

Study of High Speed Gasoline Direct Injection Compression Ignition (GDICI) Engine Operation in the LTC Regime

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Youngchul Ra
Univ. of Wisconsin

Paul Loeper
Univ. of Wisconsin - Madison

Rolf Reitz and Michael Andrie
Univ. of Wisconsin

Roger Krieger
Univ. of Wisconsin - Madison

David Foster
Univ. of Wisconsin

Russ Durrett, Venkatesh Gopalakrishnan, Alejandro Plazas, Richard Peterson and
Patrick Szymkowicz
General Motors LLC

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ABSTRACT

An investigation of high speed direct injection (DI) compression ignition (CI) engine combustion fueled with gasoline (termed GDICI for Gasoline Direct-Injection Compression Ignition) in the low temperature combustion (LTC) regime is presented. As an aid to plan engine experiments at full load (16 bar IMEP, 2500 rev/min), exploration of operating conditions was first performed numerically employing a multi-dimensional CFD code, KIVA-ERC-Chemkin, that features improved sub-models and the Chemkin library. The oxidation chemistry of the fuel was calculated using a reduced mechanism for primary reference fuel combustion. Operation ranges of a light-duty diesel engine operating with GDICI combustion with constraints of combustion efficiency, noise level (pressure rise rate) and emissions were identified as functions of injection timings, exhaust gas recirculation rate and the fuel split ratio of

double-pulse injections. Parametric variation of the operation ranges was also investigated with respect to initial gas temperature, boost pressure and injection pressure. Following the modeling, experiments were performed under the conditions suggested by the numerical results in order to confirm the feasibility of GDICI operation at full load, as well as to validate the numerical simulations.

The results showed good agreement between the experiments and the model predictions. Due to the high volatility and low cetane index of gasoline combined with reduction of combustion temperature through utilization of EGR, both PM and NOx emissions could be reduced to levels of about 0.1 g/kg-f while maintaining experimental gross ISFC at about 180 g/kw-hr. The numerical simulations helped to explain the in-cylinder spray combustion behavior and to identify characteristics of GDICI that differ from those of diesel-fueled operation. Maps of operable conditions were generated

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that allow extension of low-emission engine concepts to full load operation in high speed GDICI engine operation with fuel efficiencies comparable to those of corresponding diesel fuel operation while meeting emissions mandates in-cylinder.

INTRODUCTION

In order to continue usage of compression ignition (CI) combustion under current pollutant emission regulations, many steps must be taken to lower the oxides of nitrogen (NO_x) and particulate matter (PM) emissions of conventional diesel engines. In most advanced combustion techniques, variation of the injection timing and ignition delay is utilized to lower combustion temperatures through enhancing mixing times and/or combustion intensity. If the mixing is accelerated via either increased swirl or shortened injection duration (i.e., increased injection velocity), or if the fuel conversion rate is slowed down, auto-ignition can be timed to occur after the fuel and air are better mixed and soot levels can thus be reduced. NO_x emissions can be reduced by reducing the combustion temperature by running under highly dilute conditions (i.e., lean operation or using high levels of exhaust gas recirculation (EGR)).

One of the attractive strategies for lowering emissions is premixed charge compression ignition (early PCCI) [1,2,3]. This strategy is typically achieved with the use of a more advanced injection timing than for conventional diesel. The fuel is injected into the combustion bowl with the objective of providing a fully vaporized mixture of fuel and air, which forms mixture conditions close to a homogeneous charge. Although both the control of combustion phasing and reduction of emissions can be achieved in the PCCI regime, the amount of fuel that can be injected per cycle is limited by the high pressure-rise rate that is a consequence of pre-mixed charge auto-ignition.

Another strategy is to use relatively retarded injection timings to increase the ignition delay and to allow for fuel-air pre-mixing (late PCCI) [4,5,6,7]. When heat release occurs after top dead center (TDC), the pressure rise rate can be reduced for a given maximum heat release rate, which enables engine operation at higher load. However, if the heat release timings are retarded too much, the combustion becomes unstable and the engine efficiency decreases.

Other types of combustion use an early small injection (pilot injection) with a late injection of additional fuel to keep temperatures low [8, 9]. This method is able to combine the merits of the strategies using early and retarded injection timings. The maximum heat release rate can be reduced by spreading out the heat release through multiple injections.

One of the crucial characteristics of fuels considered for use in compression ignition engines is their autoignitability. This depends on the chemical composition of the fuel, as well as

on the evolution of the thermal state and the composition of the charge mixture. Cetane number (CN) or Research octane number (RON) are widely used as indicators of ignitability or resistance to autoignition. They are measured under pre-defined conditions, and thus can essentially indicate the effects of fuel composition on the propensity to autoignite. Typically, practical diesel fuels have CN greater than 40 and are so prone to autoignition that they can autoignite before the fuel is sufficiently mixed with air. On the contrary, gasoline fuels have CN lower than 30 (or RON higher than 60 [12]. Since a well-mixed charge condition before ignition is desirable in low emission CI combustion, fuel characteristics such as those of gasoline that provide longer ignition delay to allow fuel to mix with air are expected to be beneficial. In order to enhance the mixing with air, both high volatility and diffusivity are additional key properties of desirable fuels.

Hashizume, et al. [8] used a fuel with a CN of 19 to retard the first ignition timings in their multi-stage diesel combustion experiments. In these experiments, CN effects were successfully demonstrated in terms of ignition delay and fuel consumption by comparing with the results for a fuel with a CN of 62. Shimazaki, et al. [10] have reported that a lower cetane fuel increases the mixing time before ignition and enables lower smoke levels to be achieved compared to a higher cetane fuel. Diesel fuels of different CN and volatility characteristics were also tested by Risberg, et al. [11] in an engine. They showed that the CN describes the autoignition quality of diesel-like fuels in homogeneous charge compression ignition (HCCI) combustion well.

The effect of fuel auto-ignition quality on engine ignition timings and emissions has been studied experimentally by Kalghatgi, et al. [12,13,14] for four different fuels with different CN and volatility, including conventional gasoline. With injection timings near TDC, the gasoline had a much larger ignition delay so that the engine could be run at significantly higher loads with low smoke on gasoline compared to diesel fuel with no detriment to NO_x, CO, unburned hydrocarbon (UHC) and fuel consumption. It has been also reported that with multiple injection with gasoline fuel, the maximum heat release rate could be significantly reduced for a given IMEP while maintaining low cyclic variation compared to a single injection [14].

Weall and Collings [15] examined the operating characteristics of a light duty CI engine with regular gasoline fuel at low engine speed and load and showed that stable, low soot and NO_x emissions operation was possible with raised intake manifold pressure, temperature, and fuel stratification.

More recently, Hildingsson, et al. [16] investigated the effects of octane number of gasoline fuels on emissions in a light-duty CI engine operated at light and medium load conditions with single injection strategy and they showed that both NO_x

and soot emissions could be reduced substantially compared to the corresponding diesel operations and there is an optimal octane number fuel to minimize the emissions.

While significant experimental effort has been devoted to the study of late direct injection (DI) of gasoline fuels in conventional diesel engines, corresponding multi-dimensional modeling studies have been rarely reported. Recently Ra, et al. [17] conducted a numerical study on combustion of DI gasoline sprays in a conventional heavy-duty diesel engine. The numerical models were validated by comparing predicted results of single and double injection combustion of gasoline and diesel fuels with available measurements and the results confirmed that particulate emissions of DI gasoline engine combustion were significantly reduced compared to corresponding diesel fuel cases, mainly due to the enhanced mixing of fuel and air before ignition due to the longer ignition delay of gasoline for the same combustion phasing.

Ra and Reitz [18] further investigated the effects of parametric variation of engine operating conditions on performance of GDICI combustion in the same heavy-duty diesel engine. They found that injection timing influences both UHC and CO emissions and noise level at the same time, and split ratio variation is an effective way to reduce CO emissions and maximum pressure rise rates at only a slight expense of increased ISFC. EGR ratio variation was also found to be effective to control the maximum pressure rise rate, although its window is so narrow that mixtures are likely to misfire at levels of EGR that are reliably used in conventional diesel-fueled engine operation.

So far, with the majority of research effort focused on utilization of GDICI combustion in heavy-duty diesel engine, little attention has been paid to GDICI operation in light-duty engines. In this study, an investigation of GDICI in the low temperature combustion (LTC) regime in a light-duty engine is presented. As an aid to plan engine experiments at full load (targeting 16 bar IMEP, 2500 rev/min), exploration of operating conditions was first performed numerically employing a multi-dimensional CFD code, KIVA-ERC-Chemkin, that features improved sub-models and the Chemkin library. The oxidation chemistry of the fuel was calculated using a reduced mechanism for primary reference fuel combustion. Operation ranges of a light-duty diesel engine operating with GDICI combustion with constraints of combustion efficiency, noise level (pressure rise rate) and emissions were identified as function of injection timings, exhaust gas recirculation ratio and the fuel split ratio of double-pulse injections. Parametric variation of the operation ranges was also investigated with respect to initial gas temperature, boost pressure and injection pressure. Following the modeling, experiments were performed under the conditions suggested by the numerical results in order to

confirm the feasibility of GDICI operation at full load, as well as to validate the numerical simulations.

NUMERICAL APPROACH

PHYSICAL MODELS

For simulating the spray process and the mixing and combustion of fuel/air mixtures in the combustion chamber, various physical submodels were employed in the present KIVA-ERC-CHEMKIN code. The code is based on KIVA3V Release 2 [19] coupled with the CHEMKIN II library [20]. The added sub-models include models related to drop breakup, collision and coalescence, drop deformation, drop evaporation, wall impingement and vaporization, etc. [17, 18].

A hybrid primary spray break-up model that is computationally efficient as well as comprehensive enough to account for the effects of aerodynamics, liquid properties and nozzle flows was employed [21]. In this model, the injected fuel “blobs” are tracked by a Lagrangian method while the break-up of each blob is calculated from considerations of jet stability from Kelvin-Helmholtz (KH) instability theory. For the secondary and further break-up processes, a Kelvin Helmholtz (KH) - Rayleigh Taylor (RT) hybrid model was used.

A droplet collision model based on the stochastic particle method [19] was used, in which the collision frequency is used to calculate the probability that a drop in one parcel will undergo a collision with a drop in another parcel, assuming all drops in each parcel behave in the same manner. The probability of coalescence is determined considering the Weber number that includes the effects of density and surface tension of the liquid droplets.

Droplet deformation in terms of its distortion from sphericity is modeled using a forced, damped harmonic oscillator model, where the surface tension and viscosity of the droplet are the major properties used in the restoring force and damping terms, respectively [22]. Distortions of the droplets affect the momentum change between droplet and the ambient gas, and subsequently the drop velocities (or relative velocity between the drop and the gas) that are the governing parameters in the breakup and evaporation processes.

The droplet vaporization model [23] considers the evaporation of spray droplets using the Discrete Multi-Component (DMC) approach under temperatures ranging from flash-boiling conditions to normal evaporation. The improved model accounts for variable internal droplet temperatures, and considers an unsteady internal heat flux with internal circulation, and a model for the determination of the droplet surface temperature. The model uses an effective heat transfer coefficient model for the heat flux from the

surrounding gas to the droplet surface. Also, the variable density of the diesel-surrogate fuel as a function of temperature is considered in the governing equations and the relevant sub-models. The effective heat transfer coefficient calculated in the model is also used to determine the amount of fuel to be treated as vapor when the drop surface temperature reaches the critical temperature while the drop interior is still in the sub-critical condition.

Effects associated with spray/wall interactions, including droplet splash, film spreading due to impingement forces, and motion due to film inertia were considered in a wall film sub-model, in addition to calculations of film transport on complex surfaces with heating and vaporization of the film, and separation and re-entrainment of films at sharp corners [24]. For the turbulence calculation, the RNG k- ϵ model [25] was used.

In the two-phase transport equations, droplets are treated as point sources and the wall film fuel flow is not resolved on the computational grid. Therefore, it is assumed that in a computational cell where droplets or wall film parcels exist, the liquid vaporizes under the prevailing mixture conditions and the vapor mixes completely with the gaseous mixture within the cell. Thus, stratification of gaseous species within a single cell is not resolved.

COMBUSTION MODELS

The ignition/combustion characteristics of automotive fuels are often represented using blends of two hydrocarbons (e.g., primary reference fuel), i.e., iso-octane (iC_8H_{18}) and n-heptane (C_7H_{16}). It is widely accepted that the oxidation processes of n-heptane and iso-octane well represent the ignition and combustion characteristics of diesel and gasoline fuels, respectively. In the present study, a skeletal reaction mechanism for primary reference fuel oxidation with 41 species and 130 reactions [26] was used to calculate the detailed chemical kinetics of combustion. The mechanism has been well validated using data from HCCI [27] and direct injection engine experiments [14, 28], as well as with the ignition delay time data obtained in shock tube tests [29] for various temperatures, pressures and fuel compositions. In the present study, gasoline was modeled as PRF 87, i.e., 87 % iso-octane and 13 % n-heptane for physical properties and chemistry calculation.

For the calculation of NO_x formation, a 4 species (N, NO, N₂O and NO₂) and 12 reaction NO_x mechanism was used that has been reduced from the GRI NO_x mechanism (available online) and added to the PRF reaction mechanism. A phenomenological soot model, modified from the Hiroyasu soot model [30] was employed to predict soot emissions. For oxidation of soot, the Nagle-Strickland-Constable (NSC) model was employed in the soot model. Assuming a well-stirred reactor in each cell, changes of species concentration

were obtained from the chemical reaction calculations, which are directly integrated with the transport calculations in the CFD code.

Table 1. Engine and injector specification and simulated operating conditions. Numbers in parenthesis are the parametric variation.

Engine	
Type	light-duty, reentrant bowl piston
Bore x Stroke [mm]	82 x 90.4
Connecting rod length [mm]	161
Compression ratio	16.5
Valve timing [° atdc]	IVO=344, IVC= -132, EVO=112, EVC=388
Injector	
Number of hole	7
Included angle [degree]	155
Nozzle hole diameter [μm]	141.4
Injection timings [° atdc]	1 st pulse: -350 2 nd pulse: -40 ~ +5
Injection pressure [bar]	860 (500, 1200, 1500)
Injection amount [mg/cycle]	43.6
Operating conditions	
Engine speed [rev/min]	2500
Net IMEP [bar]	16.0
Swirl ratio	2.2
temperature at IVC [K]	343 (358)
pressure at IVC [bar]	3.25 (2.8)
O ₂ fraction @ IVC [%]	10.4 (11.5, 12.5)
fuel	Gasoline (ON=87)
Injection strategy	double
Pilot pulse split ratio [%]	40 ~ 100
Computation crank angle [° atdc]	IVC ~ EVO

COMPUTATIONAL CONDITIONS

A small-bore light-duty diesel engine with a compression ratio of 16.5 [31] was used for the simulations. Split injections through a 7-hole injector with an included spray angle of 155 ° were modeled. In the computation, the first pulse injection is assumed to be made such that the injected fuel vaporizes completely and mixes with air uniformly before IVC timing. Injection timings of the second pulse were varied between -40 and +5 ° atdc.

The initial and boundary conditions for a baseline engine operation were first obtained from the measured data of corresponding engine operation using diesel fuel [32] assuming that similar engine load will result in similar engine efficiency and thermal boundary and initial conditions. Then, the initial and boundary conditions were adjusted according to the feedback information from the measured GDICI engine operations.

The simulated cylinder gas pressure and temperature at intake valve closure (IVC) were set to 3.25 bar and 343 K, respectively, regardless of the injection timings. Note that, instead of intake port conditions, mixture conditions at IVC are used in this paper. It is also notable that, due to heat transfer from the cylinder walls, mixing of fresh air with hot internal residual gases, and a slight compression of the gas mixture during the period between BDC and IVC, the gas temperatures at IVC are normally higher than the intake port gas temperatures. The oxygen fraction at IVC was set to 10.4 %, which is equivalent to about 50% EGR ratio using the simulated exhaust gas composition for stoichiometric mixture. The baseline injection pressure was chosen to be 860 bar. For split injection cases, the proportion of fuel injected during the first pulse (hereafter, mentioned as split ratio), which was assumed to be completely vaporized and uniformly mixed with air before IVC, was varied between 40 and 90 %. Detailed specifications of the engine, injector and computed operating conditions are listed in [Table 1](#).

Note that compression ratio and nozzle hole diameter used in the simulation differ slightly from those of the experimental engine which will be discussed later. These variations are intentional to account for physical effects associated with the submodels used in the simulation and better represent the actual engine geometries.

In order to investigate effects of parametric variation, the initial gas temperature, boost level, EGR ratio and injection pressure were varied from the baseline values. When a parameter is changed the rest of parameters are kept the same as the baseline values. Detailed computational conditions of parametric variation are also listed in [Table 1](#).

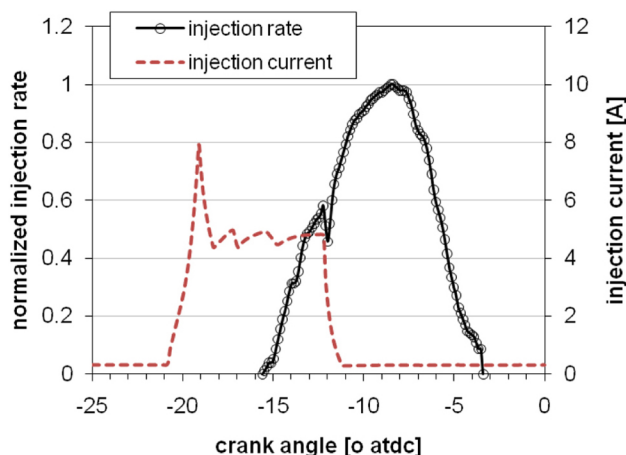


Figure 1. Measured injection rate shape of a single pulse injection. Injection pressure was 860 bar and pulse duration was 575 μ s at engine speed of 2500 rev/min. Rates are normalized by the maximum value during the injection.

In [Figure 1](#), the injection rate shape profile of a single pulse injection is shown for the baseline injection pressure of 860 bar and a pulse duration of 575 μ s at engine speed of 2500 rev/min. The start of injection command of the pulses was -21° . In the figure, rates are normalized by the maximum value during the injection. When injection amount for each pulse was changed as the split ratio varied, the duration of each pulse was changed accordingly based on the measured injection rate profiles. The pressure wave interaction between the pulses was neglected since the pulse dwell is long enough (~ 20 ms) for such wave interactions to damp out.

Three dimensional computational grids with the crevice volume resolved as an elongated top land region were employed. A 7-hole injector with an included angle between sprays of 155 degrees was modeled. To save computation load, a 1/7th sector of the full 360° mesh with periodic boundaries (corresponding to one plume from the seven-hole injector nozzle) was used. The average cell dimensions were 1.2 to 1.8 mm and 0.6 to 4.1 mm in the radial and vertical directions, respectively, with twenty cells azimuthally (see [Figure 2](#)). To resolve the crevice region, i.e., the gap between the piston and cylinder wall above the top ring, two radial cells were used with three vertical cells. This grid resolution was found to be fine enough to ensure grid insensitivity of the spray sub-models and the combustion model.

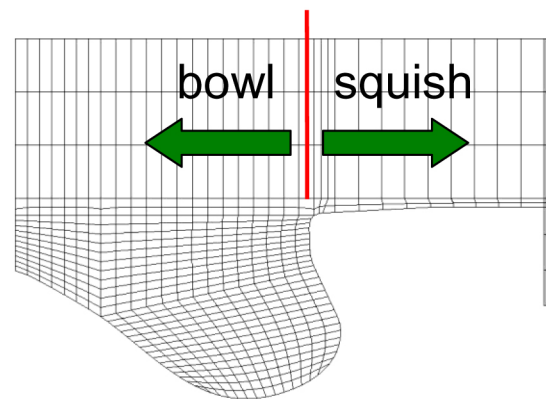


Figure 2. Vertical cross-section view of computational grid with crevice volume resolved as a thin annulus. Azimuthal angle sector span is 51.4 degrees.

EXPERIMENTAL SETUP

The engine used in this study is a single- cylinder version of a GM 1.9L, 4 cylinder light-duty diesel engine; the specifications of which can be seen in [Table 1](#). The production head and valve-train were used in conjunction with a Labeco CLR crankcase. A reentrant bowl design with lower compression ratio than the current Euro-4 compliant production configuration was used along with a slightly longer connecting rod length; all other cylinder geometry was matched to the equivalent 4-cylinder engine. The single-

cylinder engine has the capability to run up to 4000 RPM at full-power conditions using pressurized inlet air and an exhaust backpressure valve to simulate the behavior of a turbocharger. Full capability for variation of EGR rate, coolant and oil temperature and respective inlet and exhaust pressures and temperatures exists in the laboratory. [Figure 3](#) shows the schematic of the engine and test cell layout.

The injection system utilizes a BOSCH CP3.3 pump for supplying fuel rail pressure. ETAS INCA calibration software used in conjunction with the production ECU allows for up to 5 injections per cycle. A production CRIP2-MI injector was used with an experimental $7 \times 155^\circ$ mini-sac nozzle tip with a flow number of 440 cc/30sec. [Table 2](#) summarizes the injection system specifications.

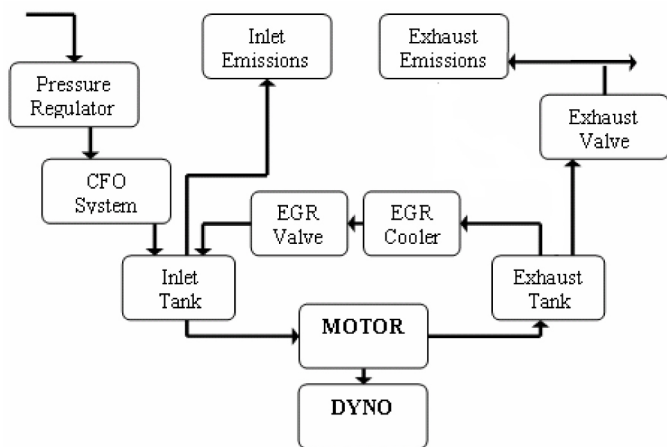


Figure 3. Schematic of test cell layout. CFO stands for Critical Flow Orifice.

The measurement of cylinder pressure data was accomplished via a piezoelectric KISTLER 6125B pressure transducer. The resolution of data was 0.1 CAD, corresponding to 3600 counts per revolution; 200 cycles of data were recorded for all heat release analysis. The combustion noise was recorded using an AVL Combustion Noise Meter (Model A01); meter output was in decibels. Injector current was measured using a Tektronix A622 AC/DC current probe. Exhaust emissions were measured using a Horiba emissions bench capable of recording CO₂, CO, (Model AIA 220), O₂ (Model FMA 220), THC (Model FIA 236 1, propane span gas) and NO_x (Model CLA 220). In addition, a reflectivity-based PM measurement was taken using an AVL Smoke Meter (Model 415S) at each operational point. The emissions sample was extracted downstream of both the exhaust surge tank and the exhaust backpressure valve. In order to obtain the real-time mass percentage of EGR, measurements on the same Horiba emissions bench were done for inlet CO₂ (Model AIA 210) and inlet O₂ (Model MPA 21A). All emissions equipment used heated lines and filters except the intake CO₂ and O₂.

The fuel used in the experiments was intended to be a typical Gasoline with Octane number of 87. The general properties of this fuel can be seen in [Table 3](#).

TEST CONDITIONS

The baseline GDICI engine operational conditions were guided by numerical simulations and are listed in [Table 4](#). The inlet oxygen concentration was set to 11.5 % $\pm$ 0.05%. Being combined with the residual gas, this gives the oxygen concentration of $\sim$ 10.7% in the cylinder at IVC. The injection delay between the command and actual injection timings is about 360 μ s, which is equivalent to 5.4 $^\circ$ CA at 2500 rev/min.

Table 2. Injection System Specifications

Fuel Injection Equipment		
Design	Common Rail	-
Injector	Bosch CRIP 2	-
Included Cone Angle	155	Deg
Nozzle Type	Mini-Sac	-
Nozzle Flow Number	440	cc/30sec
Nozzle Hole Diameter	0.139	mm
Number of Holes	7	-
Operated Rail Pressures	860, 1000, 1160	Bar
Peak Rail Pressure	1600	Bar
Fuel Pump	Bosch CP3.3	-

Table 3. Gasoline fuel properties for testing

Fuel Properties for the Typical Gasoline	
Spec Gravity	0.737
LHV [MJ/kg]	43.2
H/C ratio	1.88
Ethanol [%]	0
(RON+MON)/2	87
Temp at 10% Distilled [$^\circ$ C]	69.4
Temp at 50% Distilled [$^\circ$ C]	105.0
Temp at 90% Distilled [$^\circ$ C]	160.6