Porous Fuel Air Mixing Enhancing Nozzle (PFAMEN)

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ABSTRACT

One of the challenges with conventional diesel engines is the emission of soot. To reduce soot emission whilst maintaining fuel efficiency, an important pathway is to improve the fuel-air mixing process. This can be achieved by creating small droplets in order to enhance evaporation. Furthermore, the distribution of the droplets in the combustion chamber should be optimized, making optimal use of in-cylinder air. To deal with these requirements a new type of injector is proposed, which has a porous nozzle tip with pore diameters between 1 and 50 µm. First, because of the small pore diameters the droplets will also be small. From literature it is known that (almost) no soot is formed when orifice diameters are smaller than 50 µm. Second, the configuration of the nozzle can be chosen such that the whole cylinder can be filled with fine droplets (i.e., spray angle nearly 180°). However, injecting through a porous nozzle is not the same as an infinite number of very small holes, due to the difference in nozzle internal flow. Therefore, the nozzle tip is modeled in COMSOL Multiphysics in order to predict the outflow direction and velocity of the fuel. The Darcy-Forchheimer equation, which follows from the Navier-Stokes equation, is used for this purpose. To validate the model, experiments have been performed in the Eindhoven High Pressure Cell (EHPC) where (for vaporizing sprays) the spray is visually analyzed and (for reacting sprays) the ignition delay has been measured.

INTRODUCTION

Increasingly stringent emission legislation has lead to an intensified research on HD diesel engines to keep and improve their fuel consumption, whilst still meeting the emission legislation. Many developments such as aftertreatment, EGR and increasingly higher fuel injection pressures were introduced to the market. Modern Heavy Duty (HD) diesel engines always have Direct Injection (DI). A high pressure pump delivers fuel at 100-250 MPa to an injector with 6-8 holes with a diameter typically around 150-200 µm. After the start of injection, the fuel forms a spray where the liquid fuel breaks up into smaller droplets. In the meantime the diesel is heated up and evaporates via entrainment of high temperature gas.

As a result of the above developments, engines have already become much cleaner. However, in order to meet EPA 10/EURO VI-like emission standards, new techniques might be required or become more interesting as costs tend to further increase. Looking at results from literature [1] and [2] there is a trend that the smaller the diameter of the injection holes gets, the less soot is formed throughout the combustion process. If the diameter of the holes becomes smaller, the total outflow area decreases, resulting in a lower volume flow. By applying more holes this issue could be overcome. However, the maximum number of holes and the minimum diameter of the holes is limited. For these reasons new solutions have to be found. One possible solution would be to inject the fuel through a porous material. The porous material contains many small pores over a large surface of the injector tip, and in this respect principally differs from injection through a conventional nozzle with straight-drilled injector holes.

To assess the technical viability of such a nozzle, a numerical model was built (using COMSOL Multiphysics)
and the flow through the material and its strength were investigated. The flow through the porous material is described by the coupled Darcy-Forchheimer equation [3]. The Darcy and Forchheimer equations are derived from the Navier-Stokes equation by Neuman [4] and Chen et al. [5], respectively.

Besides the numerical investigation, some experimental measurements with the porous injector were performed. To test the porous injector concept, prototype nozzles were produced and the original injector tip was replaced by a porous tip. The injector was connected to two different common-rail setups, one at atmospheric conditions and one at engine-like conditions in a constant volume spray bomb. The atmospheric tests are quickly and relatively easily performed and the engine-like measurements are closer to real engine circumstances. With the atmospheric measurements the spray is analyzed visually, the volume flow of the injector is evaluated and durability tests are performed. With the engine-like tests, which are performed in the Eindhoven High Pressure Cell (EHPC, for details see [6]), again the spray is analyzed, ignition delay is determined and the combustion process is qualitatively analyzed.

In Figure 1 a typical fuel distribution is illustrated for conventional injectors and an ideal fuel distribution for porous injectors, respectively. It is expected that the quantity of oxygen that potentially takes part in the combustion process, is much larger with the porous injector since the occupied volume is larger. However, whether this is really the case will also depend on time scales (a.o. governed by exit velocity, which will be smaller due to the internal geometry of and resulting flow friction in the porous nozzle). This is one of the subjects investigated in this work.

![Figure 1. Fuel distribution of a conventional (a) and porous (b) injector.](image)

As is shown in Figure 1, it is the intention to acquire a spray with a – roughly – hemispherical shape. Details of the optimal spray shape will depend on the combustion concept and the shape of the piston bowl. In the next section we will briefly discuss how this can be achieved.

The outline of this paper is as follows. In the first section, some preliminary work with regard to the porous nozzle concept is reported. The second section deals with modeling of the porous injector. In the third and fourth sections the experimental setup and experimental results are discussed, respectively. Finally, a discussion of the results and some conclusions are presented.

**PRELIMINARY WORK**

In this section some preliminary work on the topic of injection through porous material is briefly discussed. To successfully inject via porous material, the properties of the material must be known. The most important are porosity and permeability. Both will be discussed here.

Porous material can be produced via sintering. With sintering, grains are pressed together at high pressure and are afterwards melted together at temperatures just beneath the melting temperature of the material. There are grains in many sizes, forms, types and materials, for example ceramics, metals, plastics, etc. The size of the grains and the pressure of the process determines for a large part the porosity and permeability of the sintered material. The porosity is defined as the volume fraction of holes in the material in relation to the total volume. The definition of permeability is the rate of flow of a liquid or gas through a porous material divided by the pressure difference that is driving the flow. In this study, stainless steel is chosen because of the favorable properties of this material in an engine environment (easy machining and high temperature resistance).

![Figure 2. Four configurations with different porosities and permeabilities.](image)

In Figure 2, examples are shown of different porosities and permeabilities. As becomes clear from the figure, a porous material is not by definition permeable, but a permeable material is by definition porous. In this study material is used that is closest to the lower left figure. Permeability and porosity depend on the internal structure of the material.

Darcy’s Law [7] is a well known (empirical) relation to describe the flow through a porous material. It can be derived from the Navier-Stokes equations under certain conditions [4] and it is defined as:
\[ \nabla \cdot \left( \frac{-\kappa_d}{\eta} \nabla p \right) = 0. \] (1)

Here, \( \kappa_d \) is the permeability as defined by Darcy, \( \eta \) the dynamic viscosity of the fuel and \( p \) is the pressure. The equation above was implemented in COMSOL Multiphysics in order to model the internal flow in the porous injector and calculate the Reynolds number. In this way, the optimal geometry of the porous nozzle is examined and Darcy’s Law can be validated. Criteria in the optimization of the porous injector are: the fuel mass flux (which should be at least equal to that of conventional injectors); the spray shape (which should resemble the hemispherical shape presented in Figure 1b); and the tensile strength (which should be larger than the tensile stresses in the nozzle, multiplied with a safety factor).

The geometries shown in Figure 3 were investigated to ascertain the influence of the length of the fuel channel. The sizes in the figures are in mm. The size of the outer diameter is chosen equal to the size of the conventional injector tip. To determine the size of the inner diameter, a few prototypes are made in which the inner diameter as shown in the figure, best agrees with the criteria mentioned above.

First, geometry (A) was investigated. On the inner edge the fuel pressure was prescribed, and on the outer edge a typical cylinder pressure at the time of injection. The flow through the porous tip is calculated with equation 1 where \( \kappa_d \) is reported by the manufacturer and \( \eta \) is a typical value of the dynamic viscosity for diesel. The values for these quantities are reported in Table 1. The “space dimension”, which is used in COMSOL Multiphysics, is 2D axial symmetry. The calculations are performed for steady-state and the influence of the flow through the channel is neglected. Whether this is justified is investigated in the next section.

### Table 1. Input parameters for simulations

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Fuel Pressure Geometry (A)</td>
<td>60 MPa</td>
</tr>
<tr>
<td>Fuel Pressure Geometry (B)</td>
<td>25 MPa</td>
</tr>
<tr>
<td>Cylinder Pressure</td>
<td>5 MPa</td>
</tr>
<tr>
<td>Density (( \rho ))[12]</td>
<td>850 kg/m(^3)</td>
</tr>
<tr>
<td>Dynamic viscosity (( \eta ))[12]</td>
<td>3.8·10(^{-3}) Pa·s</td>
</tr>
<tr>
<td>Permeability (( \kappa_d ))[13]</td>
<td>3·10(^{-12}) m(^2)</td>
</tr>
<tr>
<td>Mass flow</td>
<td>0.8 kg/s</td>
</tr>
</tbody>
</table>

From Figure 4, showing the internal velocity profile in the nozzle, it follows that there is a uniform spray velocity at the outer edge which should result in the spray shape as shown in Figure 1b. However, experiments reveal a spray shape as is shown in Figure 5. The experiments are performed with a Dedotec COOLH light source (illuminating the spray) and Dedotec COOLT3 power supply combined with a Phantom 7.1 high speed camera.

![Figure 3. Technical drawing of porous nozzle concept](image)

![Figure 4. Simulation of prototype with geometry (A); isolines of the internal velocity in the nozzle. Exit velocity is approximately 21 m/s.](image)
Figure 5. Experimental spray shape of injector with geometry (A). Atmospheric conditions, spray illuminated with high power light and captured with a high speed camera and experimental properties as listed in Table 2.

From Figure 5 it appears that the fuel spray is finely atomized, but the desired homogeneous hemispherical distribution is not realized. The fuel spray has a preferential axial direction, which means that geometry (A) does not have the desired fuel distribution of Figure 1b.

Table 2. Properties used for experiments

<table>
<thead>
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<th>Quantity</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>Material</td>
<td>Stainless steel</td>
</tr>
<tr>
<td>Mean pore size</td>
<td>10 µm</td>
</tr>
<tr>
<td>Porosity</td>
<td>32 %</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>100 MPa</td>
</tr>
<tr>
<td>Injection time</td>
<td>5 ms</td>
</tr>
<tr>
<td>Shot time</td>
<td>4 ms</td>
</tr>
</tbody>
</table>

Next, geometry (B) with the longer fuel channel was investigated. The pressure on the inner edge was lowered to reach the same mass flow as in the previous case. The other parameters were not changed.

Figure 6. Simulation of prototype with geometry (B); isolines of the internal velocity in the nozzle. Exiting velocity on the tip is 8 m/s.

From Figure 6 it was concluded that again the velocity on the outer edge was almost uniform. Experiments yield the spray shape as shown in Figure 7.

Figure 7. Experimental spray shape of injector with geometry (B); Same conditions as Figure 5.

Here, by changing the nozzle geometry, the spray shape is closer to the desired hemisphere bowl shape (Figure 1b). Of course the figure shows that the distribution locally still seems to be quite non-uniform. However, in the near future the porous material could be tuned to result in a more homogeneous distribution. This will be done by optimizing the production process. The numerical results do not (fully) correspond with the
In order to improve the numerical results, the model is adapted as will be explained in the next section.

STRESSES - The stresses in the material are also calculated. The maximum stress (Von Mises) that appears in the model with geometry (B) is 30 N/mm². The maximum allowable stress of the porous stainless steel is 90 N/mm² (from manufacturer), which indicates a safety factor of 3. This is a relatively low safety factor and experiments have demonstrated that it is not sufficient. In the Section “Experimental setup” another configuration is proposed to extend the lifetime of the porous injector.

MASSFLOW - The delivered power in diesel engines is controlled by the quantity of injected fuel. To ensure that the mass flow through the porous injector is the same as that of the conventional injector, mass flux measurements were performed by applying a high injection rate into a closed reservoir. The mass of the injected fuel is weighed and divided by the number of injections.

LIFETIME - To examine the lifetime of the porous injector, durability tests were performed. From experiments it is found that the injector tip breaks down repeatedly, at a location that is roughly indicated in Figure 9, after about 100,000 injections. This is probably due to fatigue. This value depends on the geometry, size and thickness of the porous material. To extend the lifetime more research has to be performed.

EXTENDED MODELING OF THE POROUS INJECTOR

It is clear from the previous section that the initial model is not in-line with the experimental results. A more extensive model is therefore proposed in this section. This model gives a better understanding of the fluid flow through the porous material and therefore the distribution of the spray can be predicted better. In the previous section, the flow through the porous material was obtained using Darcy’s Law. From literature, it is known that Darcy’s Law only holds for small Reynolds numbers. The Reynolds number (Re) expresses the ratio between inertial and viscous forces, in this case given by

\[ Re = \frac{\nu d \rho}{\eta} \]

where \( \nu \) is the velocity through the material, \( d \) the channel diameter (10 µm [13]), \( \rho \) the density of the fluid (850 kg/m³ [12]) and \( \eta \) the dynamic viscosity of the fluid (3.8·10⁻³ Pa·s [12]).

![Figure 9. Porous injector tip with indication of break line location](image)
First, the $Re$ number will be checked in the model as a function of the arc-length within the porous material of the nozzle, as shown in Figure 10.

![Figure 10. Configuration to calculate the $Re$ number within the nozzle](image)

The resulting Reynolds number, as predicted by the COMSOL Multiphysics model, is plotted in Figure 11.

![Figure 11. Reynolds numbers obtained from the COMSOL model as a function of arc-length](image)

Figure 11 shows that $Re$ is in the range of 15 to 190. From Figure 12, taken from Reference [3], it appears that Darcy's Law is not valid anymore in this range.

![Figure 12. Validity range of Darcy's Law [3]](image)

In Figure 12 the validation of Darcy's Law is shown in terms of Forchheimer variables Fanning friction factor ($f$) and Reynolds number ($Re$). The Fanning friction factor is a dimensionless number: $f = \frac{Δp \cdot d}{2 \cdot l \cdot ρ \cdot v^2}$, where $l$ is the mean channel length. The figure shows a transition from linear (Darcy's law) to nonlinear flow at $Re$ values between 5 and 10. Physically, this means that inertial forces can no longer be neglected with respect to viscous forces. Forchheimer (1901) proposed a nonlinear relation (Forchheimer's Law), which can also be derived from the Navier-Stokes equation [5]. Assuming stationary flow, incompressible fluid and neglecting gravity this leads to the following reduced NS-equation:

$$\nabla p = η\nabla^2 v - ρ(v\nabla)v.$$  \hspace{1cm} (2)

Via homogenization (Chen, [5]) we find:

$$\nabla p = -\frac{η}{κ_d} v - \frac{ρ}{κ_f} |v| v.$$ \hspace{1cm} (3)

In COMSOL Multiphysics this relation can be implemented in the Incompressible Navier-Stokes application in the following form:

$$0 = \nabla \cdot (p I + \frac{η}{κ_d} v + \frac{ρ}{κ_f} |v| v),$$ \hspace{1cm} (4)

where $p$ is the pressure, $η$ is the dynamic viscosity, $v$ the velocity vector, $ρ$ is the density of the fluid and $|v|$ is the absolute value of the velocity vector. $κ_d$ is the permeability defined by Darcy and $κ_f$ is the permeability defined by Forchheimer. Geometry (B) is again modeled in COMSOL Multiphysics but now using Equation (4) instead of (1). Also the fuel channel is modeled with a standard $k-ε$ turbulence model [9] instead of a boundary condition on the inlet side. Also, the porous material is modeled in greater detailed as shown in Figure 13.
Figure 13. Simulation of prototype with geometry (B) with use of Darcy-Forchheimer equation. Fuel pressure = 100 MPa.

Figure 13 shows the velocity profile after Forchheimer’s term is added. To reach the same overall mass flow, the fuel pressure is increased to 100 MPa. For other properties see Table 1. Now, by adding Forchheimer and more details, the velocity difference in outflow velocity between the tip and the side is larger. The exit velocity, the surface area and the porosity are known, so the model results can be qualitatively compared with the experimental results. After the model updates, the results now correspond to the experimental results, which will help to improve the quality of the prototypes.

From previous experiments is known that the lifetime of the porous nozzle does not satisfy the required 400 million injections. To extend the lifetime and optimize the spray shape of the porous injector the geometry is modified. The external radius was changed from 0.85 mm to 2.5 mm. The wall thickness was intended to be uniform (at 2 mm), but because of a production problem the fuel channel was drilled too deep and therefore the wall thickness on the tip is smaller than on the sides. This geometry (Figure 15) was simulated in COMSOL, resulting in the velocity contours shown in Figure 14.

EXPERIMENTAL SETUP

The Eindhoven High Pressure Cell (EHPC) [6] is a high pressure, constant volume vessel with optical access in which engine like conditions can be created, regarding oxygen level, temperature and pressure (up to 30 Mpa).

From Figure 14 it is expected that the spray shape has a preference in the axial direction. In the next sections, this porous nozzle will be tested experimentally.

A picture of the EHPC is shown in Figure 16.
The engine-like conditions can be obtained by a pre-combustion method [6]. As will be discussed later in this paper, a pre-combustion increases the temperature and pressure inside the EHPC, as shown in Figure 18. In-cylinder pressure and temperature can be chosen independently from each other. The pressure is controlled (indirectly) by the amount of filling gases and the gas temperature by the moment of injecting the test fuel. Both non-reacting (vaporizing) and reacting sprays are analyzed in the next sections.

NON-REACTING SPRAYS - To analyze the vaporization of the spray produced by the porous injector, a reaction between the fuel and oxygen is undesirable. A simple way to realize this is to put exactly the stoichiometric amount of oxygen in the burner vessel needed for the pre-combustion. After the pre-combustion no oxygen is left and the injected fuel will not burn. These vaporizing sprays are analyzed with the Schlieren technique [8].

The Schlieren method uses the difference in index of refraction between fuel vapor and air. In Figure 17, a schematic drawing of the method is shown. A powerful light (point) source generates a beam in the direction of the EHPC. With a lens, the source is collimated to form a broad parallel beam. After passing through the test vessel another lens focuses the light onto a pinhole and a high speed camera (Phantom 7.1) captures the light. In the presence of density gradients the beam is diverted onto a Schlieren stop, causing the density gradients to appear as darker regions on the camera lens.

Due to the pre-combustion, density gradients are created in the EHPC before the fuel is injected. When analyzing the recordings, one sees disturbances in the background of the spray. For low density conditions the disturbances are smaller compared to high ambient density conditions. However, the spray can still be distinguished from the background. To clarify the spray, a boundary line is drawn in the Figure 19.

REACTING SPRAYS – In the previous section, it is explained how vaporizing fuel sprays are analyzed. Now we will discuss reacting sprays. To induce auto-ignition, excess oxygen is added to the EHPC. The engine conditions are realized in the same way as with non-reacting sprays. The density is governed by the filling conditions (and is constant because of constant volume vessel), temperature and pressure are controlled by the moment of injection. In Figure 18 an example is given of a pressure curve with a reacting spray.
First, a pre-combustion takes place to achieve the high temperature and pressure. After that, the mixture cools down and at a certain point, when the temperature has the desired value, the diesel injection is performed. Depending on the ignition delay at the experimental conditions chosen, the fuel reacts with the oxygen and the temperature as well as the pressure rise again, as can be clearly seen in Figure 18 at $t = 0.775$ s.

To test the functionality of the injector, the ignition delay is examined. Ignition delay is defined as the time between the start of injection and the Start of Combustion (SOC). The measurements are performed as discussed in [14]. The moment of start of injection is known and the start of pressure rise in Figure 18 (minimum pressure) is taken as the SOC, since the experiments are performed in a constant volume vessel.

**EXPERIMENTAL RESULTS**

Figure 19 shows a Schlieren image of a spray from the porous injector. The spray does not have the desired homogeneous hemispherical distribution as shown in Figure 1b. The fuel spray, still has a preferential axial direction, however, there is a wider distribution than is the case in Figure 5.

![Figure 19. Non-reacting spray of the porous injector measured with the Schlieren technique. Conditions at the moment of fuel injection: $p_g = 1.44$ MPa, $T_g = 671$ K](image)

To predict the SOC the ignition delay is measured. In Figure 20 ignition delay versus temperature is plotted.

![Figure 20. Ignition delay: $\rho_g = 16$ kg/m$^3$, $O_2$ Vol%$=21$, fuel = EN590, $T_{inj} = 5$ ms.](image)

Figure 20 shows that the ignition delay is shorter for the porous injector (*) compared to a conventional injector (□), especially at lower temperatures. At higher temperatures the differences are smaller.

The combustion process is also observed. Figure 21 shows the combustion with the porous injector. In this case, the combustion spreads out over a large region in space. This suggests that better use is made of the available oxygen, which is expected to result in an overall leaner combustion. However, this can not yet be substantiated with quantitative results. Analysis of soot luminosity is subject of current investigation.

![Figure 21. Reacting spray of porous injector; $p_{inj} = 100$ MPa, $\rho_g = 16$kg/m$^3$ $T_g = 1000$ K, fuel = EN590, $O_2$ Vol% = 21, $T_{inj} = 5$ ms).](image)

**DISCUSSION**

During preliminary research [10] two different geometries of porous injectors were investigated numerically and experimentally and compared with a conventional injector. First, the spray shapes of the numerical models
were compared with experiments. From this comparison it became clear that the results did not match. In the model, geometry (A) shows a homogeneous distribution over the exit edge. In the experiments the spray has a preference in the axial direction. The results of geometry (B) are slightly better. The deviation in the experiments is due to oversimplifications in the model. First, the momentum of the fuel is not taken into account because the fuel channel is not modeled. Furthermore, the velocities in the porous material are such that the Reynolds (Re) number exceeds 5. The flow through porous material, however, can only be described well by Darcy’s Law up to Re numbers of 5. In this study, a nonlinear term (Forchheimer term) \[ f_{Re} \] is added and more details (fuel channel) are modeled. Now the models show a reasonable level of agreement with experiments.

The mass flow (at a given fuel pressure) through the porous injector is higher than is the case for the conventional injector (Figure 8), at least for the nozzle material and geometry used in this preliminary study. These results indicate that the desired mass flow can be realized with the investigated porous injector.

The desired lifetime of the injector is typically >1.000.000 km, the average speed over the whole life being 80 km/h at an engine speed of 1200 rpm. In this case, the injector has to inject approximately 400 million times during its lifetime. From the experiments we found that the injectors broke down after about one hundred thousand injections. This means that the lifetime of the injector is far too short. To extend the lifetime, the geometry has to be optimized, other materials have to be investigated and the production process has to be improved. This will be investigated in a later phase of the project.

As mentioned earlier, a better mixing of fuel with air leads to a decrease in soot emissions. It is likely that an adequate (i.e. to facilitate auto-ignition) fuel-air mixture is obtained faster and this leads to a shorter ignition delay. For a given fuel, the ignition delay is thus assumed to be a (local) measure for mixing quality. From the experimental results it is known that the ignition delay is much shorter for the porous nozzle, which suggests that the mixing quality of the spray process is improved.

**CONCLUSIONS**

A new fuel injector concept, based on a layer of sintered steel, was investigated numerically and experimentally. A common-rail setup has been adapted to test the porous injector. To analyze the spray at atmospheric conditions, images of the spray were captured with a high speed camera. The spray of the first prototype had a preferential axial direction instead of the desired homogeneous hemispherical distribution, which does not correspond with first simulations. After modifying the model, a new prototype was made with a larger outer diameter. The spray shape of this prototype was tested in the EHPC (constant volume chamber). It was found that the spray was not hemispherical because of a production problem.

The sprays from the simulations are compared with the experimental findings. First simulations, with Darcy’s Law did not (fully) correspond to experiments. Afterwards, when a Forchheimer term was added and more details were modeled, the results of the simulations corresponded better to the experiments.

The mass flow through the porous injector has to be at least the same as a conventional injector. From the experiments it becomes clear that the mass flow of the porous injector was higher at an equal fuel pressure. However, since this was just a prototype, this does not prove that this is generally the case for porous nozzles.

The durability of the porous injector was tested. First results indicated that normal lifetime requirements are not yet met. More research has to be performed to extend the lifetime of the injector.

Non-reacting, vaporizing sprays have been observed to estimate the spray shape. In this case the spray does not have the fully hemispherical shape due to the fact that at the moment the manufacturing process does not have the right accuracy. New prototypes have to be produced and will be tested in the future. However, the spray from the porous nozzle does have a more homogeneous distribution, based on preliminary spray visualization experiments.

Ignition delays have been measured and are found to be shorter for the porous injector than for the conventional one, especially at lower temperatures. It is very difficult to explain this right away. On the one hand air entrainment could be larger compared to the conventional. On the other hand the entrainment is expected to be very slow as the droplet speed is low. The volume of air taking part in the combustion of the porous spray may be much larger compared to conventional injectors. However, based on current experimental evidence, a firm conclusion can not yet be drawn, so that interpretation remains speculative. Further research is currently being conducted into several aspects of the porous injector nozzle.

**ACKNOWLEDGEMENTS**

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NOMENCLATURE

Latin symbols:

DI Direct Injection
d Channel diameter
EHPC Eindhoven High Pressure Cell
HCCI Homogeneous Charge Compression Ignition
HD Heavy Duty
ICE Internal Combustion Engine
K Kelvin
PCCI Premixed Charge Compression Ignition
p Pressure
Re Reynolds
SOC Start of combustion
T Temperature
TDC Top Dead Center
v Velocity

Greek symbols:

µ Micro = 10⁻⁶
κ Permeability
η Dynamic viscosity
ρ Density

Subscripts:

d Darcy
f Forchheimer
g Gas
inj Injection
conv Conventional
por Porous