

X-ray Imaging of Cavitation in Diesel Injectors

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ABSTRACT

Cavitation plays a significant role in high pressure diesel injectors. However, cavitation is difficult to measure under realistic conditions. X-ray phase contrast imaging has been used in the past to study the internal geometry of fuel injectors and the structure of diesel sprays. In this paper we extend the technique to make in-situ measurements of cavitation inside unmodified diesel injectors at pressures of up to 1200 bar through the steel nozzle wall. A cerium contrast agent was added to a diesel surrogate, and the changes in x-ray intensity caused by changes in the fluid density due to cavitation were measured. Without the need to modify the injector for optical access, realistic injection and ambient pressures can be obtained and the effects of realistic nozzle geometries can be investigated. A range of single and multi-hole injectors were studied, both sharp-edged and hydro-ground. Cavitation was observed to increase with higher rail pressures. Comparative analysis of several injectors indicates that rounding the nozzle inlet delays the onset of cavitation due to reduced separation, but does not always suppress it. Tapering the nozzle hole is an effective means of suppressing cavitation. A single-hole injector mesh was designed based on x-ray imaging, and high resolution three dimensional large eddy simulations were performed using a homogeneous relaxation model. A two-phase submerged simulation was compared against a three-phase compressible solver modeling both non-condensable ambient gas and cavitation. Projections showed good agreement between the three-phase solution and x-ray experiments.

CITATION: Duke, D., Swantek, A., Tilocco, Z., Kastengren, A. et al., "X-ray Imaging of Cavitation in Diesel Injectors," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1404.

INTRODUCTION

Injection pressures used in modern direct injection diesel engines have increased in order to improve emissions, efficiency and performance. Under these conditions, cavitation plays an increasingly important role in the internal fluid mechanics of the nozzle, causing both mechanical wear and altering the formation of the spray [1]. In turn, changes in spray behavior feed back into combustion efficiency and pollutant formation [2, 3].

Manufacturers use two main design methods to attempt to suppress cavitation. Hydro-grinding rounds off sharp corners which would otherwise act as nucleation sites due to flow separation. The other approach is to taper the nozzle hole, which increases the static pressure at the nozzle inlet.

Cavitation has been extensively investigated by installing view ports in injectors and by studying large-scale transparent models, so the complex cavitation behaviour of injectors at more modest pressures with matched Reynolds and Cavitation Numbers has been quite extensively studied [4, 5, 6]. The effects of geometric variations such as nozzle taper have also been investigated in large-scale model injectors [7]. Recently, real-size injectors have also been successfully modified for optical access, to allow flow visualisation of cavitation at typical engine-relevant injection pressures [8, 9, 10].

Numerical simulations have also been extensively used to study cavitating nozzle flow [6, 11, 12, 13, 14]. However, additional experimental data is always needed to validate the wide range of available models. Ideally, data needs to be obtained which matches the geometry, boundary and initial

conditions of unmodified production injectors. However, the nature of cavitation and its relationship with nozzle geometry under realistic operating pressures in real-scaled, unmodified nozzles is difficult to measure. The refractive indices through cavitation observed in transparent injectors causes traditional imaging techniques to saturate at relatively low vapour fractions. It is also difficult to determine how much deviation in cavitation behaviour might be expected in a production injector as compared to the results of numerical and experimental models, particularly over a wide range of nozzle geometries and operating pressures.

X-ray diagnostics have been used for some time to study dense multi-phase flows, as they are not encumbered by the optical density and access limitations of visible-light measurements [15]. Synchrotron x-ray radiography has been successfully used to measure mass distribution in fuel sprays [16, 17]. CT scanning techniques and time-resolved phase contrast propagation methods using hard synchrotron x-rays have been used to make measurements of nozzle geometry and dynamics [18, 19]. For example, Bauer et al [20] used a CT scanner to measure cavitation in a scaled up pipe flow. X-ray radiography techniques have also been used to validate numerical simulations of cavitating gasoline-ethanol blends in transparent nozzles [21, 22]. However, techniques such as CT scanning and absorption radiography are not capable of measuring *in situ* cavitation through a steel nozzle. This is because the absorption of x-rays in the fuel is very weak, and the absorption in the steel nozzle wall is very strong, leading to very poor contrast. Furthermore, the required exposure time for a dynamic system such as a fuel injector is very short (on the order of microseconds), whereas CT scanning systems require a static sample and build up the measurement over a long integration time.

Measuring cavitation inside an unmodified injector poses an additional set of challenges. Phase-contrast images are generated by two primary means of interaction between the x-ray beam and the sample; extinction of the beam due to absorption, and the propagation of diffraction caused by interference when the x-ray is phase-shifted due to a density gradient at the edge of a feature, i.e. 'diffraction edge

enhancement' [23]. In terms of extinction, the fuel is almost transparent to x-rays of sufficient photon energy to penetrate the steel nozzle, so the contrast difference between a fully liquid region and a void space is negligible. Phase contrast between fluid phases is also very weak compared to the strong interference at the nozzle walls as the change in refractive index of the x-rays is much stronger for the steel [19].

In past x-ray radiography experiments, contrast has been increased with a cerium-based additive, which absorb soft x-rays strongly [24]. The K_{1s} edge of Ce at 40.4keV also enhances the contrast of fuel to more energetic x-rays, which are within the regime with appreciable penetration through the nozzle, but would otherwise not be absorbed in fuel. In a cavitation void, which has almost zero pressure relative to the surrounding liquid, there will correspondingly be very little additive, and this difference can be measured.

We have undertaken time-resolved x-ray phase contrast measurements of cavitation in unmodified diesel injectors. Injection pressures of $P_1 = 300 - 1200$ bar have been considered. A range of nozzle geometries have been compared, allowing cursory comparisons to be drawn on the effectiveness of hydro-grinding and nozzle hole taper in a set of both single and multi-hole nozzles. The effect of chamber pressure P_2 has also been investigated at higher back pressures of 5 and 20 bar for a single-hole nozzle.

The results for a single-hole nozzle at $P_1 = 1200$ bar, $P_2 = 20$ bar have also been compared against high resolution three dimensional large eddy simulations (LES). Two homogeneous relaxation models [11] have been employed; a two-phase solver in which the nozzle is submerged in fuel, the liquid phase is considered incompressible and only cavitation vapor is considered, and a fully compressible three-phase solver in which the effects of non-condensable gas outside the nozzle on the internal flow field have been considered. A comparison with x-ray phase contrast imaging shows that the three-phase solver more accurately models the vapor distribution in the nozzle. This demonstrates the effectiveness of the experimental technique in providing validation for numerical models under realistic conditions.

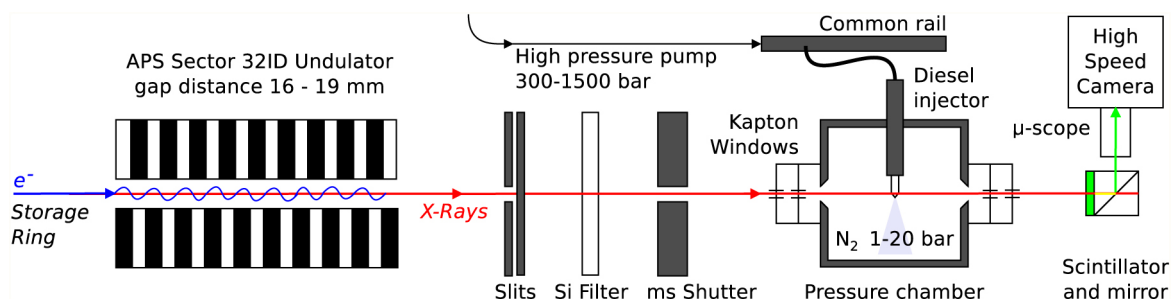


Figure 1. Schematic of experiment conducted at Sector 32ID of the Advanced Photon Source. The diagram is not to scale.

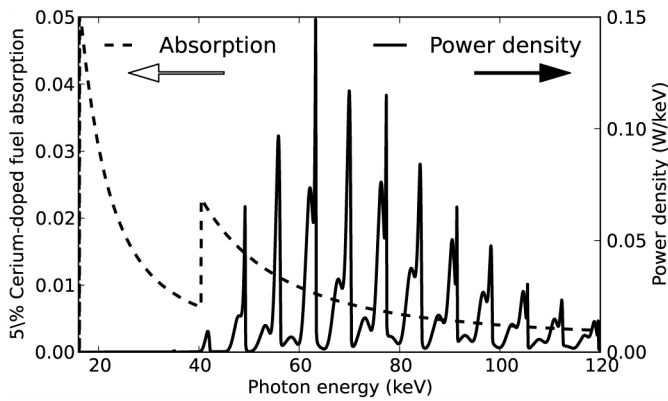


Figure 2. Computed x-ray power spectrum from the APS 32-ID beamline at a nominal ID gap distance of 17mm. The total spectrum at the detector (left vertical axis) and the observed power loss due to absorption of 150 μm of undoped and doped fuel (right vertical axis) are shown, as compared to a nozzle with no fuel.

EXPERIMENTAL METHOD

The experiment setup is shown in Figure 1. A common-rail diesel injection system powered by an electrically driven mechanical pump delivers fuel to a diesel injector, which can be fitted with a range of different injector nozzles (see Table 1). The fuel was sprayed into a chamber pressurized with N_2 up to 20 bar. Kapton windows allow the x-rays to pass through the chamber with minimum absorption. The experiments were conducted at room temperature. High chamber pressure experiments were set at pressures that match the gas density at typical engine start-of-injection conditions. In all experiments the injection duration was 1ms and the injector needle was raised to its fully lifted position.

The fuel used was a commercial diesel surrogate with approximately 5% (by volume) Ce additive. The fuel with additive was characterised to determine whether the additive had a significant effect on density, viscosity or vapor pressure. The diesel surrogate with cerium additive has a vapor pressure of approximately 190 Pa, a density of 866.6 kg/m^3 , a viscosity of 3.24 cSt and a surface tension of $2.89 \times 10^{-2} \text{ N/m}$ at 25°C. The vapor pressure of the Cerium additive is approximately 500 Pa at 25°C, several times higher than the mixture vapor pressure. The additive is dissolved in the fuel and homogeneously mixed. The time scales for flow within the nozzle are very short, such that there is insufficient time for each component in the mixture to achieve an equilibrium state. Therefore, it is unlikely that the additive will vaporise before the mixture locally reaches vapor pressure. Furthermore, the density in void regions is sufficiently low as to make any remaining additive undetectable. The additive density may therefore be used as a marker for fuel density under cold spray conditions.

The experiments were conducted at the 32-ID beam-line (APS Undulator A) of the Advanced Photon Source at Argonne National Laboratory [25]. The undulator gap was set between 16-19 mm, with a peak irradiance of $10^{14} \text{ ph/s/mm}^2/0.1\% \text{ BW}$ at

17 mm. The distance from the source to the experiment was 35.5m. For the low back pressure cases, a 5 mm Si filter was used to remove low-energy photons from the raw x-ray source. A LuAG:Ce scintillator [26] of 100 μm thickness converts the x-rays to a visible light image, which was recorded by a Photron high-speed camera at 10 kHz fitted with a 10 $\times$ long-distance microscope. The propagation distance from the experiment to the scintillator was 630 mm. The spatial resolution achieved with this system was 1.9 μm per pixel, giving a resolution of 70 to 110 pixels across the nozzle hole diameter for the different injectors studied.

A predicted x-ray spectrum at the detector computed with the XOP software package [27] is shown in Figure 2, accounting for the Si filter, losses in the chamber gas, the nozzle walls, and the scintillator. The simulated spectrum allows the effective contrast of the fuel additive to be estimated with respect to the scintillator response. The presence of a 150 μm film of liquid octane (a nominal maximum path length in the nozzle of a fuel injector) yields only a 0.25% change in detectable beam power. This is below the detectable limit for the imaging system. When 5% (by vol.) Ce is added to the fuel, the detectable change in power increases by a factor of 3. The additive absorbs approximately 1% of the incoming beam at useful photon energies (those capable of penetrating the steel). The increase in contrast due to the additive improves the signal to noise ratio such that imaging a thin film of cavitation vapor becomes feasible. It should be noted that the response of the imaging system to the quantity of vapor in the beam path is nonlinear due to the polychromatic nature of the light source, and as such the absolute quantity of vapor cannot be explicitly quantified. However, changes in recorded intensity may be effectively correlated with the spatial distribution of vapor.

NUMERICAL METHOD

In order to validate the x-ray experiments, a series of Large Eddy Simulations (LES) have been performed. The solver used in the following study is the Homogeneous Relaxation Model (HRMFoam) for cavitating and flash-boiling flows, which has been developed on the OpenFOAM platform [28]. Two variations of this solver have been employed. The first is a simplified model which neglects compressibility and the presence of non-condensable gas on the liquid phase. A complete description of the solver and validation against experimental throttle measurements can be found in Neroorkar et al [11]. The second solver is fully compressible and also models the effect of non-condensable gas (i.e. the chamber gases outside the nozzle). It should be noted that the non-condensable gas model does not include a spray breakup model, but focuses on the effect of non-condensable gas on the internal flow.

The HRM assumes that the vapor and liquid fractions of the fuel are sufficiently mixed that they may be treated as a single fluid with one set of governing equations. Since phase change is occurring, the fluid may not be in thermodynamic equilibrium, and an equation of state cannot be used as a closure.

assumption. Instead, the model assumes that the instantaneous local vapor mass fraction (x) will relax towards equilibrium conditions over some timescale θ [29];

$$\frac{Dx}{Dt} = \frac{\bar{x} - x}{\theta}, \quad (1)$$

where $\bar{x}$ is the equilibrium vapor mass fraction. The value of $\bar{x}$ is calculated based on the local pressure and enthalpy [30]. The instantaneous mass fraction is calculated from the vapor volume fraction, density and vapor density as $x = \alpha \rho_v / \rho$, where the vapor volume fraction is defined as

$$\alpha = \frac{\rho_l - \rho}{\rho_l - \rho_v}. \quad (2)$$

The time scale θ is calculated via an empirical relation proposed by Downar-Zapolski et al. [31];

$$\theta = \theta_0 \alpha^{-0.54} \psi^{-1.76}, \quad (3)$$

$$\psi = \frac{P_{sat} - P}{P_{sat} - P_{crit}}, \quad (4)$$

where the empirical constant $\theta_0 = 3.84 \times 10^{-7}$ s [11].

The mass, momentum and energy conservation equations for the fuel are expressed in terms of mass flux $\phi = \rho \mathbf{u}$;

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \phi = 0, \quad (5)$$

$$\frac{\partial \rho \mathbf{u}}{\partial t} + \nabla \cdot (\phi \mathbf{u}) = -\nabla P + \nabla \tau, \quad (6)$$

$$\frac{\partial \rho h}{\partial t} + \nabla \cdot (\phi h) = \frac{\partial P}{\partial t} + \mathbf{u} \cdot \nabla P, \quad (7)$$

where τ is shear stress. Due to phase change, Eqs. 5, 6, 7 are not closed, so Eqs. 1, 2, 3, 4 are implemented to deal with the additional terms. The off-diagonal terms are represented by $H(\mathbf{u})$ and the diagonal terms by a , such that $a_j u_j = H(\mathbf{u}) - \square P$ [29]. The implicit pressure equation is expressed in its final form as

$$\left. \frac{1}{\rho} \frac{\partial \rho}{\partial P} \right|_{x,h} \left(\frac{\partial \rho P}{\partial t} + \nabla \rho P \phi_v \right) + \rho \left(\frac{H(\mathbf{u})}{a_j} \right) - \rho \nabla \cdot \frac{1}{a_j} \nabla P + \left. \frac{\partial \rho}{\partial x} \right|_{P,h} \frac{Dx}{Dt} = 0. \quad (8)$$

A complete derivation of the pressure equation is given in [32]. The first two terms concern compressibility and are neglected in the incompressible solver since they result in an asymmetric linear system for pressure. This assumption has previously been shown to yield reasonable results [11, 21, 29, 33]. The compressible solver uses the complete pressure equation.

The compressible solver also includes a transport equation for non-condensable gas. The presence of non-condensable gas results in a new expression for density. Analogous to the fuel quality x , the non-condensable gas quality y is introduced,

$$x \equiv \frac{M_v}{M_f}, \quad y \equiv \frac{M_g}{M_{tot}}, \quad (9)$$

where M_f , M_v , M_l and M_g are respectively the fuel, vapor, liquid and non-condensable gas mass fractions, and M_{tot} is the total mass. Then, from the volume composition of the fluid,

$$\begin{aligned} V_{tot} &= V_v + V_l + V_g, \\ \frac{M_{tot}}{\bar{\rho}} &= \frac{M_v}{\rho_v} + \frac{M_l}{\rho_l} + \frac{M_g}{\rho_g}, \\ \frac{1}{\bar{\rho}} &= \left(\frac{M_v}{\rho_v M_f} + \frac{M_l}{\rho_l M_f} \right) \frac{M_f}{M_{tot}} + \frac{M_g}{\rho_g M_{tot}}, \\ \frac{1}{\bar{\rho}} &= \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right) (1-y) + \frac{y}{\rho_g}. \end{aligned} \quad (10)$$

Applying the chain rule to Eqn. 10 yields,

$$\frac{\partial \bar{\rho}}{\partial x} = \left(\frac{1}{\rho_v} - \frac{1}{\rho_l} \right) (y-1) \bar{\rho}^2, \quad (11)$$

which is then substituted into Eqn. 8.

The pressure equation is solved using a pressure-implicit split-operator (PISO) predictor-corrector algorithm. Turbulence is modeled in this study using a single k -equation large eddy simulation [28, 34]. Discretization schemes are second order, using gamma differencing for the ρ , P , h and $\mathbf{u}$ divergence terms, and upwind differencing for k divergence. Laplacians are discretized using second-order central differencing. The pressure field is solved using the fast simplified diagonal-based incomplete Cholesky pre-conditioner and a generalized geometric-algebraic multi-grid method. ρ , h , k and $\mathbf{u}$ are iteratively solved using a simplified diagonal-based incomplete LU pre-conditioner and a preconditioned bi-conjugate gradient method for the final step. The solver was run on the *Fusion* cluster facility at Argonne National Laboratory.

Table 1. Nozzles investigated in this study. (VCO = Valve-covered orifice, K-factor = $(D_{in} - D_{out})/10 \mu m$.)

Nozzle	Type	No. Holes	Hydro-grinding	Taper K-factor	Measured Inlet hole dia. (μm)	Hole entrance angle	Mean entrance radius (μm)	Cavitation observed
#1	Mini-sac	1	no	0	206	0°	4-6 (sharp)	yes
#2	Mini-sac	1	no	0	208	0°	4-6 (sharp)	yes
#3	Mini-sac	1	yes	0	132	0°	50-60 (round)	yes
#4	Mini-sac	1	yes	1.5	132	0°	35-45 (round)	no
#5	Mini-sac	3	yes	1.3	164	73°	20-25 (round)	no
#6	VCO	3	yes	1 - 1.2	164	80°	10-32 (round)	unclear
#7	VCO	3	no	0	185	80°	2-4 (sharp)	yes
#8	Mini-sac	7	yes	1.3	153	72°	40-80 (round)	no

For the submerged nozzle simulations, the only vapor present in the domain is due to cavitation and the vapor volume fraction $\alpha = V_v/V_f$ is suitable to quantify the degree of cavitation. However, for the compressible simulations with non-condensable gas, the vapor volume fraction alone is not a useful indicator since cell volumes having a high vapor volume fraction but also high non-condensable mass fraction y do not represent significant cavitation vapor; the fuel is highly dilute in these regions. As such, a corrected vapor fraction which accounts for only physically relevant vapor fractions must be defined in terms of the local mass fractions and densities.

The mass fractions of vapor and non-condensable gas (Eqn. 9) are normalized differently; by the fluid mass and total mass respectively. We thus define a corrected mass fraction of vapor x^* which is normalized by total mass;

$$x^* \equiv \frac{M_v}{M_{tot}} = x \left(\frac{M_{tot} - M_g}{M_{tot}} \right) = x(1 - y). \quad (12)$$

We now also define a corrected vapor volume fraction. First, β is defined as the volume fraction of the non-condensable gas

$$\beta = \frac{V_g}{V_{tot}} = \frac{M_g/\rho_g}{M_{tot}/\rho} = y \left(\frac{\rho}{\rho_g} \right), \quad (13)$$

and the corrected vapor fraction can be defined in terms of β as the volumetric analogy of the corrected mass fraction in Eqn. 12;

$$\alpha^* = \frac{V_v}{V_{tot}} = \frac{V_v}{V_f} \frac{V_f}{V_{tot}} = \alpha \left(\frac{V_{tot} - V_g}{V_{tot}} \right) \Rightarrow \alpha^* = \alpha(1 - \beta). \quad (14)$$

RESULTS - X-RAY PHASE CONTRAST IMAGING

A qualitative summary of the results for all nozzles imaged is given in Table 1. A raw sample x-ray image of nozzle #1 (an axial, single hole nozzle) is shown in Figure 3 with typical features indicated. Only the entrance of the nozzle is shown in the image. To observe changes in additive concentration due to

cavitation, the median background was removed from the raw images, and nozzle vibration effects were also subtracted using cross-correlation tracking of the sac wall [24, 35]. At high injection pressures, the opening of the needle causes axial deflections of the nozzle on the order of several microns due to the mechanical stress acting on the nozzle tip.

The phase contrast effects due to the nozzle geometry are mostly removed from the signal in the background subtraction. Once the background is removed, a bright region is observed near the nozzle wall, which is only visible during the injection event (Figures 4a,4c). This intensity increase is attributed to cavitation vapor near the wall; since there is less fuel and contrast agent in this region, absorption of x-rays will be reduced. Three samples of single hole nozzles, both sharp-edged and hydro-ground (#1-3), show near-wall intensity increases indicative of cavitation vapor. The cavitation appears to begin farther downstream for the hydro-ground nozzle due to the rounded inlet edge, but with similar intensity to the non-ground nozzle. However, a tapered single hole nozzle (#4) did not show any repeatable cavitation-like behavior.

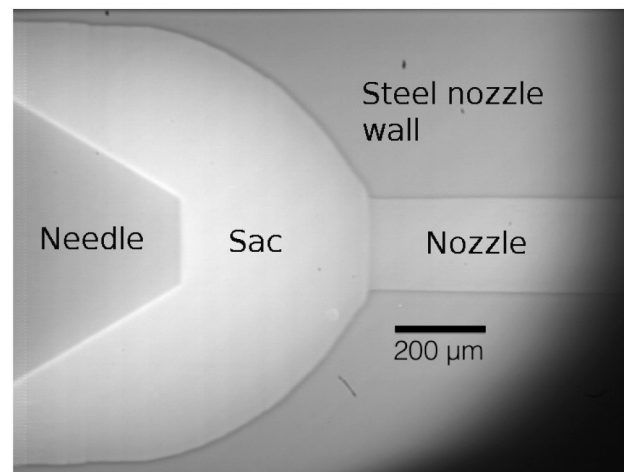


Figure 3. Raw x-ray image of nozzle #1. The needle (a), sac (b), nozzle (c) and steel wall (d) are labelled. The fuel flow is left to right.

The spatial extent of the bright regions indicative of cavitation did not change significantly with injection pressure, but the relative intensity increased as injection pressure was increased. Since change in intensity is inversely correlated with fluid density, an increase in local intensity with injection pressure suggests increasing vapor fraction in cavitating regions with increasing injection pressure.