An Investigation of Flow in Nozzle Hole of Dimethyl Ether

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Abstract. For over twenty years, DME has shown itself to be a most promising fuel for diesel combustion. DME is produced by simple synthesis of such common sources as coal, natural gas, biomass, and waste feedstock. DME is a flammable, thermally-stable liquid similar to liquefied petroleum gas (LPG) and can be handled like LPG. However, the physical properties of DME such as its low viscosity, lubricity and bulk modulus have negative effects for the fuel injection system, which have both limited the achievable injection pressures to about 500 bar and DME’s introduction into the market. To overcome some of these effects, a common rail fuel injection system was adapted to operate with DME and produce injection pressures of up to 1000 bar. To understand the effect of the high injection pressure, tests were carried out using 2D optically accessed nozzles. This allowed the impact of the high vapour pressure of DME on the onset of cavitation in the nozzle hole to be assessed and improve the flow characteristics.

1. Introduction
Dimethyl Ether (DME) is an excellent fuel for diesel combustion \cite{1, 2, 3}. It burns completely soot free and produces less NOx than the equivalent diesel. However, it is a liquefied gas of lower bulk modulus at normal temperatures. In addition, the energy content of DME by volume is also about half of that of diesel, so that nearly twice as much DME has to be injected to produce the same power. This combination of low injection pressure and large injection nozzle flow area for DME has meant that the fuel spray from the injection nozzle has been better atomised, but has shorter penetration than the equivalent diesel \cite{4}. Thus the objective of the work is to improve the flow characteristic of the nozzle hole without detriment to the spray ultimately the combustion.

2. Flow Analysis in Spray Holes
The flow characteristics of diesel injection nozzle spray holes have been improved by such measures as inlet edge rounding, by extrusion honing or hydraulic grinding \cite{5, 6}. In addition, tapering the shape of the holes has also improved the flow characteristics by reducing the cavitation tendency of the fluid. As DME has a higher vapour pressure than diesel, it will tend to cavitate more readily. Therefore, it was decided to compare the effect of these measures with DME to see how the flow characteristics and cavitation tendency of DME was affected by the hole shape and the effects on the spray break up \cite{7, 8, 9, 10}.

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2.1. Flow and optical tests on different throttle shapes
In order to study the flow differences between the various spray hole shapes, two dimensional models were cut out of 0.3mm thick plate with the geometries required. The test set up was designed to enable optical access to the throttles and also measure the throttle flow. The setup is shown in Figure 1. The plates were sandwiched between two optical quality quartz glass windows which were then clamped together between steel plates. Fuel inlet and outlet connections and windows were provided in the steel plates so that the fuel flow and flow regimes could be observed. The fuel temperature and pressure were measured before and after the throttle. Steady state flow tests were carried out with different throttle shapes to simulate the flow through injector nozzle spray holes. The flow is adjusted by changing the backpressure after the throttle. The fuel is fed at different pressures through the apparatus and the geometry of the throttle is cut out of a thin metal plate sandwiched between two glass discs. The start and development of cavitation in the orifice are of particular interest as these affect the spray break up after the throttle. In order to investigate the specific nozzle hole geometries given in the test they had to be approximated by rectangular flow channels. For this purpose, three different hole shapes were prepared for assessment, namely a straight, sharp edged hole and a plus & minus tapered hole with rounded inlet edges. A straight and tapered hole shapes are typical of today’s injection nozzles for diesel fuel. These geometries are shown in Figure 1.
Type 2 has tapered value k4 of k-factor. Type 4 has k-5 and longer hole length than Type 1 & 2. The k-factor is defined as percent of hydraulic taper ratio (HTR) shown following equation.
\[ k = \frac{(D_i - D_o)}{L} \times 100 = \frac{100HTR}{L} \]
Where, \( D_i \) = inlet diameter of hole (inlet width of throttle)  
\( D_o \) = outlet diameter of hole (outlet width of throttle)  
\( L \) = hole length (throttle length)

Fig. 1. Test set up to observe flow through throttles, and throttle type 1, 2, & 4

2.2. Flow characteristics and cavitation with optical access
The flow tests through the throttles were carried out with inlet pressures of 200, 400, and 600 bar. The pressure difference was changed by varying the back pressure. The characteristic of the Type 2 nozzle, compared to the Type 1 shown in Figure 2 in case of inlet pressure 200bar. The curves are less inclined and the choked 2-phase area is less distinctive. As shown in Figure 2, the flow with throttle type 2 is 30% higher than for type 1 which would result in correspondingly shorter injection duration. By analyzing the images over 50 shots, a statistical picture can be formed which helps to understand how and where the cavitation is formed and develops. The red colored areas show purely liquid and the blue areas are purely vapor. The color change via yellow and green to blue are regions where the cavitation occurrence is increasing. This flow improvement of Type 2 is a result of the inlet rounding and the tapering effect of the throttle shape. And the maximum flow through the Type 2 throttle occurs
at a higher pressure difference than for Type 1. Any further increase in pressure difference does not increase the flow rate. The flow rate can only be increased by increasing the inlet pressure.

These measurements were made in a way that the pressure upstream the nozzle was kept constant and the back pressure was varied. Thus, characteristic curve progressions are achieved which clearly distinguish different flow regimes. The hydraulic flow follows the Bernoulli relationship. It may either consist of a pure liquid flow regime or show local cavitation zones:

\[ m_{hyd} = C_{hyd} A \sqrt{2 \Delta P \cdot \rho} \]

Where, \( m_{hyd} \) = mass flow, \( C_{hyd} \) = flow coefficient, \( A \) = cross sectional area of throttle, \( \Delta P \) = pressure drop across the throttle = \( P_{in} - P_{out} \), \( \rho \) = density of the fluid

At specific conditions, the cavitation zones extend over the whole cross section which causes a choking of the flow due to an almost homogeneous 2-phase flow regime. This regime is compressible like a gas but it has a considerably high density due to the liquid inclusions. As a consequence, the local speed of sound drops which leads to the choked flow characteristics. Fig. 2 shows that in this regime the mass flow rates remain constant if the upstream pressure is kept constant. In other words, the flow characteristic is dependent on the throttle back pressure until a pressure difference is reached at which the flow no longer increases and remains constant independent of the backpressure. The flow characteristic of the choked flow regime is given by:

\[ m_{2ph} = C_{2ph} A \sqrt{2 \Delta P \cdot \rho} \equiv C_{2ph} A \sqrt{2 P_{in} \cdot \rho} \]

Where, \( m_{2ph} \) = mass flow, \( C_{2ph} \) = flow coefficient at choked flow condition, \( P_v \) = vapor pressure. Since \( P_v \) is negligible small, \( P_v \) is able to be assumed zero. According to these considerations the flow characteristics of different throttles can be comprehensively defined by the two flow coefficients \( C_{hyd} \) and \( C_{2ph} \). In this context it is emphasized that the combination of these two coefficients even defines the particular flow regime of any individual state. This can be explained by the following simple consideration. At the transition between hydraulic flow and choked 2-phase flow the mass flow rates from the above formulas can be equalized. This leads to the following identity

\[ C_{hyd} A \sqrt{2 \Delta P \cdot \rho} = C_{2ph} A \sqrt{2 P_{in} \cdot \rho} \]

or re-arranged:

\[ \frac{P_{in} - P_{out}}{P_{in}} = \frac{C_{2ph}}{C_{hyd}}^2 \]

This simple formula indicates that the transition point can be clearly defined by the flow coefficients and that hydraulic flow appears at \( \frac{P_{in} - P_{out}}{P_{in}} < \left( \frac{C_{2ph}}{C_{hyd}} \right)^2 \), and choked 2-phase flow at \( \frac{P_{in} - P_{out}}{P_{in}} > \left( \frac{C_{2ph}}{C_{hyd}} \right)^2 \). In this paper the transition point to the choked 2-phase flow is denominated as \( K_{lim} \).

\[ K_{lim} = \left( \frac{C_{2ph}}{C_{hyd}} \right)^2 \]

The flow coefficients, \( C_{hyd}, C_{2ph}, \) & \( K_{lim} \) of the two optical nozzles are shown in Table 1.

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Fig. 2. Cavitation development & mass flow in type 1 & 2 throttle as a function of pressure difference
It should be noted that $K_{\text{lim}}$ shows the tendency to enter the choked 2-phase flow regime.

Table 1 Flow and cavitation coefficients of the tested throttles

| Nozzle Type | $C_{\text{hyd}}$ | $C_{\text{2ph}}$ | $K_{\text{lim}}$ |
|------------|----------------|----------------|-----------------|
| Type 1     | 0.830          | 0.660           | 0.58            |
| Type 2     | 0.924          | 0.798           | 0.72            |

The flow test in the Type 4 throttle was also carried out. The Type 4 throttle with expanding flow area after the throat has $C_{\text{hyd}}$ values greater than 1. This indicates that the recovering the pressure after the narrowest section in Type 4 is better than the straight and converging throttles. The expanding throttle has a higher total flow capacity than the converging ones. Decreasing $P_{\text{out}}$ will not affect the flow once cavitation has begun in the throttles.

3. Conclusion
The optical throttles indicated that similar effects to those seen with diesel sprays could be seen with DME: i.e. increasing the radius of the inlet to the spray hole and tapering the form of the hole, the intensity of the cavitation in the fluid at the exit from the spray hole could be reduced. As a consequence it would be expected that the spray would be better atomized and have higher penetration as well as an increase in flow rate. The results showed that different nozzle hole shapes changed the flow and the cavitation onset and tendency. These characteristics should be utilized for an optimization of the spray and combustion.

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