



City Research Online

City, University of London Institutional Repository

Citation: Koukouvinis, P., Gavaises, M., Li, J. & Wang, L. (2016). Large Eddy Simulation of Diesel injector including cavitation effects and correlation to erosion damage. *Fuel*, 175, pp. 26-39. doi: 10.1016/j.fuel.2016.02.037

This is the accepted version of the paper.

This version of the publication may differ from the final published version.

Permanent repository link: <https://openaccess.city.ac.uk/id/eprint/15670/>

Link to published version: <https://doi.org/10.1016/j.fuel.2016.02.037>

Copyright: City Research Online aims to make research outputs of City, University of London available to a wider audience. Copyright and Moral Rights remain with the author(s) and/or copyright holders. URLs from City Research Online may be freely distributed and linked to.

Reuse: Copies of full items can be used for personal research or study, educational, or not-for-profit purposes without prior permission or charge. Provided that the authors, title and full bibliographic details are credited, a hyperlink and/or URL is given for the original metadata page and the content is not changed in any way.

City Research Online:

<http://openaccess.city.ac.uk/>

publications@city.ac.uk

1 Large Eddy Simulation of Diesel Injector including cavitation
2 effects and correlation to erosion damage

3 Phoevos Koukouvinis^{*,a}, Manolis Gavaises^a, Jason Li^b, Lifeng Wang^b

4 ^aCity University London, Northampton Square EC1V 0HB, United Kingdom

5 ^bCaterpillar Inc, Mossville, IL 61552 US

6 ^{*}Corresponding author, tel: +44(0)7561883907, email: foivos.koukouvinis.1@city.ac.uk

7

8

9

10

11

12

13

14

15

16

17

18

Large Eddy Simulation of Diesel Injector including cavitation effects and correlation to erosion damage

Phoevos Koukouvinis^{*,a}, Manolis Gavaises^a, Jason Li^b, Lifeng Wang^b

^aCity University London, Northampton Square EC1V 0HB, United Kingdom

^bCaterpillar Inc, Mossville, IL 61552 US

*Corresponding author: foivos.koukouvinis.1@city.ac.uk

Abstract. The present paper focuses on erosion development due to cavitation inside Diesel injectors. Two similar injector designs are discussed both in terms of numerical simulation and experimental results from X-ray CT scans. In order to capture the complex flow field and cavitation structures forming in the injector, Large Eddy Simulation along with a two phase homogenous mixture model were employed and compressibility of the liquid was included as well. During the simulation, pressure peaks have been found in areas of vapour collapse, with magnitude beyond 4000bar, which is higher than the yield stress of common materials employed in the manufacturing of such injectors. The locations of such pressure peaks correspond well with the actual erosion locations as found from X-ray scans. The present work's novelty is to correlate pressure peaks due to vapour collapse with erosion development in industrial injectors with moving needle including comparison with experiments.

Keywords: Diesel injector, LES, Cavitation, Erosion, X-Ray CT scans.

1. Introduction

Diesel injection systems play a fundamental role in internal combustion engines since they affect the formation of the fuel spray, atomization and combustion, the formed emissions and the engine efficiency. The jet velocities formed are of the order of 500m/s, with upstream pressures around 2000bar. Current trends show injection pressures to even rise to 3000bar, in order to meet the future EU legislations in emissions. However, higher pressure levels causes very high velocities through the

44 tight passages in the Diesel injector and strong accelerations in sharp direction changes (corners, fillets
45 etc.), which lead to static pressure dropping locally below the saturation pressure and causing
46 cavitation. Furthermore, cavitation may lead to erosion damage and serious degradation of the injector
47 performance, even catastrophic injector failure, which could damage the engine, if the injector tip
48 breaks off.

49 Various researchers have worked on the subject of cavitation development inside Diesel injectors
50 under varying assumptions; Sezal et al. worked on simple 2D axis-symmetric nozzles [1] and 3D
51 nozzles [1, 2] with a fully compressible approach, capable of predicting cavitation collapse pressure
52 peaks that could be linked to cavitation erosion. Salvador et al. have done extensive work on Diesel
53 injector cavitation, starting from validation studies [3], examining various geometrical features [4] and
54 needle lift influence [5] on the flow pattern inside the injector. In continuation of the aforementioned
55 work, Molina et al. [6] examined the influence of elliptical orifices on cavitation formation and
56 Salvador et al. [7] performed LES studies in Diesel injector nozzles using OpenFOAM. However all
57 the aforementioned literature work did not involve needle motion; instead needle was fixed either at
58 full or partial lift. A recent numerical work by Örley et al. [8] on Diesel injectors involves the
59 immersed boundary method, needle motion, compressibility of liquid, vapour and free gas, though the
60 focus is mainly on the developed turbulent structures and less on pressure peak/erosion development.

61 On the other hand, several works have included the needle motion for the prediction of flow pattern
62 inside the injector, however either resorted to using RANS or omitted compressibility effects. For
63 example Patouna [9] focused on the simulation of injectors at steady or moving needle conditions,
64 however the liquid was assumed incompressible and there was no effort to correlate with possible
65 erosion development. Strotos et al. [10] studied the thermodynamic effects of Diesel fuel
66 heating/cooling inside the Diesel injectors at both steady and moving needle conditions, with main
67 interest on next-generation injectors that could reach up to discharge pressures of 3000bar. Devassy et
68 al. [11] implemented a 1D-3D coupling for Diesel injector simulations throughout the whole injection
69 pulse; the 3D simulation involved needle motion and a simplistic liquid compressibility model.

70 There have been several efforts for the prediction of the cavitation erosion in Diesel injectors, see
71 e.g. the work of Gavaises et al. [12] and Koukouvinis et al. [13]. The aim of the current work is to
72 simulate the flow inside a Diesel injector in a more fundamental level, including needle motion,
73 compressibility effects of the liquid phase and also using a Large Eddy Simulation for describing
74 turbulence. Mesh motion is necessary for describing the transient effects in the injector. The reason for
75 employing compressibility effects is that the fuel density can vary as much as 10% within the injector
76 [14], not to mention the high liquid velocities that can reach a Mach number of 0.5 or more.
77 Furthermore, resorting to Large Eddy Simulation techniques is because RANS/URANS are inadequate
78 for capturing the complicate vortex patterns which affect cavitation formation [15], while even
79 modified RANS turbulence models are situational [16]. To the authors knowledge there is no other
80 work in literature that resolves the compressible turbulent flow in a moving needle Diesel injector with
81 LES, including the prediction of vapour collapse pressures and correlation with actual erosion damage
82 from CT scans of actual injectors. Furthermore, the methodology discussed in the present paper
83 involves a modified cavitation model, in order to move closer towards thermodynamic equilibrium; if
84 such a modification is not employed then unphysically high tension is predicted in the liquid.

85 The current paper is organized as follows: first an indicative description of two injector tip
86 geometries will be given, along with testing conditions and X-ray scans of the erosion damage from
87 the endurance test. Then, the numerical methodology will be presented. The simulation results of the
88 Rayleigh collapse of a vaporous bubble is examined as a fundamental test case of the methodology
89 used. Indeed, the aim of the current study is to detect the regions of the collapse of cavitation
90 structures, which is directly linked with the formation of extreme local pressure and therefore erosion
91 damage. Furthermore the simulation results of a simple throttle flow that has been previously studied
92 by Edelbauer et al. [16] will be presented as a more applied benchmark case. Finally, indicative results
93 of the simulated injectors will be shown and will be compared with the X-ray scans from the
94 experiments, showing a good correlation.

95 **2. Description of the examined injectors and testing conditions**

96 *2.1. Injector geometry and operating conditions*

97 The examined injectors are common rail injectors. The accelerated cavitation test is performed in
98 an endurance test rig, located at Caterpillar US research and development centre. Endurance testing is
99 conducted for several thousand hours, with injection pressure at 1.1-1.5 times the injector rated
100 operating pressure. The testing fuel is periodically replaced to maintain quality. The injectors are
101 mounted on the head block of the test rig and the injected fuel is collected by the collector block and
102 the rate tube, with downstream pressure adjusted by the pressure regulator at the end of the rate tube.
103 The test rig also has a heat exchanger to keep Diesel fuel temperature controlled at around 40°C in the
104 fuel tank and a computer which collects data and controls the injection frequency.

105 Two injector designs are examined, which will be referred to as Design A and Design B hereafter.
106 Both injectors have 5 hole tip and share exactly the same needle, as shown in Figure 1. Design A has
107 cylindrical holes (k-factor 0), while Design B injector has slightly tapered holes (k-factor is 1.1).
108 Moreover, Design B has a significantly smaller sac volume comparing to Design A. This characteristic
109 makes the Design B tip somewhat shorter than the equivalent of Design A. A summary of the most
110 important dimensions of the two injectors is given in Table I.

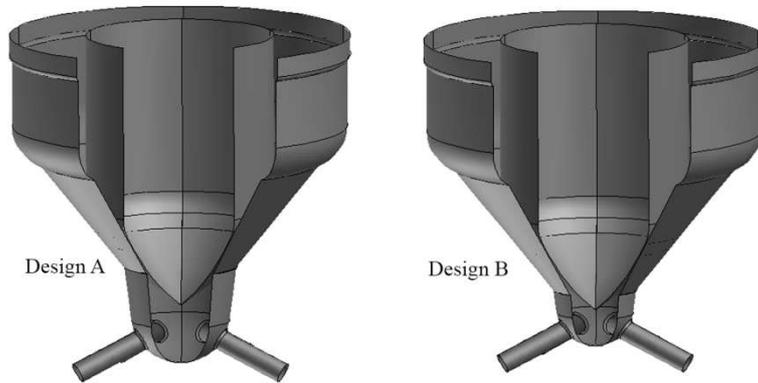
111

112

Table I. Important geometric dimensions of the examined injectors.

Geometric characteristics		Design A	Design B
Needle radius (mm)		1.711	1.711
Orifice length (mm)		1.261	1.262
Orifice diameter (mm)	Entrance - D_{in}	0.37	0.37
	Exit - D_{out}	0.37	0.359
Sac volume (mm ³)		3.35	1.19
$k - factor = (D_{in} - D_{out})/10, D$ in μm		0	1.1

113



114

115

Figure 1. Comparative view of the two designs: Left is Design A and right Design B.

116

117

118

119

120

121

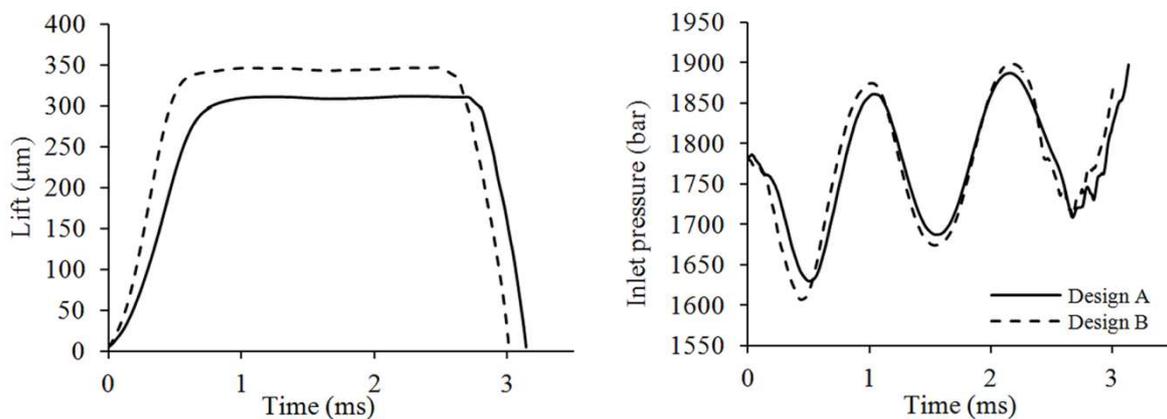
122

123

124

125

The injector operating pressure is ~ 1800 bar with inlet fuel temperature at $\sim 75^\circ\text{C}$. The collector back pressure is ~ 50 bar. Design B injector has a slightly higher needle lift, but shorter injection pulse duration comparing to Design A. The total injection duration is ~ 3 ms. Figure 2 shows the pressure inlet boundary condition and needle motion for the two designs, as predicted using the 1-D system performance analysis software, developed internally by Caterpillar Inc. The 1-D model includes the entire hydraulic circuit of the endurance bench fuel systems as well as the electronic control system. The input parameters of the 1-D model include engine speed, fuel pressure and temperature, injection duration, and regulator back pressure, etc. In the present work, simulation results mainly of the opening phase of the injectors will be presented, i.e. for a lift from 0 to $\sim 300\mu\text{m}$ (for Design A) or $\sim 350\mu\text{m}$ (for Design B).



126

127

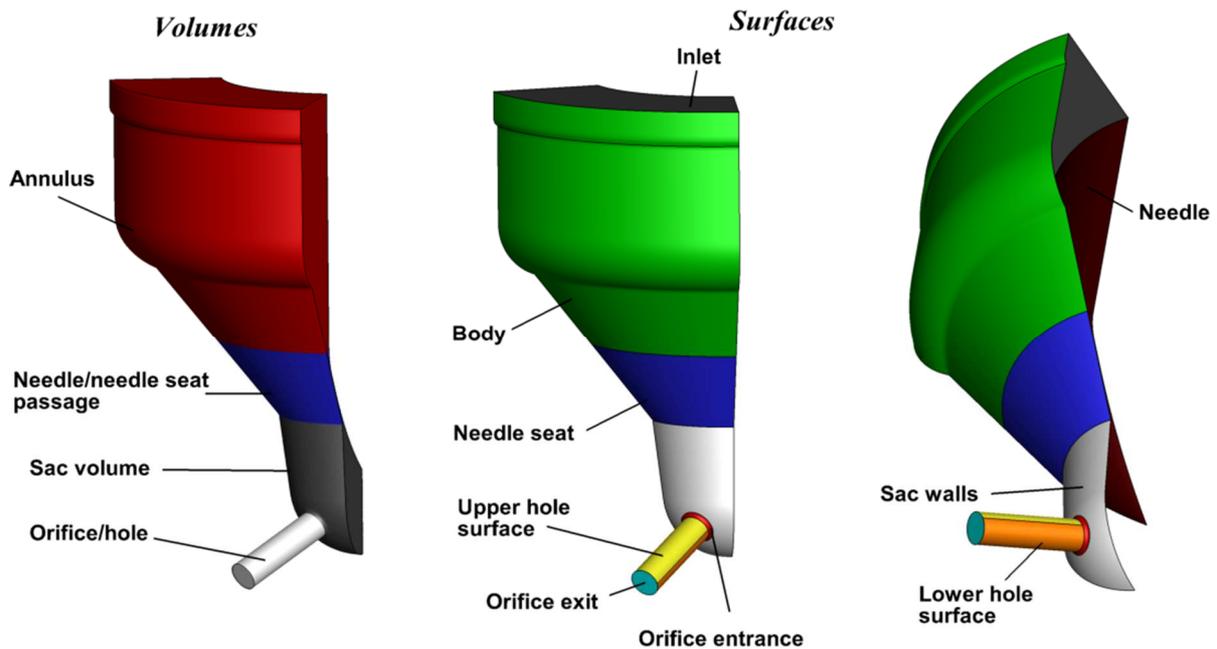
Figure 2. Needle motion and transient pressure inlet boundary condition for the two designs.

128 From hereafter the following naming convention will be used to refer to various injector parts,
129 surfaces and volumes, see also Figure 3:

130 - The injector tip volume is split into several sub-volumes, which can be identified as follows,
131 starting upstream the injector tip and following the fuel flow: *annulus*, *needle/needle seat passage*, *sac*
132 *volume* and *orifice* or *hole*.

133 - The injector tip surfaces are split into the following: the surface of the *annulus* that corresponds
134 to the larger diameter will be referred as *body*. The *needle seat* and the *needle* walls define the *passage*
135 volume. *Sac wall* is bounding the *sac volume*. *Orifice entrance* is the geometrical transition (which is
136 usually a fillet) from the *sac wall* to the *orifice* surfaces. The orifice surface may be split further into
137 the *upper* and *lower surfaces*; here *upper surface* corresponds to the surface that is closer to the inlet,
138 i.e. faces towards the upstream direction, and *lower surface* faces towards the downstream flow
139 direction, i.e. the combustion/injection chamber.

140 For more information on injector operation, components and assembly the interested reader is
141 addressed to [17].



142

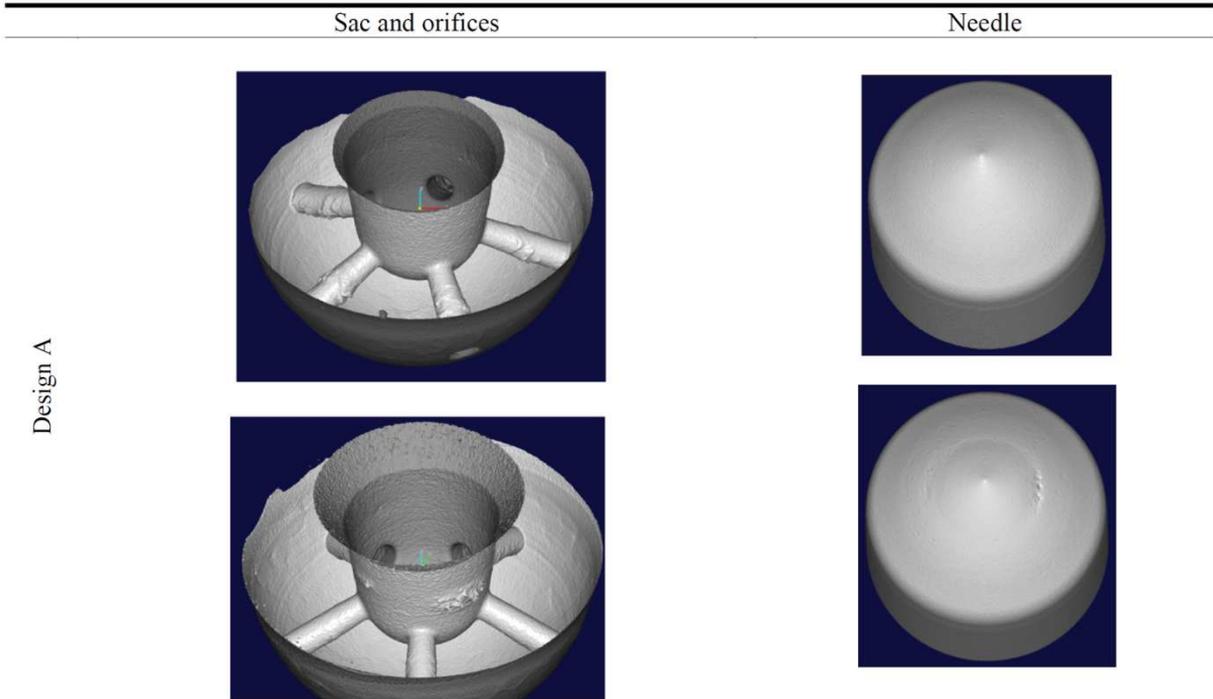
143 Figure 3. Naming convention of various injector sub-volumes (left) and surfaces (middle and right) to be used hereafter.

144

145 2.2. *Injector endurance tests and X-ray erosion patterns*

146 Figure 4 and Figure 5 show the X-ray CT scans of the sac/orifice and needles of four injectors with
147 the same endurance test hours. Figure 4 shows the erosion patterns in two design A injectors, while
148 Figure 5 shows the erosion patterns in two design B injectors. As can be seen from the relevant X-ray
149 scans, both designs are susceptible to cavitation erosion damage. Design A injector has signs of
150 erosion damage inside the sac volume that become apparent rather early, in the order of one thousand
151 hours of continuous operation. Design B injector is less prone to erosion damage, since noticeable
152 damage occurs significantly later, in the order of several thousand hours of continuous operation; even
153 then the damage is minor, in the form of a slight pit near the orifice entrance. Regarding the damage in
154 the nozzle holes, Design B injector is generally less prone to erosion damage, while the cylindrical
155 hole of Design A has signs of damage at thousand hours, which progresses more aggressively with
156 time comparing to Design B. The trend seems to change when considering the needle damage, since
157 Design A needle is almost erosion free; there are only some minor, nearly negligible, signs of erosion,
158 that do not show any change over time. Design B injector needle is more affected by erosion, since a
159 deep indentation is visible in the form of a ring of radius $\sim 0.6\text{mm}$ see Figure 5; however the erosion
160 damage does not seem to progress after formation.

161 The experimental results obtained from the endurance tests suggest that the erosion patterns are
162 consistent for Design B injector, that is a similar erosion trend develops for injectors tested, after
163 similar time intervals. However this is not the case for Design A; even though erosion locations are in
164 general the same, there is discrepancy in the erosion development among the same design after the
165 same time interval. E.g. in Figure 4 the one sac volume seems to be much less affected by erosion
166 damage than the other and on the other hand in one case the injector holes are practically ruined by
167 erosion damage, while the other is barely affected by erosion damage. It is speculated that this effect is
168 related to possible eccentric motion of the injector needle, that could alter the flow pattern inside the
169 injector and consequently cavitation formation, and slight variations of the exact test conditions.

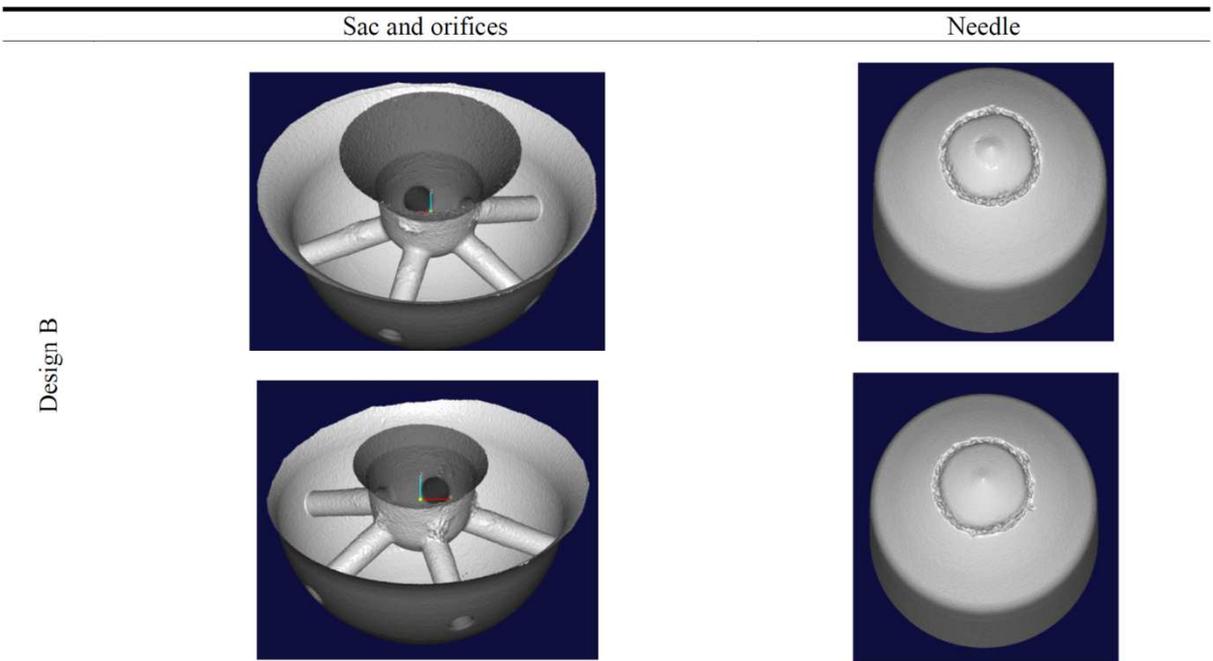


170

171

172

Figure 4. Erosion details at various locations for Design A, as found on the surfaces of two examined injectors after the same operation hours.



173

174

175

Figure 5. Erosion details at various locations for Design B, as found on the surfaces of two examined injectors after the same operation hours.

176 **3. Numerical background**

177 Numerical simulations presented in this work are based on a the solution of the Navier Stokes
 178 equations, using a commercial pressure-based solver, Fluent [18]. The equations solved consist of the
 179 continuity and momentum equations, while the energy equation has been omitted. The reason for
 180 omitting heat effects was the limited applicability of the Diesel properties library currently available
 181 [14]. As will be shown later, local pressures may reach or exceed 9000bar and, due to the polynomial
 182 nature of the Kolev properties library, negative densities may be predicted, which are meaningless;
 183 alternative libraries will be considered in future work as e.g. NIST Refprop [19], but applicability in
 184 such extreme cases is generally not guaranteed. In any case, since Diesel properties vary significantly
 185 with the pressure levels in the injection systems, both liquid phase viscosity and density are assumed
 186 variable, as functions of pressure only. For density, the Tait equation of state was used:

$$187 \quad p = B \left[\left(\frac{\rho}{\rho_{sat,L}} \right)^n - 1 \right] + p_{sat} \quad (1)$$

188 where $\rho_{sat,L}$ is the density at saturation pressure p_{sat} . This equation of state has the advantage that can
 189 handle both large and negative (up to a point) pressures. The values used for the simulations are
 190 summarized in the following table, including the liquid viscosity μ_L :

191
 192

Table II. Liquid phase properties.

Property	Rayleigh collapse	Throttle case	Design A/Design B Injectors (properties estimated at 395K)
$\rho_{sat,L}$ (kg/m ³)	998.2	830	747.65
p_{sat} (Pa)	2340	4500	$1.1 \cdot 10^5$
B (MPa)	300	167	110
μ_L (Pa.s)	10^{-3}	$2.1 \cdot 10^{-3}$	$\log_{10}(10^6 \mu_L / \rho) = 0.035065275 - 0.000234373 p/10^5$

193

194 For all materials the exponent n is set to 7.15, since such values correspond to weakly compressible
 195 materials such as liquids [20]. Properties for the injector flow are considered on an average
 196 temperature level of 395K. This value was estimated through simplified 1D analysis for the pressure
 197 levels in the injector [10], given a range of the discharge coefficient from ~ 0 (valve closed, estimated

198 outlet temperature $\sim 427\text{K}$) to ~ 0.8 (valve fully open, estimated outlet temperature $\sim 359\text{K}$) and is an
 199 estimated average during the injection event; note that the theoretical minimum outlet temperature for
 200 the injectors is $\sim 324\text{K}$, for operation at a discharge coefficient of unity, which would apply for the
 201 ideal case without friction losses. Also, liquid dynamic viscosity is prescribed with a relation provided
 202 by N. Kolev [14], applied for the same temperature level as above.

203 For inclusion of cavitation effects, an additional transport equation is solved for tracking the vapour
 204 phase, of the form:

$$205 \quad \frac{\partial(a\rho_v)}{\partial t} + \nabla(a\rho_v \mathbf{u}) = R_e - R_c \quad (2)$$

206 where a is the vapour fraction, ρ_v is the vapour density, \mathbf{u} is the velocity field and R_e , R_c are the mass
 207 transfer rates for condensation (c) and evaporation (e), prescribed by the Zwart-Gerber-Belamri model
 208 [21]. Vapour properties are set considering the saturation conditions of each material:

209
 210

Table III. Vapour phase properties.

Property	Rayleigh collapse	Throttle case	Injectors
ρ_v (kg/m ³)	0.0171	0.286	6.5
μ_v (Pa.s)	$9.75 \cdot 10^{-6}$	$7.5 \cdot 10^{-6}$	$7.5 \cdot 10^{-6}$

211

212 Here it must be mentioned that while vapour is treated as incompressible, the vapour/liquid mixture
 213 is compressible, due to mass transfer terms; in fact it can be proved that the dominant term affecting
 214 the mixture compressibility is the mass transfer term, see [22]. Moreover, under the assumption of
 215 cavitation formation at approximately constant pressure equal to saturation, the vapour density should
 216 be approximately constant. Of course, possible compressibility effects, such as shock waves, in the
 217 pure vapour phase cannot be captured in this way, but their effect on the results is questionable.

218 The two phase model is a homogenous mixture model that assumes mechanical equilibrium
 219 between the two phases, i.e. both liquid and vapour phase share the same pressure and velocity fields.
 220 The mass transfer model behaves as a non-thermodynamic equilibrium model, since metastable

221 conditions of liquid tension, i.e. negative pressures, may develop. While such scenarios have been
222 found in delicate laboratory experiments, see for example [22-25], it is rather questionable if they are
223 possible to exist in industrial flows and especially the highly violent flow inside a throttle or a diesel
224 injector. For this reason, the mass transfer terms have been increased in order to limit the existence of
225 negative pressures inside the computational domain as much as possible; after the tuning the minimum
226 pressure inside the throttle is approximately -1bar and in the injector is approximately -20bar. Without
227 tuning the liquid tension would be at least one order of magnitude higher.

228 Apart from the simple benchmark case of the Rayleigh collapse, LES methodologies were used for
229 the rest cases, in order to capture the complicated turbulent structures which significantly contribute to
230 the cavitation structures. The throttle case was simulated with the Coherent Structure Model (CSM)
231 [16, 26] in order to be consistent with the relevant published results [16], whereas the injectors were
232 simulated with the Wall Adapted Local Eddy-viscosity (WALE) LES model [27]. Both models are
233 much better behaved in wall-bounded flows, since the eddy viscosity diminishes at the near wall
234 locations, contrary to the standard Smagorinsky model.

235

236 **4. Simulations**

237 *4.1. Collapse of a spherical vapour bubble*

238 Since the aim of the two phase model employed is to predict the Rayleigh collapse of a vaporous
239 structures in the liquid fuel, it is reasonable to test the capability of the model in the prediction of
240 collapse of a spherical vapour bubble in an infinite liquid domain of higher pressure. For this test, a
241 simple 2D-axis symmetric configuration is used involving water at pressure of 1bar and a vapour
242 bubble of $R_0=10\mu\text{m}$ at saturation conditions, i.e. 2339Pa. It is important to mention that the farfield
243 boundary is set at 100 bubble radii away from the bubble; early trials have shown that setting the
244 boundary closer leads to an earlier collapse, due to bias imposed from the boundary. The configuration
245 resembles the well known Rayleigh collapse, where the radius of the bubble reduces in an accelerating
246 manner, with bubble wall velocity tending to infinity. In that case, the bubble collapse velocity is

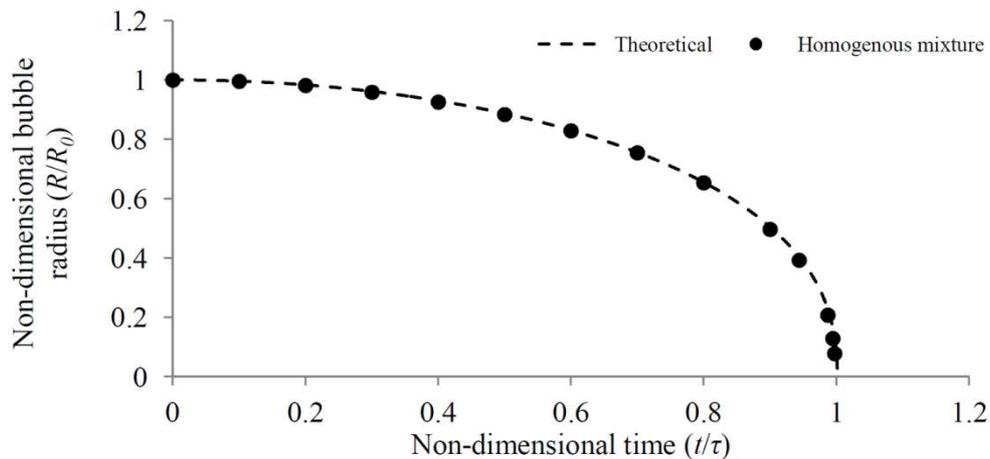
247 given by the following relation [22]:

$$248 \quad \frac{dR}{dt} = -\sqrt{\frac{2}{3} \frac{p_\infty - p_v}{\rho} \left[\left(\frac{R_0}{R} \right)^3 - 1 \right]} \quad (6)$$

249 which can be integrated numerically, till the characteristic Rayleigh time τ of bubble collapse:

$$250 \quad \tau \cong 0.915 R_0 \sqrt{\frac{\rho}{p_\infty - p_v}} \quad (7)$$

251 For the aforementioned conditions, the Rayleigh time is $\tau = 0.925 \mu\text{s}$. In Figure 6 comparison between
 252 the theoretical solution and the numerical solution with the two phase model is provided, showing an
 253 excellent agreement. This gives confidence that the results of the two phase model can be applied in
 254 arbitrary shaped cavitation structures, for which there is no theoretical solution; such structures
 255 however develop inside the injector and it is crucial that their collapse is captured properly.



256

257 Figure 6. Rayleigh collapse of a vapour bubble with the two phase model employed.

258 4.2. Throttle case

259 The throttle case examined is described in great detail in [16]; the throttle is formed on a metal
 260 plate sandwiched between two sapphire glasses for external observations. The cross-section of the
 261 throttle is $295 \times 300 \mu\text{m}$ and has a length of $993 \mu\text{m}$. A total pressure inlet is imposed 13 throttle widths
 262 upstream and a constant pressure outlet is imposed 30 throttle widths downstream, in order to
 263 minimize boundary influence as much as possible. The case examined has a pressure difference

264 300bar to 120bar from inlet to outlet and velocities up to ~250m/s develop inside the constriction.
265 From experimental observations, significant cavity shedding occurs, with cavitation reaching almost
266 till the middle of the channel length [16] and erosion is estimated to start from 120 μ m till 730 μ m from
267 the channel entrance, while being heavily pronounced in the area between 260-530 μ m from the
268 channel entrance [16].

269 Given the flow conditions inside the throttle, the Reynolds number is ~29000, which corresponds
270 to a Taylor length scale, λ_g [28]:

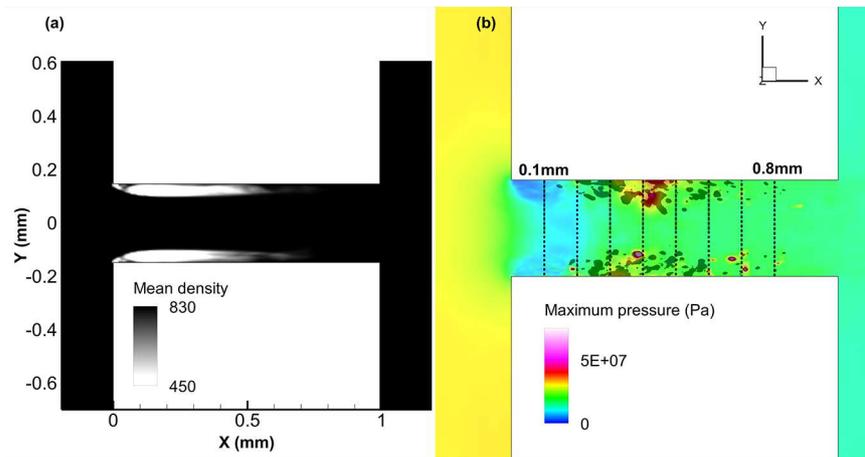
$$271 \quad \lambda_g = \sqrt{10} \text{Re}^{-0.5} L = 5.5 \mu\text{m} \quad (10)$$

272 where L is an indicative length scale of the geometry; here the throttle width has been used, i.e.
273 300 μ m. The Taylor length scale is useful for LES studies, since it can be used to estimate the
274 transition between inertial to viscous scales. The goal of the LES study is to simulate the anisotropic
275 scales larger than the Taylor length scale and to model the smaller viscous isotropic scales. Given this,
276 the resolution in the core of the throttle is 5 μ m, with refinement near the walls. The topology of the
277 mesh is block structured, with refinement at the throttle region. The time step used is 4ns, which
278 corresponds to a CFL of ~0.2, enabling to capture the highly transient fluid patterns. The simulation
279 was run for 50 μ s; assuming a Strouhal number of 0.3, commonly found in cavity shedding [22], the
280 period of one cavity oscillation is ~4 μ s, thus the total simulation time is more than 12 oscillation
281 periods which was considered enough for collecting statistics of the flow field.

282 In Figure 7 indicative results from the simulation are shown; the throttle is placed in such a way
283 that its plane of symmetry is positioned on the xy plane, i.e. the throttle is formed as an extruded
284 surface of the shown geometry in the normal direction. Both plots focus in the area of interest, at the
285 throttle. The flow moves from the negative to the positive direction of the x -axis.

286 As shown in Figure 7a, the averaged cavity length spans from the throttle entrance till a length of
287 0.5mm downstream the throttle, in accordance with the data reported in the work of Edelbauer et al.
288 [16]. In Figure 7b indicative locations of accumulated pressure peaks over the simulation time of

289 magnitude over 500bar are shown; these peaks are caused by the collapse of cavitation structures and
290 may reach values of even 1600bar locally.



291
292 Figure 7. Indicative results from the throttle simulation: (a) the averaged density distribution and (b) pressure peak location.

293 The black isosurface corresponds to peaks of magnitude higher than 500bar. The dashed lines are placed every 0.1mm.

294
295 As can be seen, pressure peaks are mainly located at the +y and -y walls of the throttle and not at
296 the -z and +z. Moreover, pressure peaks start to occur after 0.1mm and almost disappear after 0.7mm,
297 with the vast majority occurring between 0.2 and 0.6mm. Of course, the coverage of the walls with
298 pressure peaks is rather low, but this is reasonable given the simulation time. In any case, the locations
299 of pressure peaks is in a good agreement with the reported results.

300

301 4.3. Diesel injector - Case set-up

302 The Diesel injector tip geometries are shown in Figure 1. Since both injectors have five orifices,
303 only 1/5th of the domain was considered and periodic boundary conditions have been employed at the
304 sides of the domain. In fact, for a proper replication of the turbulence phenomena one might have to
305 simulate the full 360° of the Diesel injector, however this would impose a much higher computational
306 cost, considering also the mesh resolution that had to be used, thus a compromise had to be made. The
307 needle motion is assumed to be in the axial direction only, so any eccentricity effects were omitted.
308 Eccentricity effects might be important, especially during the early opening and late closing phases,

309 however such data are not currently available; besides including eccentricity would impose a full
310 injector tip simulation, which, as mentioned before, would be much more computationally expensive.

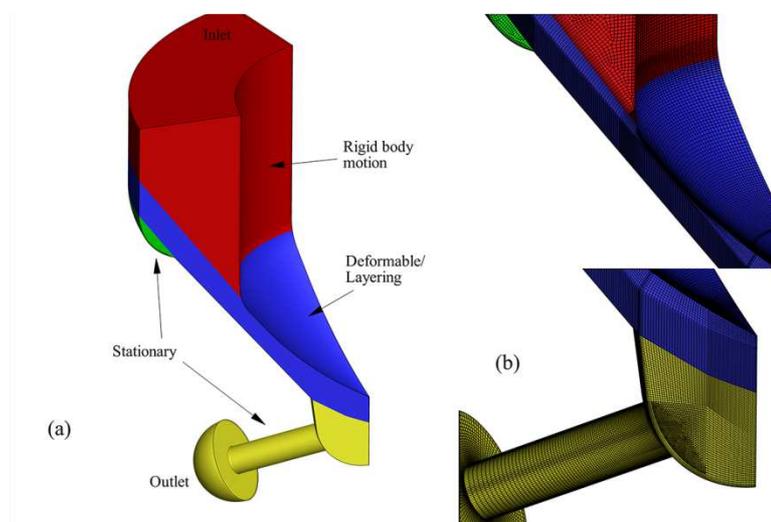
311 Pressure boundary conditions are set according to the upstream pressure profile (Figure 2) and
312 downstream pressure, while needle motion is set according to the lift profile. Note also that at the end
313 of the orifice of the injector an additional hemispherical volume was added (Figure 8a), in order to
314 move the influence of the outlet boundary further away from the orifice, especially considering that
315 cavitation structures may reach or even exit the orifice, as it will be shown later. The configuration
316 resembles the injection test benches (see section 2.1) where fuel is squirted into a collector filled with
317 liquid. The computational domain was split in a set of moving, deforming and stationary zones, as
318 shown in Figure 8a.

319 The computational mesh used is mainly hexahedral block-structured, with the exception of a zone
320 in the sac before the orifice entrance, which is unstructured tetrahedral. Mesh motion is performed
321 with a smoothing algorithm which stretches the cells in a uniform way at low lifts (from 5-40 μm),
322 while at higher lifts (40 μm till max. lift) a layering algorithm has been employed, adding/removing a
323 layer of cells as the needle moves every 7.5 μm . The mesh resolution used in critical areas where
324 cavitation develops, such as the sac volume and the orifice, is 7.5 μm with additional refinement near
325 walls. Given an average Reynolds number inside the injector orifice of ~ 30000 , an estimation of the
326 Taylor length scale, λ_g , is $\sim 7\mu\text{m}$, using the orifice diameter as a characteristic length scale, see Table I.

327 The needle lift was initially set at 5 μm with 10 cells in the gap between needle and needle seat.
328 Zero needle lift cannot be modelled with the methodology described so far, since this would require to
329 change the topology of the computational mesh. Alternatively, a 'closed valve' could have been
330 implemented with an artificial blockage at an interior boundary at the needle passage. In any case,
331 lower lifts have been avoided, in order to prevent as much as possible high aspect ratio cells and mesh
332 distortion, that could potentially have an impact on stability and accuracy of the results. An initial flow
333 field was obtained from a steady state run of pure liquid flow with a laminar flow assumption. Given
334 the fact that the Reynolds number at the minimum lift condition is ~ 1000 , calculated using the needle

335 lift as a length scale, not significant turbulence is expected to be generated at this stage. As will be
336 shown later, during the opening of the needle significant turbulence develops inside the sac volume
337 and orifice. The total cell count of the computational mesh is initially ~1million cells, but as the needle
338 moves, additional cell layers are introduced, so the mesh size increases to ~1.75 million cells.

339 A bounded central scheme (hybrid between central and second order upwind) was used for
340 momentum discretization, while second order upwind for density and QUICK for volume fraction.
341 Time advancement was performed with an implicit, second order, backward differentiation with a time
342 step of 5ns, in order to be able to capture the complicated turbulent patterns; the estimated CFL for
343 this time step and the minimum cell size is ~0.5, assuming a velocity of 500m/s. The implicit time
344 integration avoids time step restrictions due to compressibility effects, which would further limit the
345 time step to even lower values.



346
347 Figure 8. (a) Splitting of the geometry to accommodate mesh motion (b) details of the mesh at the needle seat passage and
348 sac volume.

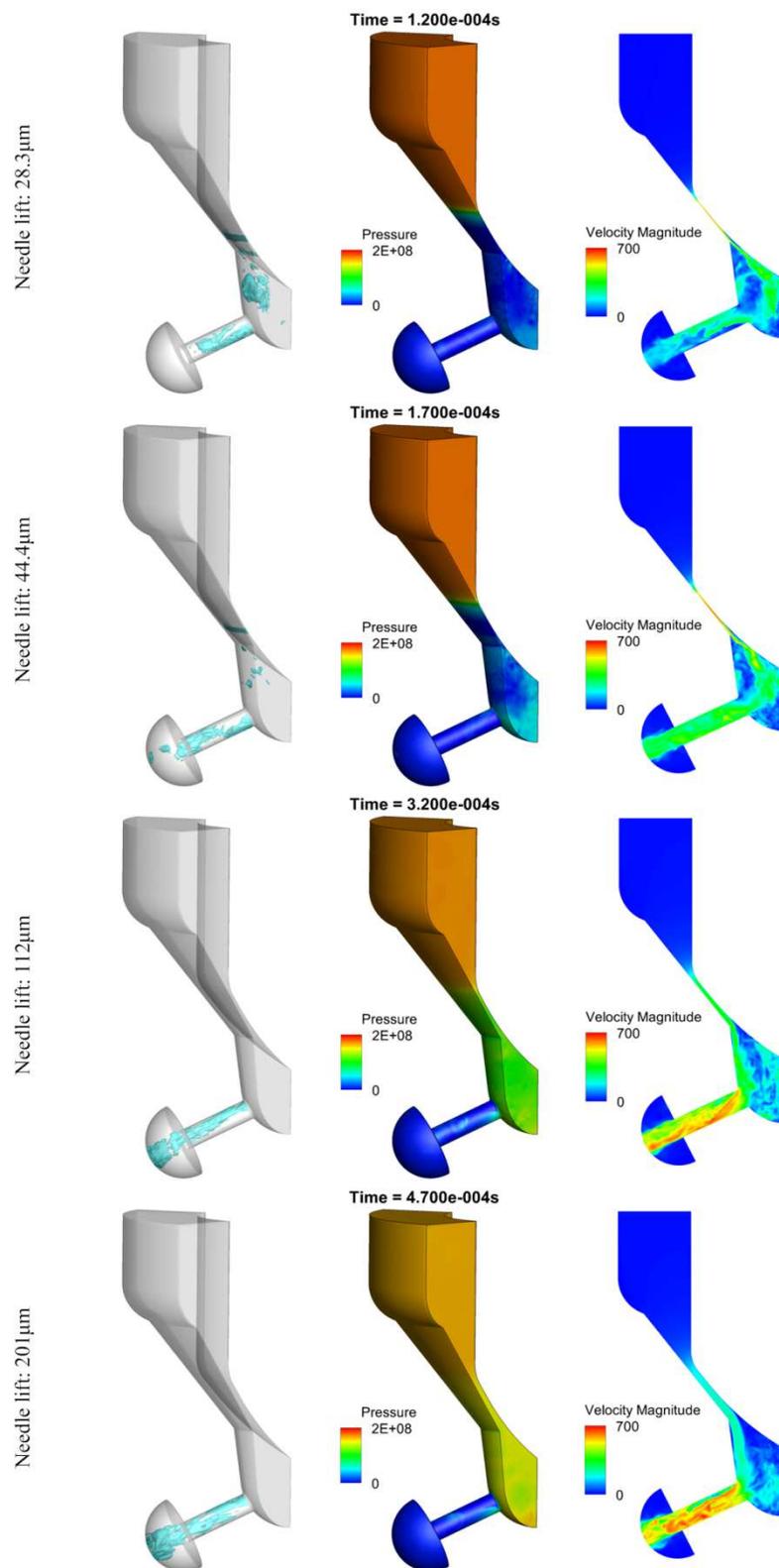
349 4.4. Diesel injector - simulation results

350 In both injectors, cavitation is predicted to occur initially at the gap between the needle and the
351 needle seat. For design A, indicative flow field results are shown in Figure 9. At the very early
352 opening stages of Design A injector a large part of the sac volume is filled with vapor/liquid mixture;
353 this seems to be related to the large sac volume of the injector in combination with the needle motion

354 profile imposed. This vaporous structure quickly collapses, causing a pressure peak at the sac wall on
355 the axis of symmetry see also Figure 12, as flow moves in from upstream the injector and the orifice
356 exit. Cavitation in the passage between the needle and the needle seat remains till $180\mu\text{s}$ from the
357 beginning of the simulation, that corresponds to a needle lift of $\sim 48\mu\text{m}$. Cavitation inside the sac
358 volume is caused by strong turbulence and vortices; indeed, as visible at $120\mu\text{s}$, even at a lift of $28\mu\text{m}$
359 the shear layer instabilities between the liquid jet from the needle/needle seat passage and the liquid
360 cause a very complicated flow field inside the sac volume. Note also that the liquid jet formed at the
361 needle/needle seat passage is attached at the needle surface. Cavitation in the sac volume persists till
362 $220\mu\text{s}$ or a lift of $65\mu\text{m}$; beyond this point the minimum pressure in the sac volume has risen to a level
363 of 40bar, preventing formation of cavitation. At $110\mu\text{s}$ cavitation forms at the entrance of the orifice,
364 close to the lower orifice surface. From that point onwards, cavitation structures may span in the
365 whole orifice length and may even exit the orifice, see also Figure 10 showing the instances of flow
366 regions with pressure below saturation. Later on, from $280\mu\text{s}$ till $320\mu\text{s}$ there is a transition in the
367 cavitation formation from the lower orifice surface to the upper orifice surface; as shown in Figure 9,
368 at $320\mu\text{s}$, corresponding to a lift of $112\mu\text{m}$, cavitation spans on the upper orifice surface mainly. This
369 effect coincides with the attachment of the liquid stream, moving in from upstream the injector tip, to
370 the sac walls instead of the needle (see also Figure 9, at $320\mu\text{s}$). From that point till the maximum lift,
371 cavitation forms at the upper orifice surface, with occasional cavitating vortices located at the centre of
372 the orifice.

373 In Figure 11 indicators of the significant turbulence in the orifice and sac volume are shown, which
374 justify the existence of cavitating vortices. Figure 11a shows the tangential velocity distribution at four
375 locations inside the orifice; as shown, tangential velocities may exceed 160m/s locally and may even
376 peak at 300m/s near the orifice entrance. Figure 11b shows the coherent vortical structures that form in
377 the sac volume, orifice and even extend beyond the injector; note that vortical strings may form inside
378 the sac volume and extend in the orifice as well. The second invariant of the velocity gradient tensor
379 has been used to indicate vortical structures [29], since positive values correspond to coherent vortices

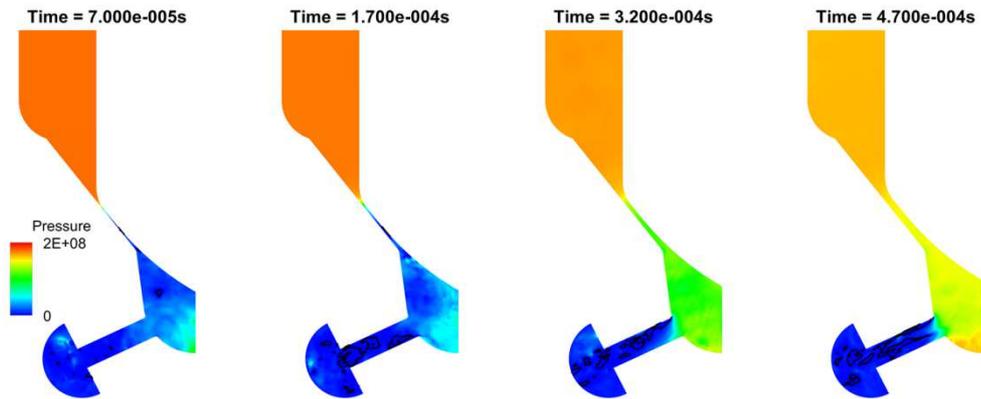
380 (also known as Q-criterion) [30, 31].



381
382

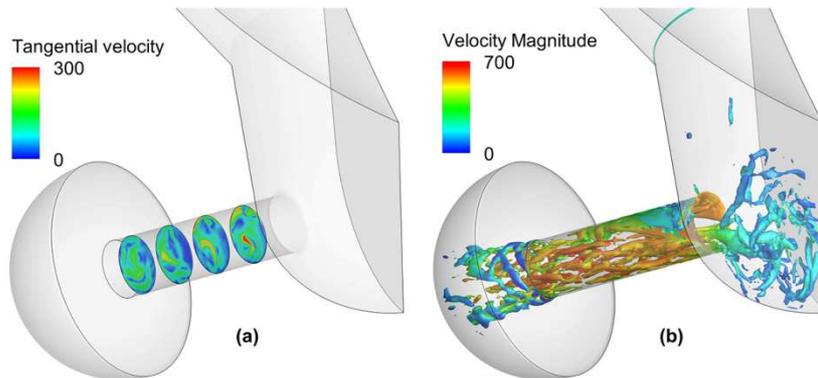
Figure 9. Indicative instances during the needle opening phase of Design A. From left to right, vapour isosurface at 50%, instantaneous pressure field and instantaneous velocity magnitude at the mid-plane of the injector.

383



384
385
386

Figure 10. Instantaneous pressure field at the mid-plane of the Design A injector. The thick black line shows regions where local pressure is less or equal to saturation pressure.



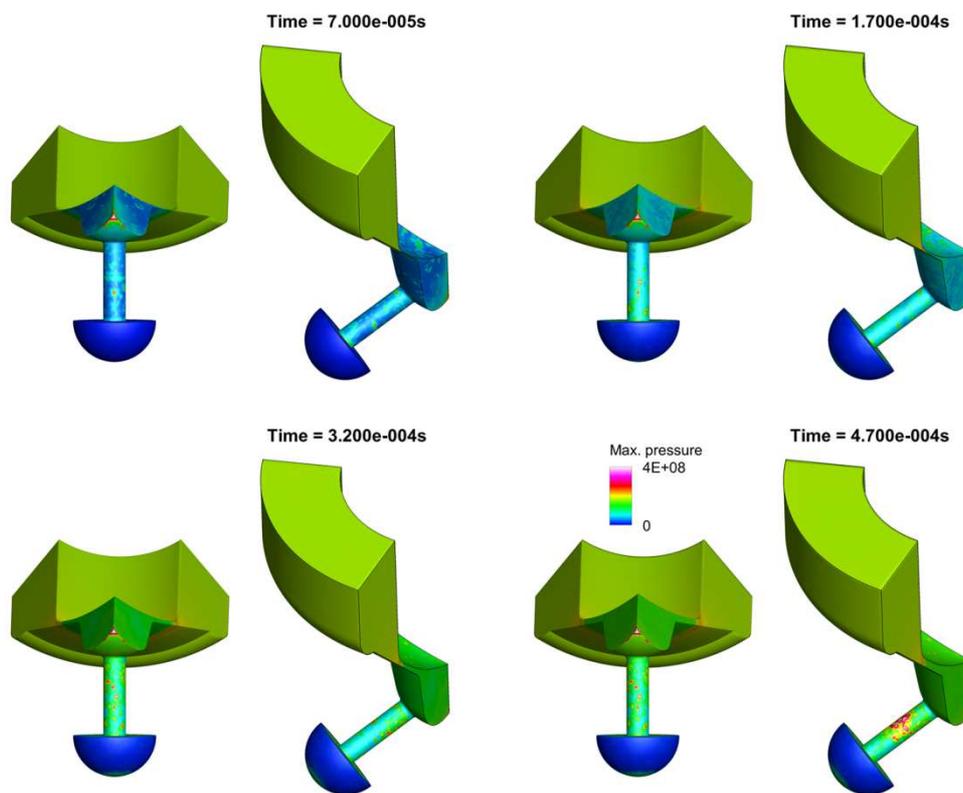
387
388
389
390

Figure 11. Design A at 560 μ s and 250 μ m lift (a) Instantaneous tangential velocity distribution on slices normal to the orifice. (b) Instantaneous isosurface of the second invariant of the velocity gradient tensor, showing vortex cores (value $5 \cdot 10^{12} \text{ s}^{-2}$) and coloured according to the local velocity magnitude.

391
392
393
394
395
396
397
398

In Figure 12 the temporal evolution of the maximum accumulated pressure peaks (that is local pressure maximum) on various injector surfaces of Design A is shown; note that red colour corresponds to peak pressures of 3000bar, purple to 3500bar and white to 4000bar. As a comparison it is mentioned that the yield stress of Stainless Steel 316 is 200-400 MPa, see [32, 33] ; thus locations of pressure peaks beyond 3000bar could indicate sites of plastic deformation/work hardening which is a prior stage of material removal. At 70 μ s, there is a pressure peak at the sac wall intersection with the axis of symmetry; this was observed to be caused from the initial vapour formation in the sac volume.

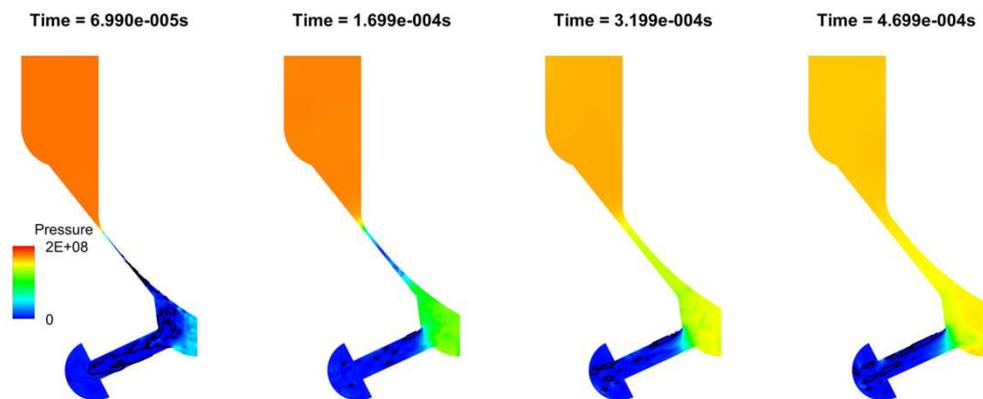
399 During the cavitation formation at the lower orifice surface ($110\text{-}320\mu\text{s}$), several pressure peaks with
400 magnitude higher or equal to 3000bar are accumulated at the lower surface (with some peaking at
401 4500bar), due to vapour structure collapse; these peaks are formed from $\sim 20\%$ of the orifice length,
402 downstream the entrance, till the exit of the orifice. Later on, as cavitation moves near the upper
403 orifice surface, some scattered pressure peaks occur at the sides of the orifice. Eventually, as cavitation
404 established at the upper orifice surface, vapour structure collapses form a cluster of pressure peaks
405 there, almost at 45% of the orifice length, downstream the entrance. Note that the needle is free of
406 significant pressure peaks, as well as the sac volume surface.



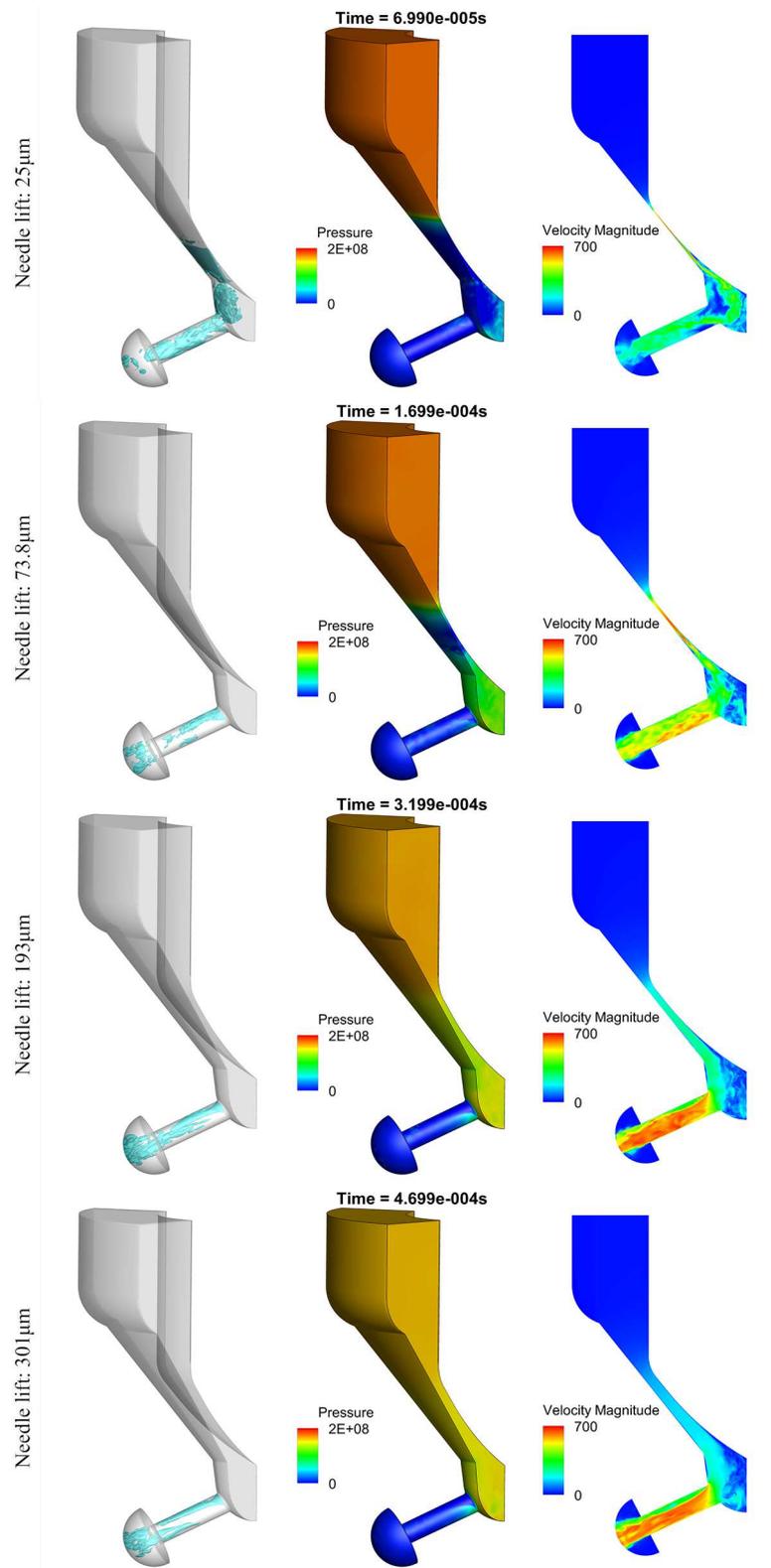
407
408 Figure 12. Time evolution of the maximum pressure on various locations of Design A injector walls.

409
410 Cavitation occurrence in Design B shows some similarities to the Design A, however there are
411 some fundamental differences, see also Figure 14. First of all, a significant difference is that there is
412 no vapor filling of Design B sac volume at the early opening stages. Cavitation between the needle and

413 needle seat starts from the beginning of the injection opening till $170\mu\text{s}$ or needle lift of $74\mu\text{m}$;
414 comparing with Design A cavitation persists in this location at a higher lift but shorter duration (for
415 Design A $180\mu\text{s}$ and $47\mu\text{m}$ lift). As in Design A, the jet formed at the passage, initially attaches on the
416 needle surface, forming a large cavitating vortex inside the sac; sac cavitation first appears at $20\mu\text{s}$ or
417 needle lift of $9\mu\text{m}$ and remains till $160\mu\text{s}$ or $67\mu\text{m}$, which is a similar lift as Design A. The vortex
418 formed in the sac forces the flow to enter from the lower orifice surface, beginning from $30\mu\text{s}$ and
419 $12\mu\text{m}$ lift till $140\mu\text{s}$ and $56\mu\text{m}$ lift; cavitation at the lower orifice surface forms much earlier in Design
420 B injector than Design A. Later, at $\sim 160\mu\text{s}$ and $67\mu\text{m}$ lift (see Figure 14), a transition occurs that the
421 flow attaches on the sac wall instead; from that point onwards cavitation develops at the upper orifice
422 surface. Again, sporadic occurrence of vortex cavitation near the centre of the orifice is found, but in
423 less extent than Design A; this is justified by the hole tapering and the developed turbulence inside the
424 orifice, as will be shown later. As in Design A, cavitation structures may temporarily reach the orifice
425 exit and even extend outside of the injector, see also Figure 13.
426



427
428 Figure 13. Instantaneous pressure field at the mid-plane of the Design B injector. The thick black line shows regions where
429 local pressure is below or equal to saturation pressure.

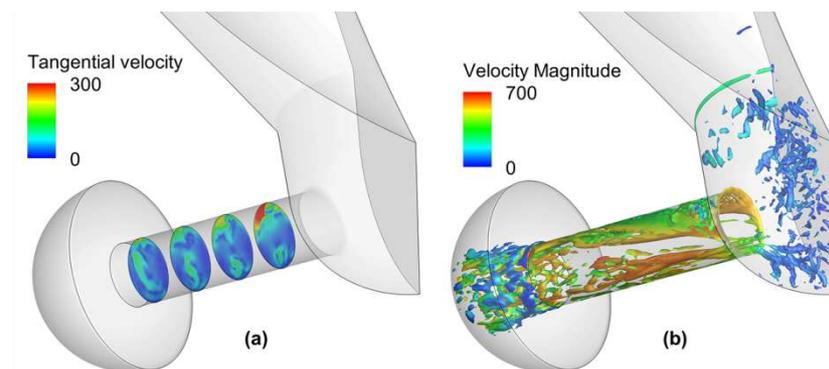


430

431 Figure 14. Indicative instances during the needle opening phase of Design B. From left to right, vapour isosurface at 50%,

432 instantaneous pressure field and instantaneous velocity magnitude at the mid-plane of the injector.

433 In Figure 15a the tangential velocity distribution at four locations inside the orifice is shown, at the
434 same lift as Design A in Figure 11a; while the maximum tangential velocity in both cases is $\sim 300\text{m/s}$
435 and is located near the orifice entrance, the average tangential velocity in Design B is lower than the
436 one in Design A by almost 25-45%, depending on the location; less near the orifice entrance, more
437 near the orifice exit. In Figure 15 the coherent vortical structures are shown as an isosurface, for the
438 same value as Design A. One observation is that vortical structures are not that developed/extended
439 inside the orifice; this agrees with the fact that there are lower tangential velocities in the orifice slices
440 in Figure 15a. On the other hand, there are more scattered structures throughout the whole sac volume
441 in Design B.



442 Figure 15. Design B at $400\mu\text{s}$ and $250\mu\text{m}$ lift (a) Instantaneous tangential velocity distribution on slices normal to the orifice.
443 (b) Instantaneous isosurface of the second invariant of the velocity gradient tensor, showing vortex cores (value $5 \cdot 10^{12} \text{ s}^{-2}$) and
444 coloured according to the local velocity magnitude.
445

446
447 In Figure 16 the temporal evolution of the maximum accumulated pressure peaks on various
448 injector surfaces of Design B are shown. Here it is visible that very early, at $70\mu\text{s}$, the intense
449 cavitation in the sac volume causes significant pressure peaks at the needle surface; actually wall
450 pressure peaks may even reach instantaneous values of over 5000bar (local pressure at spots of the
451 bulk liquid volume may locally reach 9000bar). Later on, after $320\mu\text{s}$, pressure peaks start to form at
452 the upper orifice surface, due to cavity shedding developing near this region. Also, some spots of
453 pressure peaks appear on the sac volume, whereas the lower orifice surface is totally clean of pressure
454 peaks.

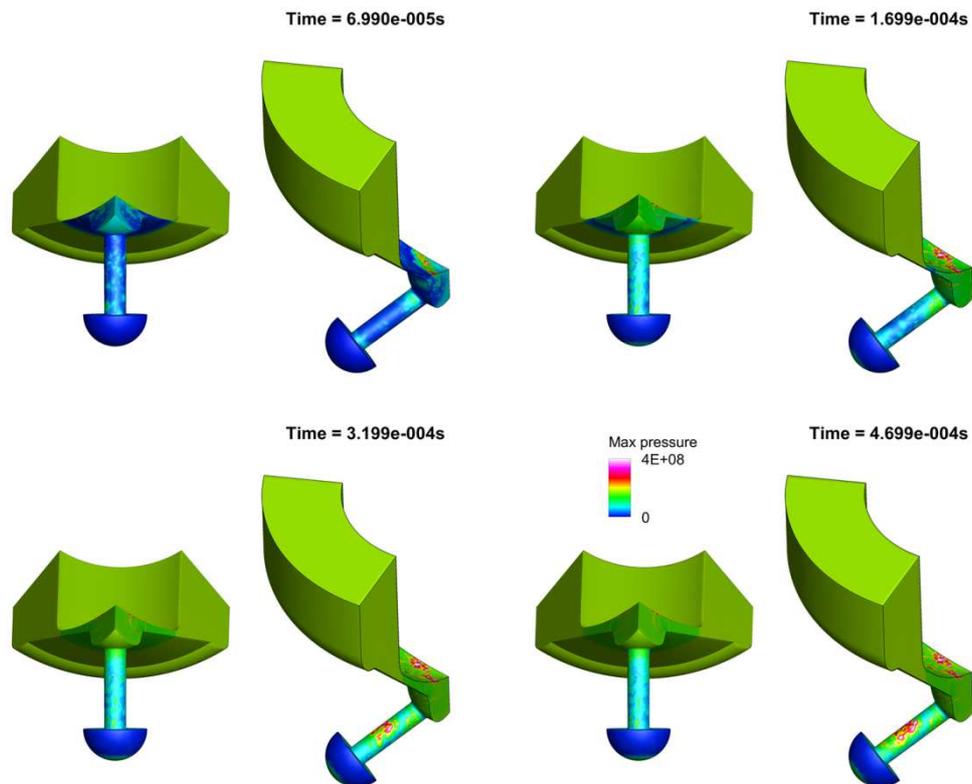


Figure 16. Time evolution of the maximum pressure on various locations of Design B injector walls.

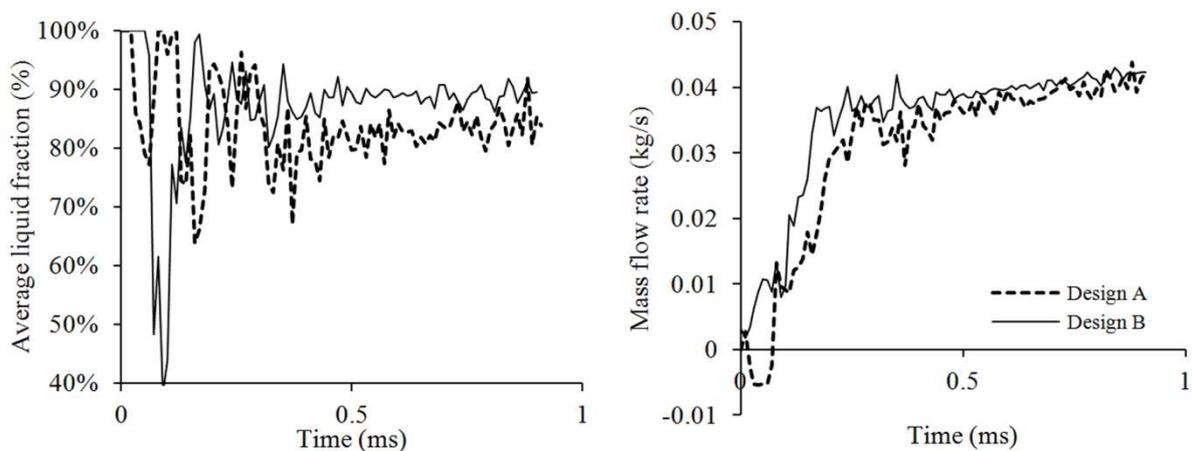
455
456

457 5. Discussion

458 Cavitation presence in the sac volume of the Design B was found to be higher than that of Design
 459 A injector, without considering the initial vapour filling of Design A (which is probably due to the
 460 imposed needle motion at the first time steps). Whereas there is no significant difference in the
 461 velocity field development in the two injectors, i.e. the flow initially attaches on the needle and then
 462 on the sac, the fundamental difference is that in Design B the needle moves faster than in Design A, by
 463 ~50%. At low lifts, this reduces the pressure in Design B sac causing more cavitation there, due to the
 464 imposed flow acceleration from the fast needle displacement.

465 On the other hand, cavitation presence in the form of cavitating vortices is more extensively found
 466 in the orifice of Design A injector; the same applies for flow turbulence. This seems to be related to
 467 the hole tapering; indeed the cylindrical hole of Design A injector promotes cavitation formation. On
 468 the other hand, the conical orifice in Design B injector reduces the amount of cavitation vortices inside
 469 the hole, leaving almost only a vaporous layer at the upper orifice surface. The flow is also more

470 ordered and with less tangential velocity component in the orifice sections of Design B injector.
 471 Another way to illustrate these effects is by examining the mass flow rate and the average vapour
 472 fraction at the orifice exit, as shown in Figure 17: the liquid fraction and the mass flow rate is higher in
 473 Design B injector at high lifts operation. Note also that at the early opening stages of Design A
 474 injector a slight flow reversal is found at the injector outlet; as before, this is related to the imposed
 475 needle motion and the significantly large sac volume of Design A.



476
 477 Figure 17. Average liquid volume fraction and mass flow rate through the orifice exit for the examined injectors - opening
 478 phase. The jagged lines are due to the low sampling rate.

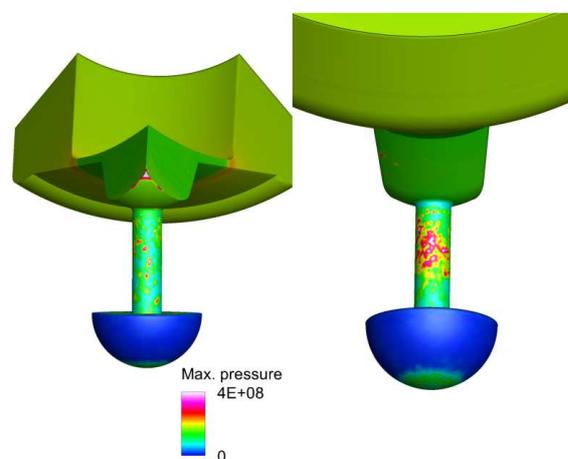
479 An important observation for the flow in both injectors is that, even though the flow is well ordered
 480 upstream the injector and in the passage between the needle and the needle seat, there is significant
 481 turbulence generation inside the sac volume, due to the sudden expansion, and the orifice, due to the
 482 strong flow direction change. Indeed, the maximum Reynolds number at the annulus upstream the tip
 483 is ~ 10000 , occurring at the maximum lift; this means that the flow upstream the injector tip will be
 484 transitional at maximum lift and laminar at lower lifts. Information on possible turbulent fluctuations
 485 upstream the injector tip have not been prescribed, since currently such data are not available. Still, the
 486 presence of significant turbulence downstream the needle/needle seat passage can be explained by the
 487 strong shear instabilities of the fuel stream rushing in the sac volume.

488 Regarding erosion prediction for Design A injector, pressure peaks significantly exceeding 3000bar
 489 are found at scattered spots at the lower orifice surface, spanning from 20% of the orifice length till

490 the exit of the orifice, and a densely populated pressure peak region at the upper orifice surface, see
491 Figure 18. Both of these facts could potentially correlate to the erosion patterns of Design A in some
492 cases, e.g. see Figure 4. Such pressure values are comparable to the yield stress of metal alloys, thus
493 the existence of such collapses can detrimentally contribute to local fatigue. Material exposed at such
494 pressures, over time may undergo plastic deformation and material removal, changing the local flow
495 field and potentially enhancing cavitation damage downstream. Simulations indicate that the needle of
496 Design A injector is practically clear of high pressure peaks, which also correlates well with the barely
497 observable erosion from the experiments.

498 A good trend is found for Design B as well; from the experiments a clear pattern is identified with
499 erosion formation on the needle surface in the form of a deeply engraved ring shape, at the upper
500 orifice surface and at some spots on the sac wall upstream the orifice. As shown in Figure 19, these
501 locations are predicted very well from the simulations:

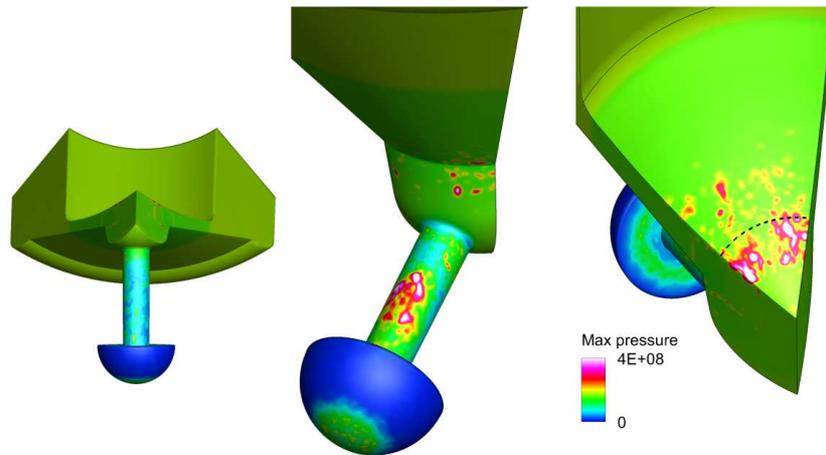
- 502 - High pressure peaks are found in a circular pattern on the needle of Design B injector. Local
503 pressures may exceed 5000bar.
- 504 - Pressure peaks of more than 4000bar are found at the upper orifice hole in a clustered
505 arrangement. The lower orifice surface is clean of high pressure peaks.
- 506 - Sporadic pressure peaks of pressures higher than 3500bar are found at the sac wall.



507

508

Figure 18. Accumulated pressure peak distribution at various locations of Design A.



509

510

511

Figure 19. Accumulated pressure peak distribution at various locations of the Design B. The dashed line denotes a radius of 0.6mm.

512

513

514

515

516

Unfortunately, the simulation is very demanding from a computational point of view, requiring significant time to compute, mainly due to the very small time step required. These simulations have been running each on one 12CPU Xeon E5-2630 v2 @ 2.6GHz computer for 3 months to get to this point; potentially there could be a benefit by running in a distributed parallel environment with much more processors.

517

6. Conclusion

518

519

520

521

522

523

524

525

526

527

This paper outlines the potential of 2-phase cavitation models in the prediction of erosion effects, by tracking the Rayleigh collapse of vapor structures. The methodology is tested in a benchmark case of the collapse of a spherical vapor bubble. Then, it is applied in a more complicated case of a throttle resembling the injector passages and the opening phase of a Diesel injector. LES turbulence models have been used, since in the cavitation literature there are enough indications that RANS/URANS models may be situational. Erosion in complicated geometries is correlated to pressure peaks that form during the collapse of vapor structures. In the injectors examined, these peaks may reach pressures of more than 4000bar, depending on the location. It is highlighted that such pressures are higher than the yield stress of common materials, e.g. SS316, and can contribute to the plastic deformation of material which is the first stage in the work hardening process before material removal. Indicative CT scans are

528 provided for the two examined injectors after endurance testing. CFD results of Design A show some
529 resemblance to the experimentally observed erosion patterns; the needle is free of erosion, whereas
530 pressure peaks are found inside the orifice, at both upper and lower surfaces. Design B shows a much
531 greater consistency in the erosion development. Moreover there is very good agreement of the
532 predicted pressure peak locations with the observed erosion patterns: high pressure peaks are found on
533 the needle surface, at the upper orifice surface and at sporadic locations of the sac wall, all being in
534 accordance with the experiment. The present work's novelty is to use such a methodology in a diesel
535 injector with a moving needle and correlating the pressure peaks due to vapor collapse with erosion
536 damage, determined from experiments. Continuation of this work will involve examination of further
537 injection stages, as well as possible inclusion of eccentricity effects or upstream turbulence
538 fluctuations, should these information be available.

539

540 **Disclaimer**

541 CAT, CATERPILLAR, their respective logos, “Caterpillar Yellow,” the “Power Edge” trade dress as
542 well as corporate and product identity used herein, are trademarks of Caterpillar and may not be used
543 without permission. ©2015 Caterpillar All Rights Reserved.

544

545 **Nomenclature**

D_{in}	Orifice entrance diameter (m)
D_{out}	Orifice exit diameter (m)
p	Pressure (Pa)
B	Bulk modulus (Pa)
ρ	Density (kg/m^3)
$\rho_{sat,L}$	Density at saturation (kg/m^3)
n	Tait equation exponent (for liquid) (-)
p_{sat}, p_v	Saturation/Vapour pressure (Pa)

μ_L	Dynamic viscosity of the liquid (Pa.s)
a	Vapour fraction (-)
ρ_v	Vapour density
\mathbf{u}	Velocity field
R_e	Evaporation rate (kg/m ³ /s)
R_c	Condensation rate (kg/m ³ /s)
μ_v	Vapour dynamic viscosity (Pa.s)
R	Bubble radius (m), index 0 denotes initial radius
p_∞	Pressure at far field (Pa)
τ	Rayleigh time (s)
λ_g	Taylor length scale (m)

546

547 **Acknowledgements**

548 The research leading to these results has received funding from the People Programme (IAPP Marie
549 Curie Actions) of the European Union's Seventh Framework Programme FP7/2007-2013/ under REA
550 grant agreement n. 324313.

551

552

553 **References**

554

555 [1] Sezal I. *Compressible Dynamics of Cavitating 3-D Multi-Phase Flows*, in *Fakultat fur*
556 *Maschinenwesen*2009, Technischen Universitaet Muenchen.

557 [2] Sezal IH, Schmidt SJ, Schnerr GH, Thalhamer M, and Förster M. Shock and wave dynamics
558 in cavitating compressible liquid flows in injection nozzles. *Shock Waves* 2009; **19**(1): p. 49-
559 58 DOI: 10.1007/s00193-008-0185-3.

- 560 [3] Salvador FJ, Romero JV, Roselló MD, and Martínez-López J. Validation of a code for
561 modeling cavitation phenomena in Diesel injector nozzles. *Mathematical and Computer*
562 *Modelling* 2010; **52**(7–8): p. 1123-1132 DOI: <http://dx.doi.org/10.1016/j.mcm.2010.02.027>.
- 563 [4] Salvador FJ, Carreres M, Jaramillo D, and Martínez-López J. Analysis of the combined effect
564 of hydrogrinding process and inclination angle on hydraulic performance of diesel injection
565 nozzles. *Energy Conversion and Management* 2015; **105**: p. 1352-1365 DOI:
566 <http://dx.doi.org/10.1016/j.enconman.2015.08.035>.
- 567 [5] Salvador FJ, Martínez-López J, Caballer M, and De Alfonso C. Study of the influence of the
568 needle lift on the internal flow and cavitation phenomenon in diesel injector nozzles by CFD
569 using RANS methods. *Energy Conversion and Management* 2013; **66**: p. 246-256 DOI:
570 <http://dx.doi.org/10.1016/j.enconman.2012.10.011>.
- 571 [6] Molina S, Salvador FJ, Carreres M, and Jaramillo D. A computational investigation on the
572 influence of the use of elliptical orifices on the inner nozzle flow and cavitation development
573 in diesel injector nozzles. *Energy Conversion and Management* 2014; **79**: p. 114-127 DOI:
574 <http://dx.doi.org/10.1016/j.enconman.2013.12.015>.
- 575 [7] Salvador FJ, Martínez-López J, Romero JV, and Roselló MD. Computational study of the
576 cavitation phenomenon and its interaction with the turbulence developed in diesel injector
577 nozzles by Large Eddy Simulation (LES). *Mathematical and Computer Modelling* 2013;
578 **57**(7–8): p. 1656-1662 DOI: <http://dx.doi.org/10.1016/j.mcm.2011.10.050>.
- 579 [8] Örley F, Hickel S, Schmidt SJ, and Adams NA. LES of cavitating flow inside a Diesel injector
580 including dynamic needle movement. *Journal of Physics: Conference Series* 2015;
581 **656**(012097) DOI: 10.1088/1742-6596/656/1/012097.
- 582 [9] Patouna S. *A CFD study of cavitation in real size Diesel injectors* in *Departamento de*
583 *Maquinas y Motores Termicos* 2012, Universitat Politecnica de Valencia: Valencia.

- 584 [10] Strotos G, Koukouvinis P, Theodorakakos A, Gavaises M, and Bergeles G. Transient heating
585 effects in high pressure Diesel injector nozzles. *International Journal of Heat and Fluid Flow*
586 2015; **51**(0): p. 257-267 DOI: <http://dx.doi.org/10.1016/j.ijheatfluidflow.2014.10.010>.
- 587 [11] Devassy M, Caika V, Sampl P, Edelbauer W, and Greif D. Numerical investigation of
588 cavitation injector flow accounting for 3D-needle movement and liquid compressibility effects.
589 in *Fuel Systems for IC Engines*, IMechE, London 2014.
- 590 [12] Gavaises M, Papoulias D, Andriotis A, Giannadakis E, and Theodorakakos A. Link Between
591 Cavitation Development and Erosion Damage in Diesel Injector Nozzles. *SAE Technical*
592 *Paper 2007-01-0246* 2007 DOI: 10.4271/2007-01-0246.
- 593 [13] Koukouvinis P, Bergeles G, Li J, Wang L, Theodorakakos A, and Gavaises M. Simulation of
594 cavitation inside diesel injectors, including erosion modelling. in *Fuel Systems for IC Engines*,
595 IMechE, London 2014.
- 596 [14] Kolev N. *Multiphase Flow Dynamics 3*. 2007: Springer Berlin Heidelberg.
- 597 [15] Coutier-Delgosha O, Reboud JL, and Delannoy Y. Numerical simulation of the unsteady
598 behaviour of cavitating flows. *International Journal for Numerical Methods in Fluids* 2003;
599 **42**: p. 527-548 DOI: 10.1002/flid.530.
- 600 [16] Edelbauer W, Struel J, and Morozov A. Large Eddy Simulation of cavitating throttle flow. in
601 *SimHydro 2014:Modelling of rapid transitory flows*, Sophia Antipolis 2014.
- 602 [17] Egler W, Giersch JR, Boecking F, Hammer J, Hlousek J, Mattes P, Projahn U, Urner W, and
603 Janetzky B, *Fuel Injection Systems*, in *Handbook of Diesel Engines*, Mollenhauer K and
604 Tschöke H, Editors. 2010, Springer-Verlag Berlin Heidelberg. p. 127-174.
- 605 [18] ANSYS. *ANSYS Fluent 15.07*, 2013.
- 606 [19] Lemmon EW, Huber ML, and McLinden MO. *NIST Standard Reference Database 23:*
607 *Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1*, 2013,
608 National Institute of Standards and Technology, Standard Reference Data Program:
609 Gaithersburg.

- 610 [20] Ivings MJ, Causon DM, and Toro EF. On Riemann solvers for compressible liquids.
611 *International Numerical Methods for Fluids* 1998; **28**: p. 395-418 DOI: 10.1002/(SICI)1097-
612 0363(19980915)28:3<395::AID-FLD718>3.0.CO;2-S.
- 613 [21] Ji B, Luo X, Wu Y, Peng X, and Duan Y. Numerical analysis of unsteady cavitating turbulent
614 flow and shedding horse-shoe vortex structure around a twisted hydrofoil. *International*
615 *Journal of Multiphase Flow* 2013; **51**(0): p. 33-43 DOI:
616 <http://dx.doi.org/10.1016/j.ijmultiphaseflow.2012.11.008>.
- 617 [22] Franc J-P and Michel J-M. *Fundamentals of Cavitation*. 2005: Kluwer Academic Publishers.
- 618 [23] Brennen C. *Cavitation and Bubble Dynamics*. 1995: Oxford University Press.
- 619 [24] Trevena DH. Cavitation and the generation of tension in liquids: Review article. *Journal of*
620 *Physics D: Applied Physics* 1984; **17**: p. 2139-2164 DOI: 10.1088/0022-3727/17/11/003.
- 621 [25] Heyes DM. Liquids at positive and negative pressure. *Physica Status Solidi (B)* 2008; **245**(3):
622 p. 530-538 DOI: 10.1002/pssb.200777706.
- 623 [26] Kobayashi H and Wu X. *Application of a local subgrid model based on coherent structures to*
624 *complex geometries*, 2006, Center for Turbulence Research. p. 69-77.
- 625 [27] Nicoud F and Ducros F. Subgrid-scale stress modelling based on the square of the velocity
626 gradient tensor. *Flow, Turbulence and Combustion* 1999; **62**: p. 183-200.
- 627 [28] Pope S. *Turbulent Flows*. 2000: Cambridge University Press.
- 628 [29] X. Jiang and Lai CH. *Numerical techniques for direct and large eddy simulations*. Numerical
629 analysis and scientific computing. 2009: Chapman & Hall / CRC. 276
- 630 [30] Green MA, Rowley CW, and Haller G. Detection of Lagrangian Coherent Structures in 3D
631 Turbulence. *Journal of Fluid Mechanics* 2007; **572**: p. 111-120.
- 632 [31] Wu JZ, Ma HY, and Zhou MD. *Vorticity and Vortex Dynamics* 1ed. 2006: Springer-Verlag
633 Berlin Heidelberg.
- 634 [32] Young SG. *Cavitation damage of stainless steel, nickel and an aluminum alloy in water for*
635 *ASTM round robin tests*, 1968, National Aeronautics and Space Association: Washington D.C.

636 [33] Berchiche N, Franc JP, and Michel JM. A Cavitation Erosion Model for Ductile Materials.

637 *ASME Journal of Fluids Engineering* 2002; **124**: p. 201-207 DOI: 10.1115/1.1486474.

638

639