressure cycles together with all the other quantities. It was hence possible to calculate the overall air-to-fuel ratio, the Indicated Mean Effective Pressure (IMEP), which in a CFR engine represents the engine load, and engine indicated efficiency.

The overall air-to-fuel ratio (α) was measured in terms of *lambda*, defined as:

$$\lambda = \frac{\alpha_{Measured}}{\alpha_{Stoichiometric}} = \frac{G_{Air}}{14.7 \cdot G_{Gasoline} + 16.9 \cdot G_{NG}} \qquad (1)$$

(G_{Air}, G_{NG} and $G_{Gasoline}$ are respectively the air, NG and gasoline mass flows, while 14.7 and 16.9 represent the stoichiometric air-to-fuel ratio of gasoline and natural gas).

Table 3. Test conditions during spark ignition tests.

Engine speed	900 RPM
Intake temperature	52°C
Engine coolant temperature	100°C
Compression ratio (CR)	6 with gasoline, 7 with NG
Air-to fuel ratio (α)	Stoichiometric for both NG and gasoline
Engine load	controlled with MAP=0.6 to 1 bar
Spark advance	controlled for LPP=15 CAD ATDC without knocking

The experimental results of the HCCI tests were compared, in terms of IMEP, indicated efficiency and pollutant emissions, with those registered running the CFR engine in ordinary spark ignition mode, both with 100% gasoline or 100% NG. During SI tests (Table 3), the engine load was regulated varying the Manifold Absolute Pressure (MAP) from 1 to 0.6 bar, which revealed the minimum allowable value for a stable SI combustion in the CFR engine. For both fuels the engine was run only with stoichiometric air-to-fuel ratio, with a compression ratio of 6 for gasoline and 7 for NG. By means of preliminary evaluations, it was observed that these parameters allow optimal combustion phase without knock occurrence in each SI test: this was performed setting spark timing so as to maintain the Location of Peak of Pressure (LPP) at 15 Crank Angle Degrees (CAD) After Top Dead Centre (ATDC), as recommended in [14]. Dealing with IMEP evaluation, a crucial factor is represented by the determination of engine Top Dead Centre position, which should be carried out within the accuracy of 0.1 CAD (see [14] and [15]): a capacitive sensor Kistler 2629B has been employed by the authors for a correct evaluation of TDC position for each compression ratio adopted in the test.

The entire experimental campaign was carried out using a single sample of commercial gasoline (with 85 MON) and a single sample of NG, whose composition is reported on Table 4.

Table 4. Composition and properties of the natural gas used.

Components	% vol.
Methane - CH_4	85.79
Ethane - C_2H_6	7.86
Propane - C_3H_8	1.61
N-butane - C_4H_{10}	0.19
Isobutane - C_4H_{10}	0.28
Butylene - C_4H_8	0.05
Isopentane - C_5H_{12}	0.06
N-pentane - C_5H_{12}	0.06
Carbon dioxide - CO_2	1.04
Nitrogen - N_2	2.96
Helium - He	0.09
Properties	
Motor Octane Number [6],[13]	122.1
Lower heating value [MJ/kg]	46.0
Stoichiometric air/fuel mass ratio	16.9

4. Experimental results

This preliminary experimental study demonstrates that NG-gasoline mixtures can be used to realize, in certain operative conditions, the HCCI combustion without knock occurrence. Furthermore, as introductory observation, it is worth to mention that none of the test conditions explored allowed a stable and knock free HCCI combustion using only natural gas or gasoline. The experimental results here presented define the range of stable and knock free HCCI as function of three engine operative conditions: compression ratio (CR), intake temperature (T_{IN}) and mixture composition (i.e. NG mass fraction).

As compression ratio and T_{IN} increase, the HCCI combustion range increases in terms of NG mass fraction. As shown in Figure 2, the maximum range of engine load was registered with CR 16 and T_{IN} 165 °C. In these operative condition it was possible to run the engine in HCCI with NG-gasoline mixtures containing between 20% and 80% in mass of NG.

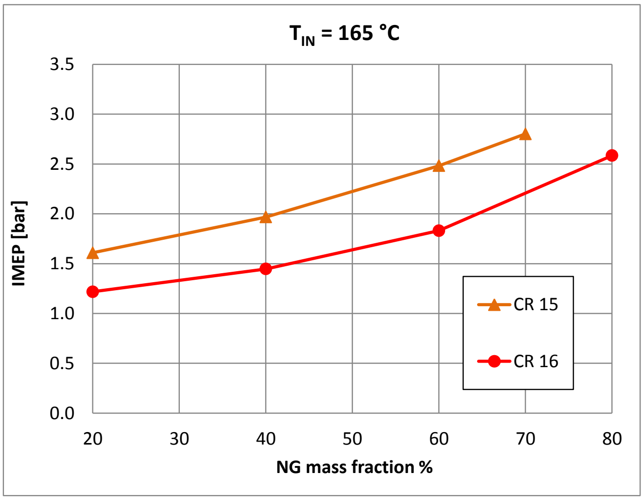

Figure 2 - Maximum HCCI engine load as function of NG mass fraction for different CR, at T_{IN} 165 °C.

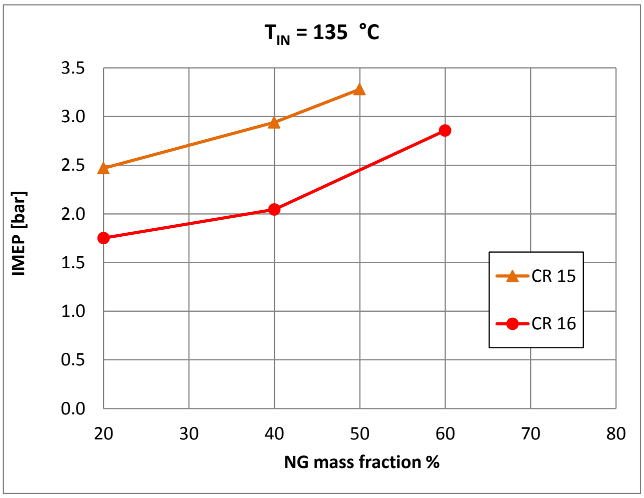

Figure 3 - Maximum HCCI engine load as function of NG mass fraction for different CR, at T_{IN} 135 °C.

Furthermore, as shown in Figure 2 and Figure 3, for fixed CR and T_{IN}, the maximum knock free engine load is proportional to the NG mass fraction. The higher IMEP values were achieved running the engine at the lower T_{IN} (135 °C) with CR=15: the lighter thermodynamic conditions, in effect, prevented knocking occurrence even with high combustion heat release. Similar considerations can be made observing Figure 2 and Figure 3: for each NG mass fraction, a CR reduction entailed an increase of the engine load.

For a better understanding of the real potential of the newly tested combustion process, a comparison is here carried out between the experimental results obtained with HCCI and with ordinary SI mode (both with NG or gasoline). To this purpose the measured indicated engine efficiency are reported in Figure 4 for both combustion modes as function of the measured engine IMEP.

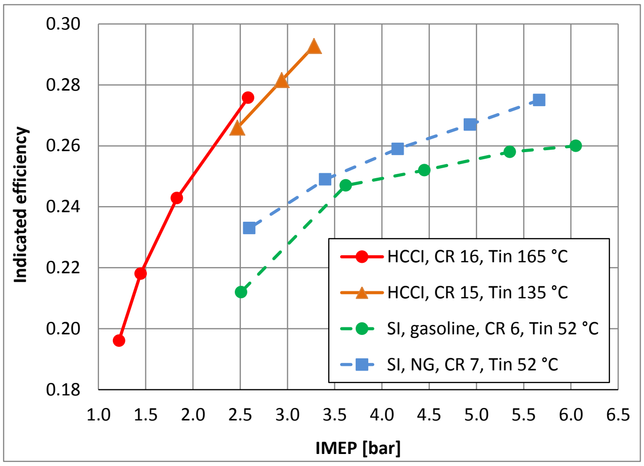

Figure 4 - Indicated efficiency as function of IMEP.

As concern the HCCI mode, the two curves in Figure 4 delimit the range of variation of the measured IMEP: CR 16-TIN 165°C, which entailed the lower IMEP values (Figure 2), and CR 15-T_{IN} 135°C, which instead allowed to reach the higher IMEP values (Figure 3). The two dashed curves instead refer to the IMEP measured in spark ignition mode with only pure fuels (gasoline or NG). Pure natural gas SI combustion shows higher efficiency respect to the pure gasoline mode thanks to the higher compression ratio (7 rather than 6, due to the very high knock resistance of NG, as shown in Table 4) and to the better mixing capability with air.

As can be observed, the IMEP range with HCCI combustion (Figure 4) varies from 20% to 55% of the IMEP measured in SI mode, whose maximum value resulted 6 bar with gasoline. In the range common to both combustion modes, i.e. between 2.5 and 3.28 bar (Figure 4), the HCCI combustion featured significantly higher indicated efficiencies respect to SI mode: a mean increment of 16% has been measured with respect to natural gas SI, while a mean increment of 24% has been obtained with respect to gasoline SI. The reasons of these increments are multiples: first of all, the HCCI mode is not affected by the pumping losses caused by the traditional throttle regulation adopted for SI engine; the higher compression ratio used in HCCI mode, necessary for autoignition, and, not less important, the significant reduction on the overall combustion duration with respect to SI mode.

As a result, HCCI combustion represents an efficient alternative to the traditional SI combustion in partial load conditions. During the HCCI tests, the engine load was regulated, similarly to CI engine, varying the air-to-fuel ratio: lambda ranged from 3.20 to 1.75 (significantly higher than the values commonly adopted for spark ignition engine, comprised between 0.8 and 1.1).

The experimental results showed other interesting advantages of HCCI combustion for what concerns pollutant emissions. In HCCI mode, in effect, both Carbon Oxide (CO) and Carbon Dioxide (CO_2) emissions resulted much lower (about halved) with respect to the measured values in SI mode. In particular, CO values ranged from 0.09% to 0.18% (vol.), while CO_2 emissions resulted between 3.61% and 6.64%, with an increasing trend as function of engine load due the corresponding reduction of air-to-fuel ratio.

The most interesting result however concerns the emissions of Nitric Oxide (NO), which showed a drastic reduction, of two order of magnitude, respect to the levels measured in SI mode, which varied between 2000 and 3700 ppm (Figure 5) as function of engine load. In HCCI mode NO emissions instead revealed a mean value of 20 ppm, with a maximum of 85 ppm in correspondence of the maximum engine load (IMEP 3.28 bar) experienced with CR=15 and T_{IN}=135°C. As regard SI mode, in the range of IMEP between 2.5 and 3.28 bar, considering a probable catalytic converter efficiency of 90%, the lowest emission of 2000 ppm would be reduced to 200 ppm, which is still higher than the raw emission in HCCI mode. These almost negligible NO emissions measured in HCCI mode are mainly due to the lower peak temperature which characterize this combustion process.

With respect to SI operation, a disadvantage has been found in the higher hydrocarbon (HC) emissions measured in HCCI mode (300 to 500 ppm, as shown in Figure 6), probably caused by the different combustion process (no flame propagation in the combustion chamber) and to the lower combustion temperature. However, these raw emission levels could be easily reduced by means of a common oxidation catalytic converter, especially if the large available excess of air is considered.

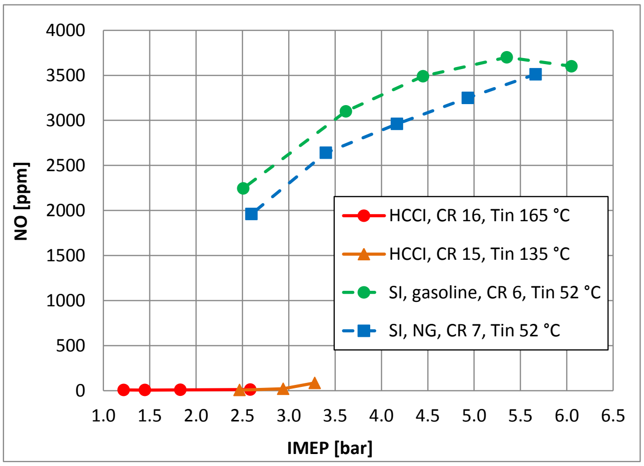

Figure 5 - NO emissions as function of IMEP.

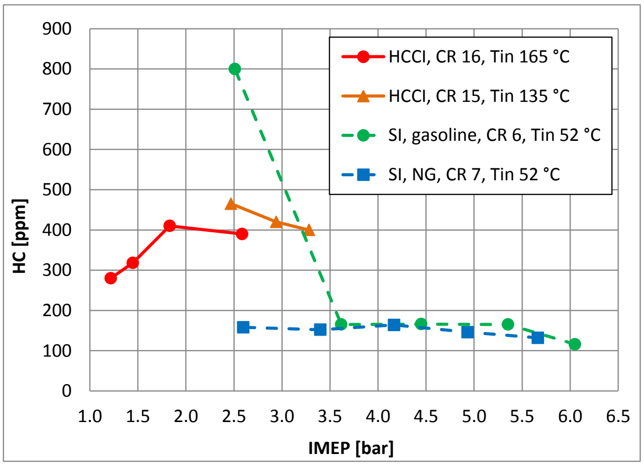

Figure 6 - HC emissions as function of IMEP.

5. Conclusion

The aim of this work was to experience a particular internal combustion engine process which combines Double Fuel combustion with Homogeneous Charge Compression Ignition. No references were found about this specific topic, hence the original results here presented, obtained by the experimental campaign carried out by the authors, covers a certain lack of literature and demonstrate that Double Fuel HCCI combustion can be achieved with natural gas-gasoline mixtures, reaching significantly increments in indicated efficiency (up to +24%) with respect to conventional spark ignition mode, in a load range between 20% and 55% of the maximum SI engine load. The greatest advantage of the Double Fuel HCCI combustion however resulted to be in the pollutant emissions, which, with the only exception of HC, revealed a strong reduction. In particular, in HCCI mode, lower NO emissions of two order of magnitude have been measured respect to SI mode, with a mean value of only 20 ppm. In light of the constricting environment saving rules and increasing cost of fuels, the negligible NO emissions, as well as the high indicated engine efficiency, represent the main and most interesting results obtained in this preliminary experimental study.

Nomenclature

ATDC	After Top Dead Centre
CAD	Crank Angle Degree
CFR	Cooperative Fuel Research
CO	Carbon Oxide
CR	Compression Ratio
HC	Hydro Carbon
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
LPG	Liquefied Petroleum Gas
MON	Motor Octane Number
NO	Nitric Oxide
NG	Natural Gas
RPM	Revolutions Per Minute
T_{IN}	Inlet temperature
SI	Spark Ignition
x_{NG}	Natural gas mass fraction
α	Air to fuel ratio
G_{Air}	Inlet air mass flow
G_{NG}	Natural gas mass flow
$G_{Gasoline}$	Gasoline mass flow

Acknowledgements

The authors would like to express their acknowledgments to Mr. Beniamino Drago for his invaluable technical support.

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