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Injector Nozzle Hole Parameters and their Influence on Real DI Diesel Performance

MIKAEL LINDSTRÖM

Licentiate thesis
Department of Machine Design
Royal Institute of Technology
SE-100 44 Stockholm

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Academic thesis, which with the approval of Kungliga Tekniska Högskolan, will be presented for public review in fulfillment of the requirements for a Licentiate of Engineering in Machine Design. The public review is held at Kungliga Tekniska Högskolan, Brinellvägen 64, Stockholm in room M36, 26th of January 2009 at 14:00.

Pour Camille, Jane, Gunnar, Pierre, Desirée, Felicia et Magnus.

I. Abstract

A modern diesel engine is capable of running efficiently with low exhaust gas emissions over a wide operating range. This is thanks to techniques such as turbocharging, EGR, charge air cooling and an advanced fuel injection process. The fuel injection process is important for the combustion and emission formation in the diesel engine. The fuel injector has to atomize and vaporize the fuel as it is injected. During the combustion the emission formation has to be kept to a minimum. Very strong pressure gradients are present in a modern diesel injection nozzle, this causes cavitation to occur in the nozzle holes. The influence of cavitation on flow parameters such as the various discharge coefficients is discussed. The occurrence of cavitation helps the spray break up and it can keep the nozzle holes free from deposits. Excessive amounts of cavitation can lead to hole erosion and thus impact the long term operation of the nozzle in a negative way. Hole erosion as well as other mechanisms can cause hole to hole variations in fuel spray impulse, mass flow, penetration etc. This is a very important issue in any low emission diesel engine, especially during transients, as less than optimal conditions have to be handled. The influence of hole to hole variation on fuel consumption and emissions is not very well known and this thesis contributes to the field. As a part of this work a fuel spray momentum measurement device was developed and tested.

Any automotive engine needs to be able to perform quick transitions between different loads and speeds, so called transients. In a turbocharged diesel engine with EGR issues related to the turbocharger and the EGR-circuit arise. A diesel engine has to run with a certain air excess in order to achieve complete combustion with low emissions of soot. When turbocharging is used the turbocharger turbine uses some of the exhaust enthalpy to drive the turbo compressor, in this way the engine is provided with boost pressure. In order for the engine and turbocharger to function at the higher load and thus higher mass flow rate the turbocharger has to increase its rotational speed and the surface temperatures have to settle at a new thermodynamic state. Both of these processes take time and during this time the combustion process may have to proceed under less than optimum circumstances due to the low boost pressure.

II. Acknowledgements

I would like to thank the following people for valuable ideas and help with this thesis and the papers it contains: Hans-Erik Ångström, Ernst Winklhofer, Jonas Holmborn, Lars Dahlén, Andreas Cronhjort, Fredrik Wåhlin, Per Risberg and Anders Björnsjö.

III. List of papers

Paper I

Lindström, M., Ångström, H-E., "Development and Testing of Some Variants of a Fuel Spray Momentum Measurement Device" SAE World Congress 2009 (submitted)

Paper II

Lindström, M., Ångström, H-E., "A Study of Hole Properties in Diesel Fuel Injection Nozzles and its Influence on Smoke Emissions" THIESEL 2008

IV. Table of contents

1 Introduction

An internal combustion engine has the difficult task of transforming chemically bound energy into mechanical work. The first stage of the process is to transform the chemical energy in the fuel into heat by combustion, this can be done with almost 100 % efficiency, the difficult part is to turn the heat into mechanical work with high efficiency. A combustion engine can be seen as a combustion system coupled to a heat engine, see Figure 1.

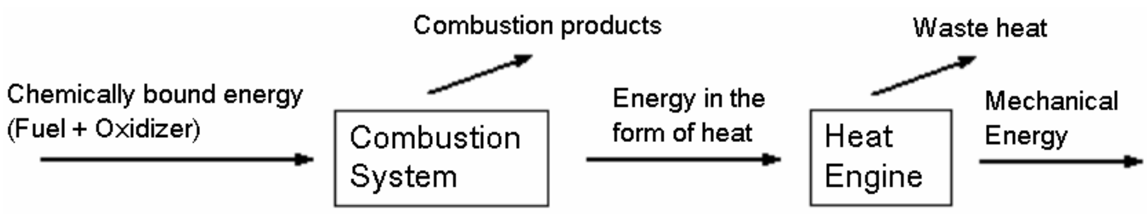

Figure 1. Principal layout of a combustion engine.

This basic sketch can be used to describe any type of combustion powered engine including piston engines, gas turbines, rockets, power plants etc. In a diesel engine the combustion system consists of the fuel injector and the combustion chamber. In a spark ignited (SI) engine the combustion system consists of a sparkplug and the combustion chamber. In a direct injected (DI) engine the combustion chamber mostly consists of a piston bowl. In both types of engine the heat engine is made up of the piston mechanism.

In the case of a hydro carbon fuel the ideal combustion process would apart from heat result in nothing but carbon dioxide and water vapor. In reality thousands of chemical species are formed. Some of these are toxic or environmentally hazardous enough and formed in enough amounts that it has been necessary to legislate about maximum allowed emitted levels. Figure 2 shows the legislated emission levels of nitrous oxides (NO_x) and particulates (PM) for heavy duty (HD) diesel engines in Europe, source for data: *dieselnet.com* [1] (somewhat simplified diagram, Euro 6 not finalized).

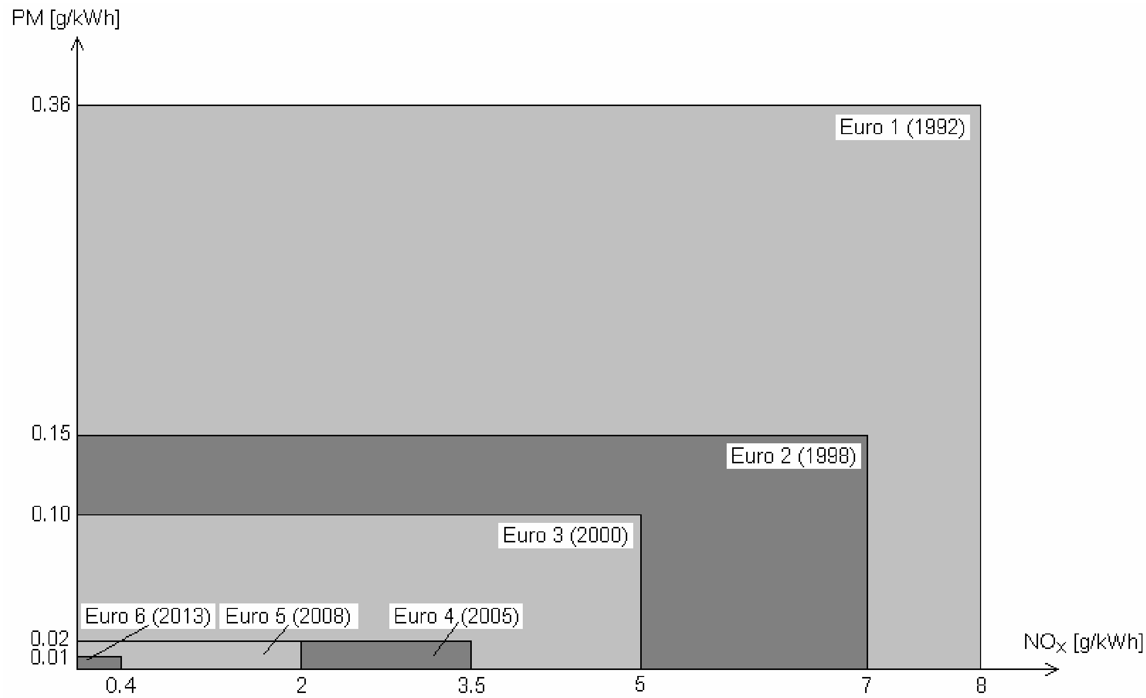

Figure 2. Legislated emission levels of NO_x and PM for Heavy Duty in Europe. [1]

It can be seen that the legislated levels have been reduced substantially over the past 15 years. Other substances like unburnt hydro carbon and carbon monoxide are also regulated. Generally NO_x and particulates are considered the two types of emissions that are the most demanding to reduce.

Two basic types of piston engines exist, the SI or gasoline engine and the compression ignited diesel engine. In the SI engine the fuel is mixed with the air when it flows into the intake ports or in the case of direct injection inside the cylinder during the compression stroke. The charge is compressed and ignited with a spark plug and burns with a flame front starting at the spark plug and moving outwards in the cylinder until the entire charge has been burnt. This approach has the advantage that it is possible to run the engine at stoichiometric air/fuel ratio which allows the use of a fairly simple yet highly efficient after treatment system, the three way catalyst. Another advantage is that the speed of the combustion process is controlled by the amount of turbulence in the cylinder and that this turbulence increases when the engine speed increases. The engine speed is thus mainly limited by mechanical factors such as valve train and piston speed. This makes it possible to build a SI engines with very high power density. In the SI engine the fuel is already mixed into the air before or during the early stages of the compression stroke. If the compression is too high the charge will auto ignite and result in loss of combustion phasing control, excessive pressure rise rates and increased heat transfer to the combustion chamber walls. The efficiency of the SI engine is limited by the fact that the compression ratio has to be limited.

As the SI engine only can operate close to stoichiometric air/fuel ratio the airflow has to be throttled at part load which substantially limits part load efficiency. The diesel engine on the other hand is not limited by knock since the fuel is not present in the charge during compression. Therefore high compression ratios in combination with high boost pressures are possible. In the diesel engine the power output can be regulated by only changing the fuel flow, no throttle which would decrease part load efficiency is necessary. The main challenge for the diesel engine is its emissions. The diesel engine ingests and compresses a gas charge in which no fuel is present. When the fuel is injected around top dead center it auto ignites because of the high temperature and pressure. The initial part of the combustion is premixed due to the ignition delay but the main part of the combustion process consists of mixing controlled diffusion flames. In order to achieve complete combustion without excessive amounts of soot formation with such a process it is necessary to have an air/fuel ratio higher than stoichiometric, a so called air excess. Because of the high combustion temperatures nitrous oxides are formed as some of the nitrogen present in the charge is oxidized. A number of techniques have been introduced over the years to reduce the emissions produced and improve the efficiency and power density of the diesel engine. These include turbocharging, exhaust gas recirculation (EGR), charge air cooling and improvements to the fuel injection process. The fact that it is very hard to match the efficiency of a diesel engine makes it the primary engine choice for heavy vehicles as well as for a growing part of light vehicles in spite of the R&D-intensive emission reduction efforts that have been made and still are necessary.

2 Diesel combustion and emission formation

In a diesel engine the fuel is injected into a highly compressed gas volume. The temperature and pressure of the gas causes the fuel to auto ignite. Some residence time is required for ignition as the thermo chemical reactions involved do not take place instantaneously. Therefore the initial phase of the combustion event is premixed since some fuel has had time to mix with air during the ignition delay. After the premixed phase the combustion continues with fuel being burnt in mixing controlled diffusion flames.

A conceptual model of DI diesel combustion proposed by Dec [2] is a widely accepted description of the principles of mixing controlled combustion. Figure 3 shows the proposed composition and progression of the diesel flame as a function of crank angle degrees after start of injection (ASI). The initial stage (0.0° - 4.5° ASI) of the fuel spray development includes atomization, vaporization and air entrainment into the jet. A growing vapor phase develops around the spray that eventually forms the so called head vortex. The auto ignition phase ranges from 3° - 5° ASI. Using chemiluminescence the start of combustion can be detected at around 3.5° ASI. The occurrence of poly-aromatic hydrocarbons (PAH) can be detected between 4.5° - 5° ASI in the fuel vapor-air mixture. This is followed by soot formation between 5° - 6° ASI. The part of the combustion corresponding to the premixed spike in the heat release rate (HRR) starts at 4° - 6.5° ASI. The HRR starts to increase sharply at 4.5° ASI. At this point the leading portion of the spray is highly chemiluminescent but there is little sign of significant fuel break down. At 5° ASI the fuel breaks down and large PAH´s form across the leading portion of the spray where the equivalence ratio is 2 – 4, i.e. fuel rich. By 6° ASI soot starts to occur as small particles throughout the downstream portion of the jet, the pattern is subject to large cycle to cycle variation.

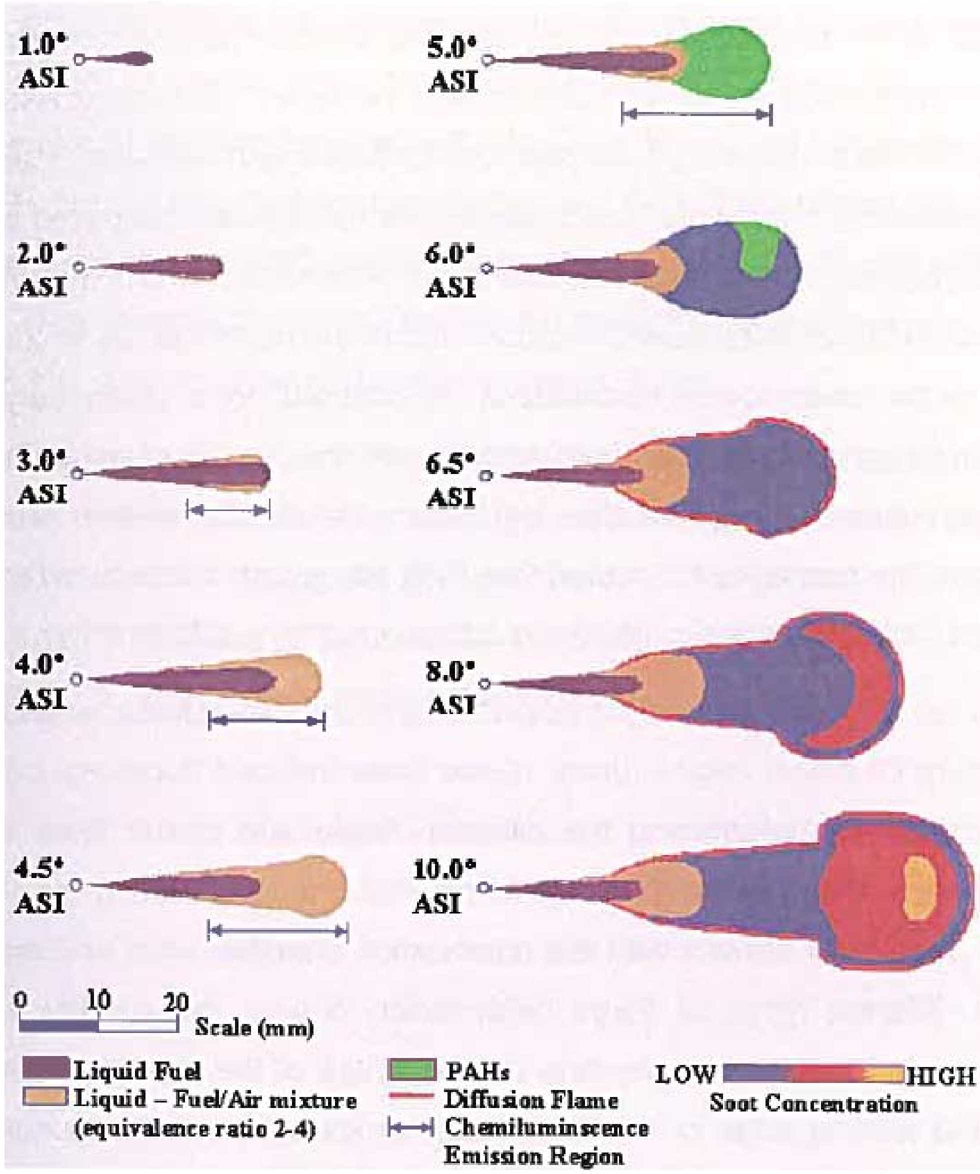

Figure 3. Conceptual model of DI diesel combustion by Dec. [2]

These particles arise from fuel rich premixed combustion. At 7° - 9° ASI the premixed spike in the HRR ends and a non-transient mixing controlled flame has developed. The mixing controlled flame forms and starts to stabilize at 5.5° - 6.5° ASI as can be seen as a thin line encircling the flame at 6.5° ASI in the Figure 3. This leads to a reduction of liquid length, probably because of local heating. There is a high soot concentration zone close to the leading edge inside the developed diffusion flame. These particles are larger than the particles that are also formed around the circumference of the liquid core. Thermal NO formation occurs around the hot circumference of the diffusion flame as is shown on the Figure 4 from Charlton [3].

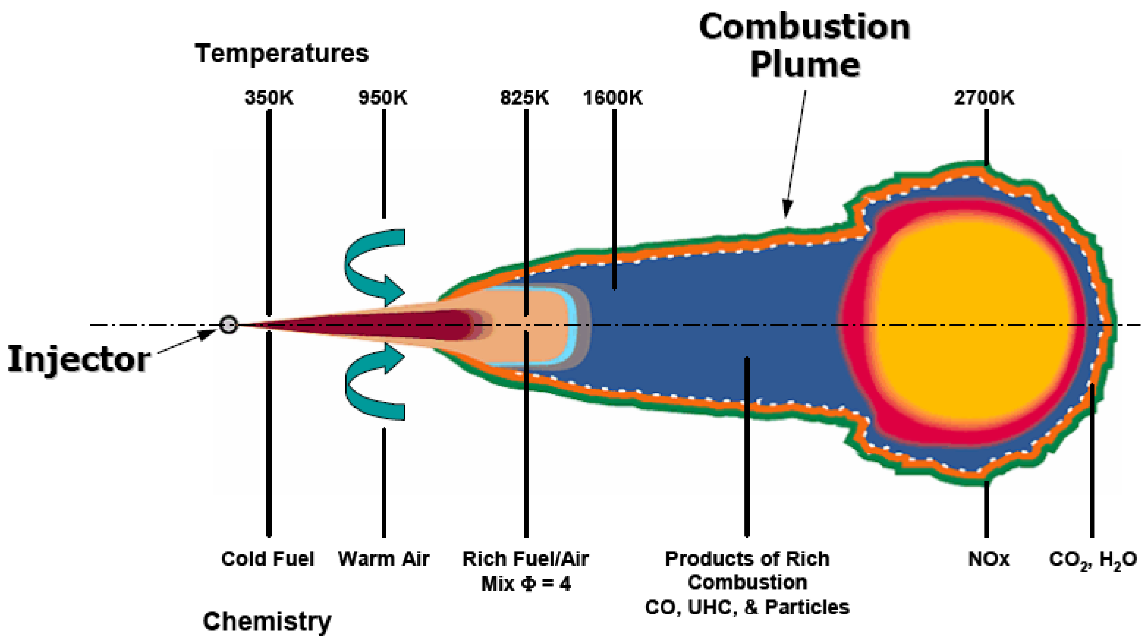

Figure 4. Schematic of a diesel flame with temperatures and chemistry. [3]

Figure 4 illustrates temperatures at various locations in a diesel flame. The highest temperature occurs on the flame surface and this is also where the NO_x is formed.

Picket and Siebers [4] investigate non sooting low temperature diesel combustion by studying diesel sprays and flames with various diagnostics methods. Fuel jet air entrainment is estimated from fuel jet cone angle. In traditional mixing controlled diesel combustion, high levels of nitrous oxides (NO_x) and soot are formed. The temperatures can exceed 2600 K which leads to NO_x-formation as the nitrogen of the gas charge is oxidized. Soot formation occurs inside the envelope of the flame in the fuel rich regions of the jet. Soot concentrations can be very high during the combustion. However, most of the soot is burnt off before the opening of the exhaust valve. Using O_2-concentrations as low as 10 % to simulate the use of Exhaust Gas Recirculation (EGR) they can limit the flame temperature to 1980 K while still avoiding soot formation. They explain the lack of soot formation with fuel air mixing upstream of the lift off length.

3 The fuel injection process

As mentioned before the purpose of a combustion system in an engine is to burn the fuel and thus turn it into heat. The characteristics of the combustion process in a diesel engine are partly determined by the gas state in the combustion chamber determined by factors such as boost pressure, compression ratio, charge temperature and EGR-rate. The fuel injection process also has a major influence on the combustion and emission formation processes. Factors that strongly influence the atomization and combustion of the fuel are injection pressure, fuel injection timing, hole parameters, the interaction between the fuel sprays and the geometry and gas flow in the combustion chamber, the use of multiple injections and rate shape. Some complex mechanisms are involved in the mechanical interaction between the initial liquid fuel jet and the gas charge as the fuel is atomized prior to combustion. The following chapter describes some mechanical aspects of the fuel spray.

3.1 The fuel spray

When high pressure fuel exits a nozzle hole a jet is formed. The jet can disintegrate through a variety of mechanisms. Figure 5 from Lefebvre [5] shows a liquid jet with surface wave instabilities.

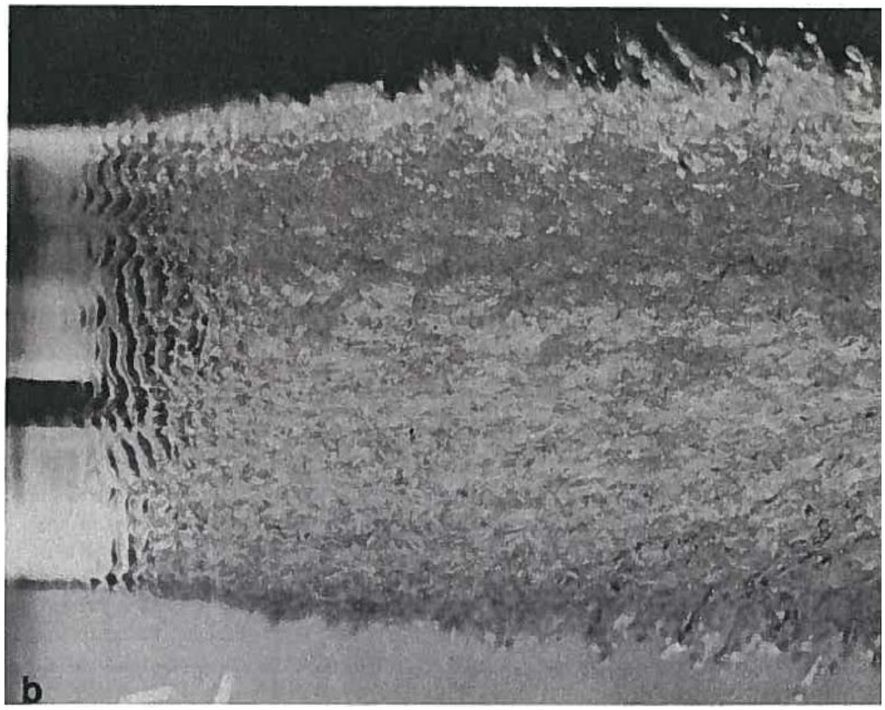

Figure 5. Liquid jet with surface wave instabilities and break up. [5]

The surface waves are a consequence of the so called Rayleigh jet break up mechanism. This results in large droplets which may proceed to break up further. The Reynolds number and the Weber number are two important parameters for fuel sprays. They are defined as:

The Reynolds number: $$Re = \frac{\rho \cdot U \cdot d}{\mu} \quad (1)$$

The Weber number: $$We = \frac{\rho\, U^2 d}{\mu} \quad (2)$$

Where: U – Velocity

d – Droplet diameter

μ – Viscosity

ρ – Density

σ – Surface tension

The *Re*-number is the ratio between inertial and viscous forces and the *We*-number is the ratio between momentum force and surface tension force. According to Lefebvre [5] several modes of liquid disintegration exist, see Figure 6.

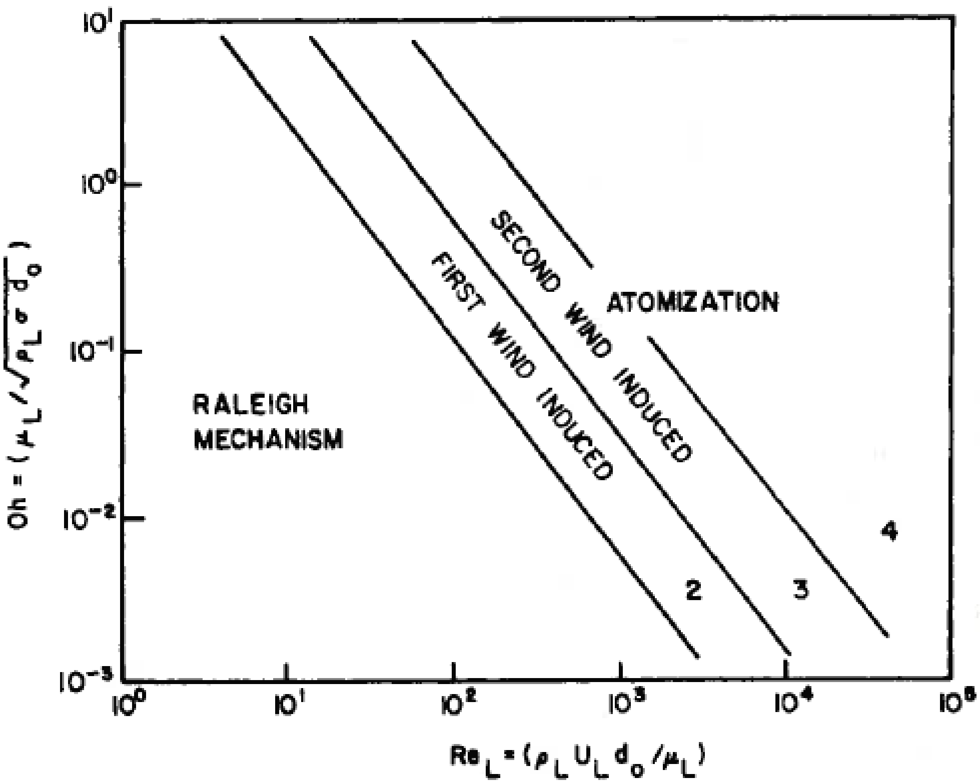

Figure 6. Modes of liquid disintegration. [5]

These modes depend on the Reynolds number and the Ohnesorge number which in turn is a function of the Reynolds number and the Weber number. In the first and second wind induced regimes the fuel jet is broken into large droplets which in turn break into smaller droplets. Lee and Reitz [6] have made a review of various break up the theories. The table in Figure 7 illustrates these mechanisms.

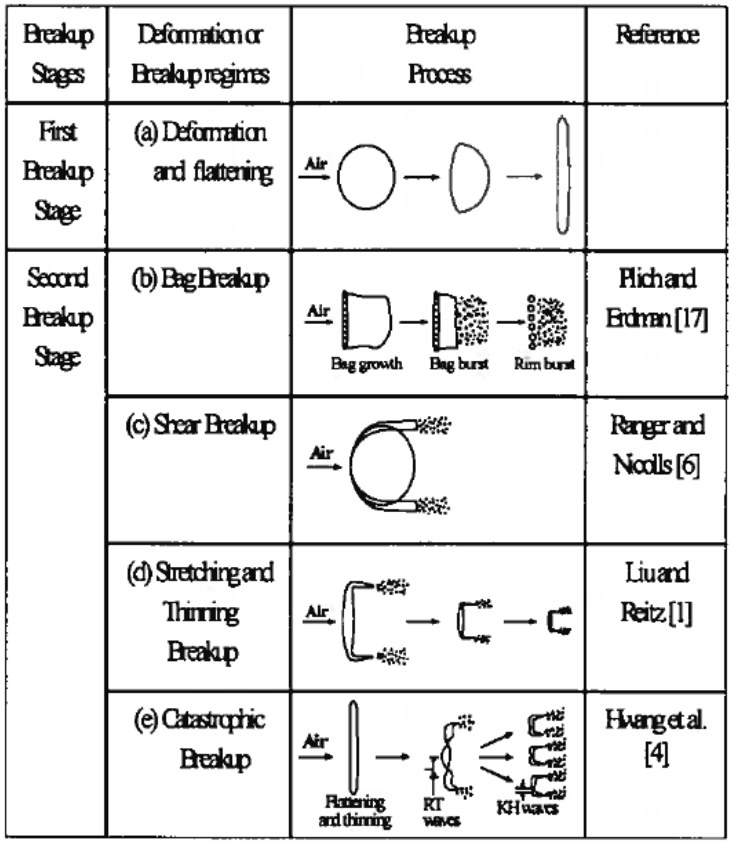

Figure 7. Proposed break up mechanisms. [6]

In the first stage of the droplet brake up, the drop changes shape and is flattened to a disc. The second stage consists of the droplet breaking up from the flattened disc into smaller droplets. Some different break up types can be encountered depending on the circumstances. At the high pressure in a modern diesel fuel injection system the fuel spray is mostly in the direct atomization regime on the right side in Figure 6.

In addition to the mechanical disintegration of the fuel, air entrainment and vaporization are important parts of the fuel injection process. Adam et al. [7] have studied fuel sprays and flames in a rapid compression machine using a nano-spark shadowgraph photography technique. A camera with the shutter opened is located in a completely dark room, the fuel spray is illuminated using a 30 ns spark. Using this technique it is possible to get a picture of liquid phase, vapor phase and droplets. They found that increasing the injection pressure does not increase the spray penetration much. The explanation for this is that the increased fuel spray momentum that follows with an increased injection pressure is consumed in atomizing the fuel. The investigation of Adam et al. ranges from low to intermediate injection pressures and they report that increased fuel injection pressure leads to smaller droplets which promotes vaporization. This may be interpreted as a transition into the higher disintegration regimes as shown in Figure 6.

The process of air entrainment is an important aspect of diesel combustion. Ishikawa and Zhang [8] have studied air entrainment in diesel sprays by measuring air movement around a spray. It is very difficult to measure this type of air movement, one method is to add fine tracer particles to the gas. Ishikawa and Zhang have placed a heated stainless steel wire in the air close to the spray, the hot wire creates density differences in the air which are tracked using a shadow graph method. They compare their measurements with the so called momentum theory by Wakuri et al. [9] which calculates the average air/fuel-ratio in a non burning fuel spray from the distance z to the nozzle:

$$\frac{M_{air}}{M_{fuel}} = \frac{2 \tan \theta}{\sqrt{c}} \sqrt{\frac{\rho_a}{\rho_f}} \frac{z}{d} \tag{3}$$

Where θ is the spray angle, *c* is a contraction coefficient and *d* is the nozzle hole diameter.

The relations found by Ishikawa and Zhang between their measurements and the momentum theory is shown in Figure 8.

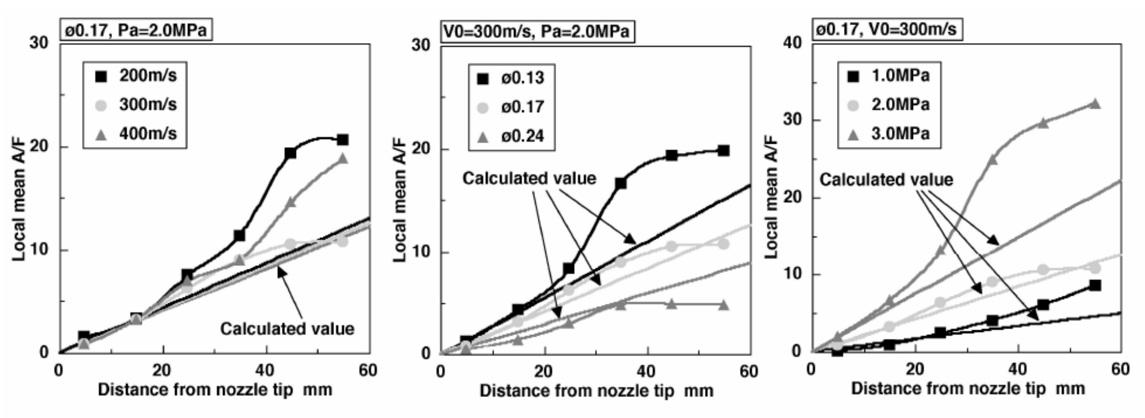

Figure 8. Comparison of measurements with momentum theory. [8]

The graph to the left shows three different initial injection velocities, the middle one three hole diameters and the right one three different back pressures. In some cases the calculation substantially underestimates the

A/F-ratio compared to the measurement after about 20 mm. This is because the momentum theory uses the initial spray angle whereas the real sprays starts to spread more widely due to increased turbulence and vortex generation at the surface of the spray. In some intervals the momentum theory overestimates the A/F-ratio, Ishikawa and Zhang explains this as being caused by spray unsteadiness.

3.2 The cavitation phenomena

If the pressure in a flowing liquid locally falls below the vapor pressure of the liquid, vapor bubbles are formed. This phenomenon is called cavitation and can take place in many types of machines which in some way involve flow with strong pressure gradients, for instance pumps, propellers and diesel fuel injection systems. If excessively strong cavitation occurs the material surrounding the flow may be damaged by cavitation induced erosion. If a cavitation bubble collapses on or near the surface the sudden collapse causes a jet to be formed which strikes the surface and if the energy of the jet is sufficient some material will be eroded off. Once the surface has been pitted by erosion the process may continue at an accelerated rate. The increased surfaces roughness may promote the formation of even more cavitation bubbles and the already weakened surface may be more sensitive to further erosion. If this happens inside the holes of a diesel fuel injection nozzle the resulting change in geometry will influence the fuel injection process negatively. Since the pressure gradient in a diesel nozzle hole can be over 2000 bar/mm of hole length the occurrence of cavitation is basically unavoidable. The cavitation phenomenon is not exclusively negative though. A controlled amount of cavitation will not damage the nozzle and will even have some advantages. Cavitation increases the atomization of the fuel and it can keep the nozzles free from coke deposition which may otherwise interfere with the fuel flow. The commonly used ultrasonic cleaning method does in fact work through cavitation. Desantes et al. [10] have in a very comprehensive way compiled older material in combination with their own work in the field of nozzle flow and cavitation. Some of the theory from this publication is given below.

When fuel flows through the inlet of a nozzle hole a low pressure zone is formed. This causes a recirculation and thus an area reduction, a so called “vena contracta”, see Figure 9.

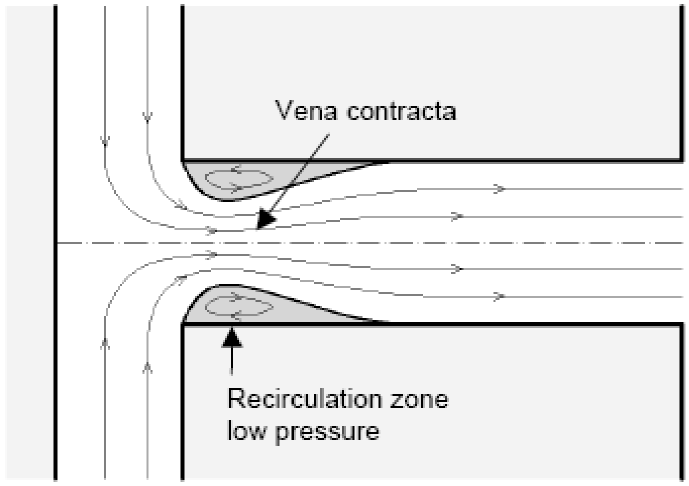

Figure 9. Flow separation in nozzle inlet. [10]

If the separation is not rotationally symmetrical around the circumference of the hole a phenomenon called hydraulic flip can occur as shown by Soteriou et al. [11]. The unsymmetrical boundary layer causes the spray to "bend" away from the direction of the hole, typically downwards since the fuel usually flows from above the inlet forming a separation bubble at the upper corner.

The flux of mass and momentum through the hole can respectively be defined as functions of the velocity u, the density ρ and the flow area A:

$$\dot{m}_f = \int_{A_{geo}} u \cdot \rho \cdot dA \tag{4}$$

$$\dot{M}_f = \int_{A_{geo}} u^2 \cdot \rho \cdot dA \tag{5}$$

The low pressure zone at the inlet causes a turbulent boundary layer to be formed close to the hole walls, see Figure 10. Assuming that the velocity profile of the fuel jet is still uniform for the actual fuel jet, some loss factors can be derived. The assumption means that the flow is regarded as if it was flowing through a hole with a reduced diameter.

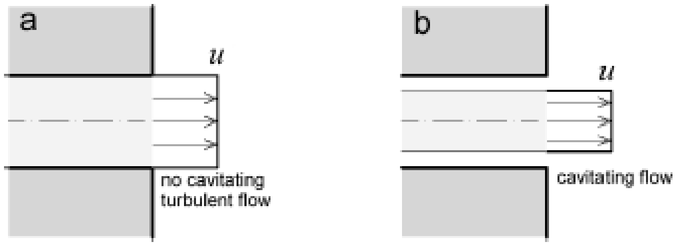

Figure 10. Reduced fuel jet area caused by flow separation. [10]

The assumption that the velocity profile will be fairly uniform is confirmed by flow visualizations in a 2D channel made by Winklhofer et al. [12], see Figure 11.

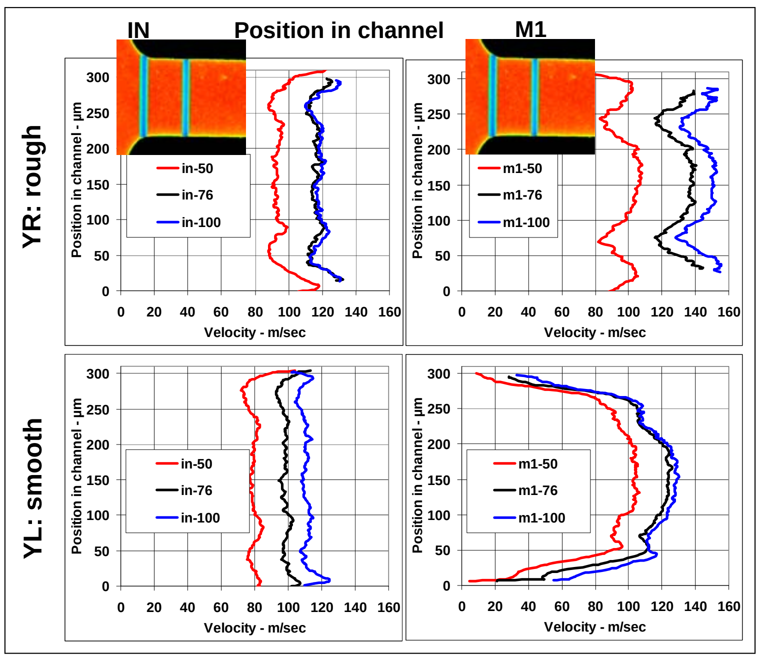

Figure 11. Velocity profiles measured using flow visualization in 2D channel. [12]

The flow velocity profile is characterized at two different positions “IN” and “M1” at 50, 76 and 100 bar pressure drop in one rough and one smooth channel. In the rough channel there is a cavitation layer which leads to a more square velocity profile at the M1 location than in the smooth channel.

A coefficient C_a can be defined to relate the actual hole area occupied by the jet to the full hole area without a boundary layer:

$$C_a = \frac{A \cdot \rho}{A_{geo} \cdot \rho_l} \tag{6}$$

Where the A and ρ are the values for the real case and A_{geo} and ρ_l are the ideal ones without the boundary layer.

The smallest area at the vena contracta is marked as c in Figure 12 according to a definition by Nurick which Desantes et al. refers to.

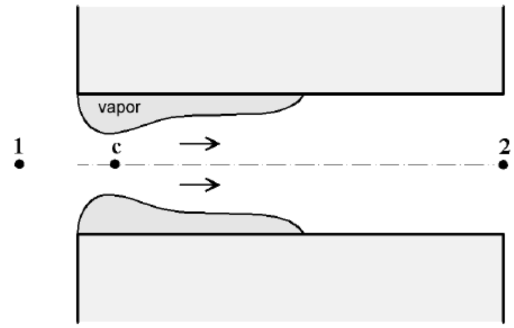

Figure 12. Smallest area according to Nurick´s model. [10]

A contraction coefficient C_c can be defined:

$$C_c = \frac{A_c}{A} \tag{7}$$

Where A_c is the area at c in Figure 12 and A is the full hole area.

The actual or effective velocity through the hole can be defined using the flux of mass and momentum:

$$u_{ef} = \frac{\dot{M}_f}{\dot{m}_f} \tag{8}$$

A theoretical loss free velocity can be derived from Bernoulli's equation:

$$u_{th} = \sqrt{\frac{2\,\Delta P}{\rho_l}} \tag{9}$$

A velocity coefficient C_v can be defined as the fraction between the effective and theoretical velocities:

$$C_v = \frac{u_{ef}}{u_{th}} \tag{10}$$

One measurement of cavitation magnitude is the so called cavitation number. This can be defined in several ways. Desantes et al [10] use a definition by Nurick:

$$K = \frac{P_1 - P_v}{P_1 - P_2} \tag{11}$$

Where P means pressure, the index 1 means hole inlet and 2 means hole outlet, P_v is the vapor pressure of the fuel.

In most publications the cavitation number is called CN and is defined in another way than Nurick´s K. Winklhofer et al. [12] uses this definition:

$$CN = \frac{(P_1 - P_2)}{(P_2 - P_v)} \tag{12}$$

The terminology has been changed to correspond to the ones previously used. With this definition a higher number means more cavitation, the opposite of the Nurick definition.

Winklhofer et al. gives a value of P_v as 20 mbar at 30° C. In a modern diesel engine application P_1 typically ranges from 1000-2500 bar and P_2 from 100-200 bar. Therefore the term P_v may be neglected since it is very close to zero compared to the other pressures. This simplification is used by Argueyrolles et al. [13].

The commonly used discharge coefficient can now be defined as:

$$C_d = C_c \sqrt{K} = C_v \cdot C_a \tag{13}$$

A coefficient for the momentum can be defined as:

$$C_M = C_d \cdot C_v \tag{14}$$

With an increasing amount of cavitation (decreasing K, increasing CN) the C_d will start to decrease at some critical point, see Figure 13.

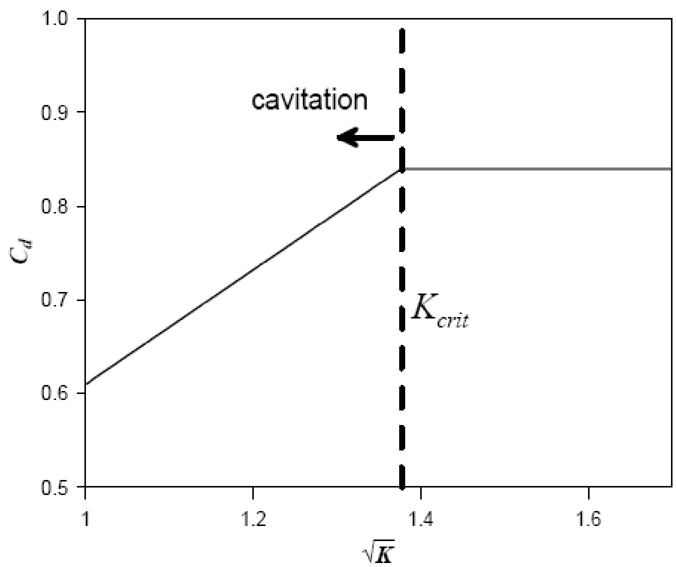

Figure 13. Decrease of C_d on at critical K-number according to Nurick's model. [10]

Desantes et al. calls this a mass flow collapse. This phenomenon is related to the onset of choked flow. Choked flow is a term used for a flow state through a restriction where the downstream pressure no longer influences the velocity. In a diesel engine application this typically occurs at injection pressures higher than approximately 400 bar.

In Figure 14 Desantes et al. summarizes the study by presenting the various factors calculated from experimentally obtained fuel spray data.

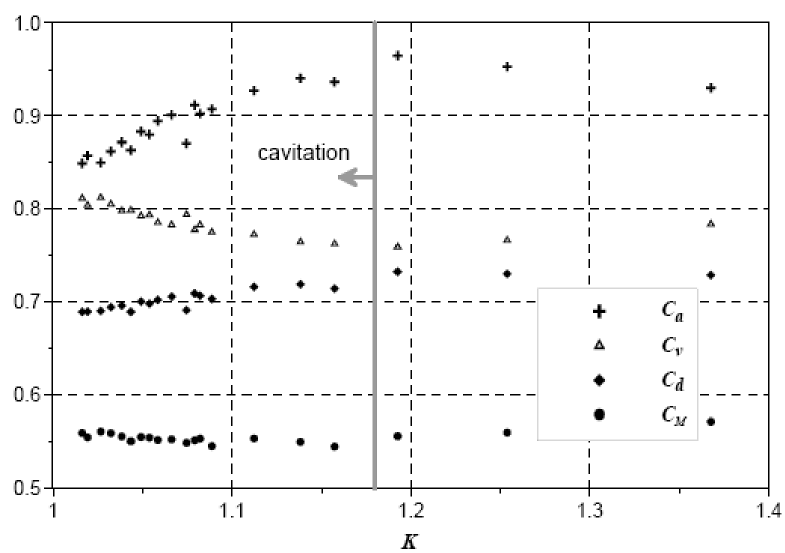

Figure 14. Variation of the four coefficients with cavitation number K. [10]

In Figure 14 cavitation onset starts at values lower than approx. K = 1.18. It is clear that the C_a starts to drop after this point due to the increasingly thick boundary layers that are formed by cavitation and thus reduces the effective area. The C_v increases with increasing cavitation, which might be caused by the area reduction in the hole. C_d is the product beween C_a and C_v, looking at the various points in Figure 12 this seems to be valid for the experiments. It also starts to drop after cavitation onset which is predicted by the theory. *Cm* remains fairly constant, since it is independent of the cavitation number.

3.3 Coke deposition in the nozzle holes

As mentioned before it is possible to use the cavitation phenomena to keep the nozzle holes free from coke deposits. Coke deposition is an important issue that must be kept under control in order to achieve a fuel injection process with good long term performance.

Beside factors like fuel composition and nozzle temperature, cavitation has an influence on how much coke will deposit in the nozzle. Argueyrolles et al. [13] have made an extensive investigation into the relation between factors which influence cavitation and nozzle coking. They also state that nozzle temperatures above 300° C as well as the presence of Zn or Cu in the fuel at as low levels as 1 ppm can cause coking problems. Lubricity additives in the fuel may contribute to metal uptake, especially if the additives are acid. Long term tests are performed using nozzles with different amounts of hydro grinding and hole conicity in order to investigate how these parameters influence the coking. Not only hydro grinding but also a divergent conical hole decreases the amount of cavitation. Figure 15 shows the results of 10 h long coking tests with nozzles with different setups.

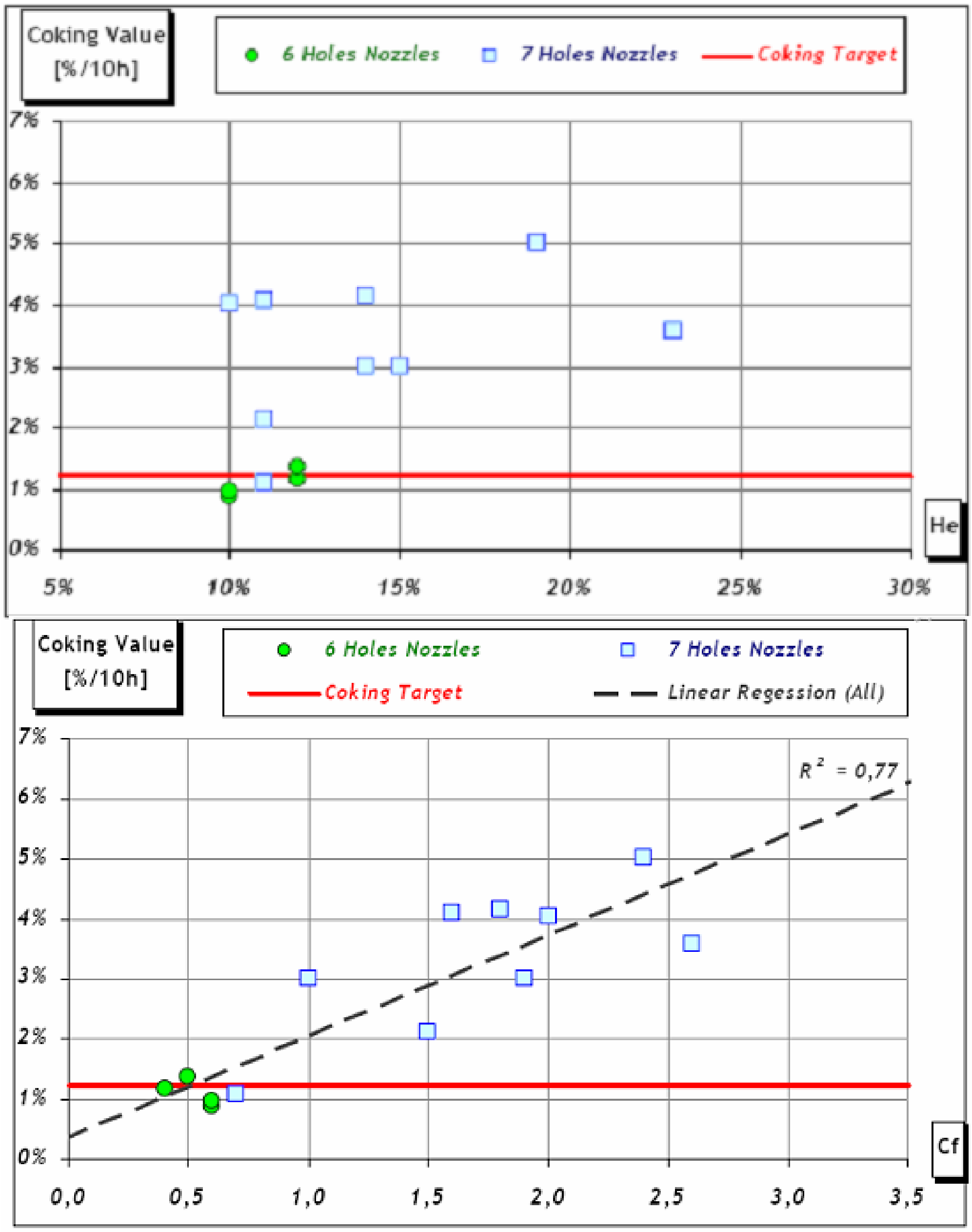

Figure 15. The influence of hydro grinding and conicity on nozzle coking. [13]

The top graph in Figure 15 shows the influence of hydro grinding on nozzle coking. It seems that a low degree of hydro grinding does not always give a low coking value even though a higher degree of hydro grinding seems to increase the coking value. It can thus be concluded that there is some kind of link between the two parameters even though it might not be a direct link. The bottom graph shows the influence of hole conicity on the coking value. Here there is a more clear correlation.

4 Transient diesel engine operation

When a vehicle is used in traffic it has to be possible to quickly change the engine speed and load due to traffic conditions, hills and gear shifts. When the emission levels are certified, driving cycles intended to reflect real traffic conditions are used. One of these cycles is the European Transient Cycle, ETC, shown in Figure 16. The top graph shows the engine speed in percent of maximum and the bottom graph shows the torque in percent of maximum.

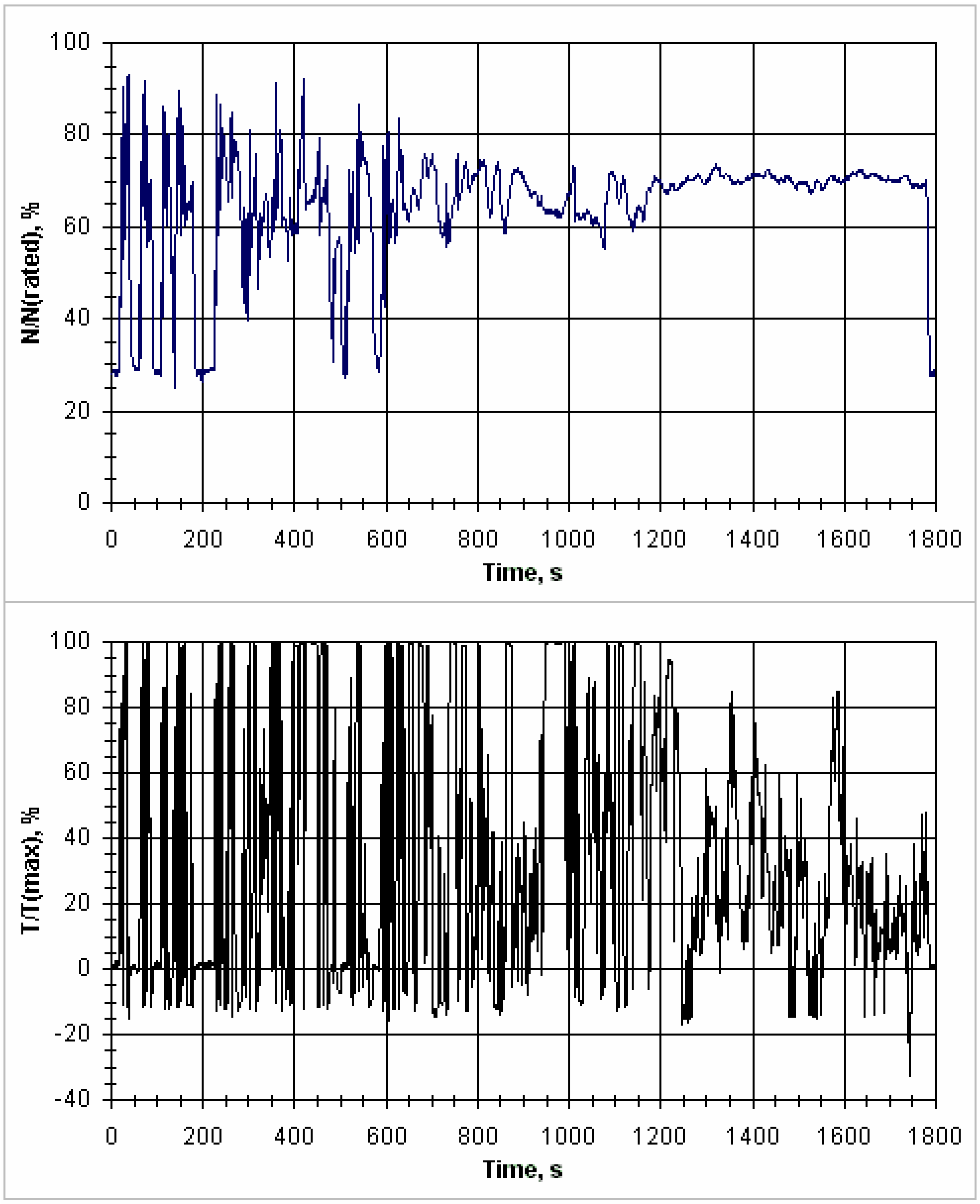

Figure 16. Engine speed and torque as a function of time in the ETC. [14]

It is clear from Figure 16 that the engine has to spend a substantial part of the test cycle under non-steady conditions, so called transients. Several issues arise because of this, most importantly the turbocharger lag in a turbocharged engine. The problem was well formulated by Winterbone et al. [15] in 1977: “When rapid load changes are applied to turbocharged diesel engines they will usually produce black smoke or, in the extreme, stall. This is because the turbocharger is unable to supply sufficient air for complete combustion; the turbocharger has a slower response than the fuel pump.” Even though a lot has happened since then to reduce both the stationary and the transient emissions the basic problem is still the same. Figure 17 shows a schematic layout of a modern diesel engine.

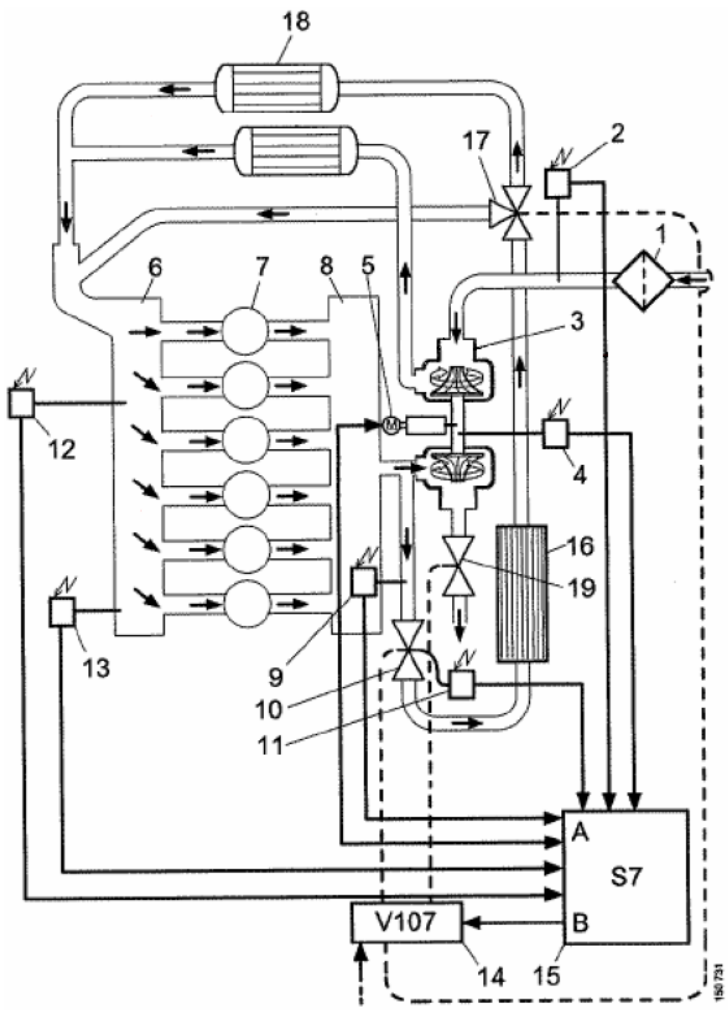

Figure 17. Principal layout of a modern diesel engine.
(Picture from Scania CV AB)

The engine in the Figure is equipped with a common rail fuel injection system which allows a series of multiple injections at high pressure to be made. It also has a Variable Geometry Turbocharger, VGT (3) and a charge air cooler. It has an Exhaust Gas Recirculation system (EGR) with a water cooled EGR-cooler (16) and an air cooled EGR-cooler (18). The latter can be bypassed if necessary. The engine also has an advanced Engine Control Unit (ECU) (15) which controls and monitors the operation of the engine by affecting a number of actuators and by collecting measurement data from a number of sensors.

It is always easier to have control over the engine operation when running at a stationary operating point than when making a transition between two such points, a so called transient. There are several reasons why transient operation is more difficult than stationary operation. One reason is thermal lag, i.e. the fact that it takes some time for the various engine parts to reach thermal equilibrium after a quick load change. Another reason is that the EGR-rate may not be possible to fully control and may vary between the different cylinders. During transients it can also be difficult for the various sensors needed to control the engine to get precise measurements due to problems with averaging and sensor response time. Actuator response can also cause problems.

A transient with increasing load can be divided into three parts. The first part is the load increase to the immediately available torque. If the engine runs at the initial load with a larger λ-margin than necessary the injected mass can instantly be increased accordingly without having to wait for an increased air flow. After this rather short phase comes the turbocharger lag that lasts for a couple of seconds. After the new turbo state is mechanically settled, a few additional seconds is required to allow thermal stabilization. [16] The power of the turbine is directly proportional to the inlet temperature. When a load increase occurs, the exhaust temperature increases but the fact that the exhaust manifold has to be heated to a new equilibrium temperature causes a lag in the temperature increase. [17]

Probably the largest difficulty in the transient operation of a turbocharged engine is the fact that the turbocharger has a lag due to its inertia. When an engine runs at stationary part load there is a certain mass flow though the engine provided by the turbocharger compressor. The air mass flow is sufficient to allow the combustion of a certain amount of fuel. The amount of fuel determines the amount of enthalpy that is available for conversion into mechanical energy by the piston-crank assembly. It also determines the amount of exhaust enthalpy that is available for the turbocharger turbine. When the engine runs at higher load the available exhaust enthalpy is larger and can therefore permit a larger airflow through the engine. The problem arises during the transition from the low load case to the high load case. The turbocharger needs to increase its rotational speed in order to operate with a higher mass flow and since the turbocharger rotor has inertia this takes a certain time. During this so called turbo spool up time the combustion system may have to work under less than optimal conditions, primarily due to decreased λ. Figure 18 illustrates the principle of how λ is influenced in a transient with increasing load in a turbocharged diesel engine.

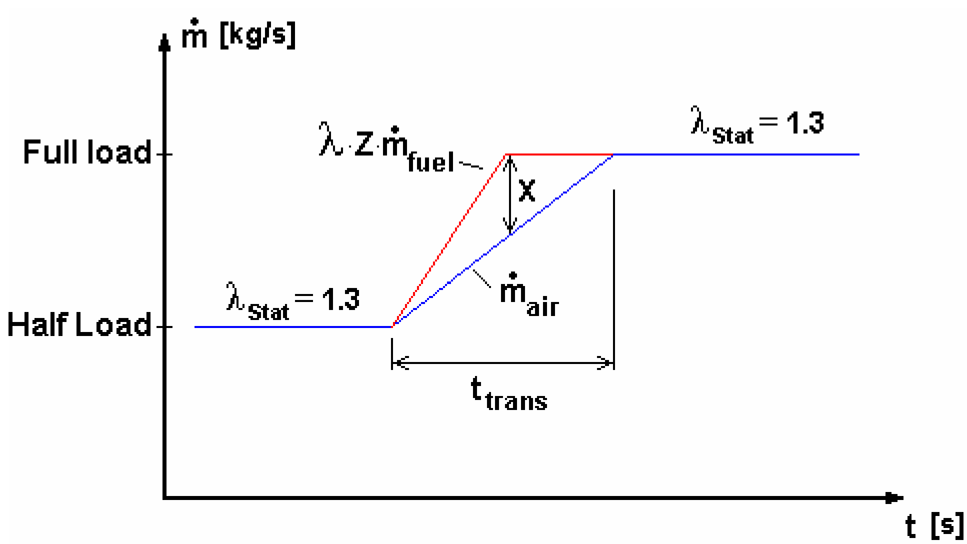

Figure 18. Mass flow during load increase.

Initially the engine operates stationary at half load with a certain λ, for instance λ = 1.3, which allows it to achieve close to 100 % combustion efficiency and low soot emissions. The engine then has to make a transition to full load and it is desired that it runs with the same λ at this point. In the diesel engine unlike an SI engine there is no air throttle to open and thus increase the air flow through the engine. The load step has to start with an increased injected fuel mass. This causes a drop in λ until the turbocharger has spooled up which is illustrated by separation of the air mass flow curve and the fuel flow curve which is scaled with the air-fuel number Z and the λ. A lower transient λ can be defined using the stationary lambda and the distance X between the curves:

$$\lambda_{Trans} = \frac{1}{X+1}\lambda_{Stat} \tag{15}$$

If the fuel flow is increased instantaneously as is basically possible with a diesel fuel injection system the lambda drop will be very severe. If the fuel flow increase is ramped up the lambda drop will be reduced:

$$t \rightarrow \infty \Rightarrow X \rightarrow 0 \Rightarrow \lambda_{\mathrm{Trans}} \rightarrow \lambda_{\mathrm{Stat}}$$

This is a strategy that is actually used to reduce transient exhaust emissions. Unfortunately it is not a very good solution as it slows down the response and drivability of the vehicle. The fact that the fuel flow increase is ramped up also causes the available exhaust enthalpy to be ramped up instead of reaching its full value instantaneously. This further slows down the turbocharger spool up time.

It is clear that a better solution than intentionally slowing down the response of the engine is desirable. There are some solutions which may improve engine transient operation such as usage of elaborate fuel injection-, VGT- and EGR-control strategies.

5 EGR-circuit and turbocharger

As the legislated emission levels have become more stringent one important part of designing engines that comply with the legislation can be the use of Exhaust Gas Recirculation (EGR). The purpose of using EGR in a diesel engine is to add extra gas mass in the combustion chamber in order to limit the maximum flame temperature. A certain amount of burnt fuel mass releases a certain amount of energy in the form of heat. The specific heat capacity of a substance is given in the unit J/kg·K, which represents how many Joules are required to increase the temperature of one kilogram of the substance by one Kelvin. Thus by adding some extra mass of inert gas to the intake air the maximum gas temperature is reduced. Since the formation rates of nitrous oxides depend exponentially on the temperature a small temperature decrease can radically decrease the NO_x-emissions. The temperature lowering effect of EGR is amplified by the fact that the specific heat capacity of exhaust gas is larger than for air. By using EGR-cooling the in cylinder temperatures are further reduced. In order to maintain a certain brake mean effective pressure (BMEP) when increasing the EGR-rate and thus diluting the charge a higher boost pressure is required to feed the same mass of oxygen into the engine.

Serrano et al. [18] have made a study of an EGR-circuit during transient engine operation. They found that the use of EGR can lead to NO_x-reduction of up to 60%, however it also causes an increase in fuel consumption and HC- and smoke-emissions, combined with a rougher engine operation. This places an upper limit on the usable EGR-rate. At steady state an increased boost pressure can reduce these problems but during transient conditions the problem with turbocharger lag makes the solution more complicated.

The EGR is usually diverted from the exhaust stream before the turbocharger turbine, the so called short route or high pressure type circuit. Therefore, the exhaust enthalpy that is rerouted to the EGR-circuit is unavailable for the turbocharger turbine. This is especially serious during a transient with increasing load. One strategy may therefore be to close the EGR-valve, at least in the initial part of the transient. This not only increases the available exhaust enthalpy but it also removes the partial pressure in the gas charge occupied by EGR and leaves more “place” for air. This strategy can of course create problems with high NO_x-emissions during the time that the valve is closed. Another problem which may be encountered during a transient is cylinder to cylinder variations in EGR-ratio which may be increased by quickly opening and closing the EGR-valve.

Arcoumanis et al. [19] have studied the influence of EGR by using an optical engine where flame temperature and in cylinder soot can be estimated using a two color method. They find that an EGR-rate of 50 % reduces the flame core temperature by 100 K and that the soot oxidation is reduced. Normally, there is a strong correlation between KL-factors and measured exhaust soot. This correlation becomes less strong when using EGR.

Variable Geometry Turbocharging (VGT) is used to improve the control of the turbocharging process. An array of variable guide vanes can change the flow area and incidence angle to the turbine. On an engine with high boost pressure and high EGR-rate, VGT is also useful for controlling the engine back pressure in order to create a pressure gradient to drive large amounts of EGR-gas. Like all other engine actuators the operation of the VGT and the EGR-valve has to be controlled precisely and according to a good strategy in order to attain the full benefit of the techniques. This is especially important during transient operation.

6 Real nozzles and application in combustion system

As discussed previously the fuel injection performance is important for low emission combustion. During the last 20 years the maximum fuel injection pressure in the available systems has increased rapidly. Today injection pressures of about 2500 bar are used and in the near future even higher pressures may be available. The development of electronic injection control has led to an increasing controllability of the injection event. Precise control of the fuel pressure, injection phasing and the use of multiple injections has increased the possibilities to influence the combustion process. This increased controllability together with the injection pressure increase is responsible for a large part of the emission reductions that have occurred in diesel engines during the past 20 years.

Current combustion chambers for passenger car and truck diesel engines typically utilize direct injection system with a fairly shallow piston bowl and a central fuel injector with 5 – 8 holes. The injectors can either be unit injectors with one cam driven pump element per injector or of the common rail type with a crankshaft driven pump that feeds the injectors through a high pressure fuel accumulator, the fuel rail. A multi-hole nozzle can be classified into one of two basic types depending on the design of the needle seat and nozzle hole inlets. Figure 19 from Roth et al. [20] shows a needle sac type on the left and a VCO (Valve Covered Orifice) on the right.

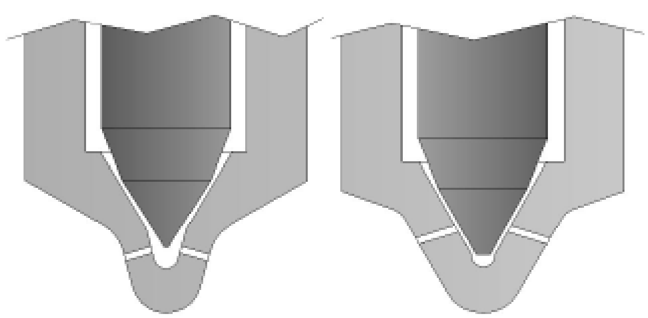

Figure 19. Sac- and VCO-type fuel injection nozzles. [20]

The advantage of the VCO-type is that the needle covers the holes when it is closed. Therefore, the fuel shuts off abruptly at the end of the fuel injection and there is no sac volume which boils off or flows at low pressure as in the sac-type. The small fuel portion which is injected at low pressure at the end of injection may cause increased smoke emissions. The VCO on the other hand has the disadvantage of being more sensitive to needle misalignment. The fuel passes through the very small gap between the needle and nozzle body and the slightest misalignment gives rise to variations in the flow area and thus the fuel flow for the different holes. In order to achieve a well functioning combustion system that comply with strict steady state and transient emission norms it is important to keep tight control of nozzle hole to hole variations.

6.1 Hole to hole variations

The term hole to hole variations is used to describe differences in the amount or rate for parameters such as mass flow, impulse, spray cone angle and penetration. The issue of hole to hole variations is generally recognized as playing an important part in the combustion and emission formation processes. Some publications have been made on the subject but the influence of hole to hole variations is very complex and there is still little understanding of the mechanisms involved. The holes in a modern injection system are very small, typically 50 – 250 µm, and they are manufactured using a complicated EDM (Electro Discharge Machining) process. This makes it very difficult to keep the geometrical properties of the holes within certain specifications. Due to the complexity of the spray formation and combustion process it is in fact very difficult even to know how tight the specifications should be. As discussed previously not only the diameter and conicity of the holes have an influence on the flow parameters in the holes, but also the inlet rounding has a major impact and the inlet rounding is quite difficult to measure.

Hole to hole variations may have several causes. They can be the result of poor manufacturing quality, erosive damage or varying amounts of coke deposits in the nozzle holes. The latter two may be caused by differences in the hole inlets which affects the cavitation and thus the erosion and coking as discussed previously. Hole to hole variations may also appear in nozzles made for angled installation in 2-valve engines as demonstrated by Kull and Krüger [21]. The angled tip results in different inclination angle and thus a different flow path for each of the holes, see Figure 20.

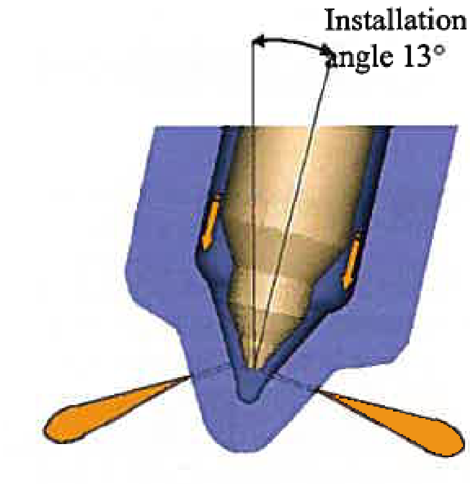

Fig 20. VCO nozzle for angled installation in a 2-valve engine. [21]

Kilic et al. [22] demonstrate the influence that hole inclination can have on the cavitation and thus the flow. Figure 21 shows the result of a simulation of nozzle hole flows for two inclination angles.

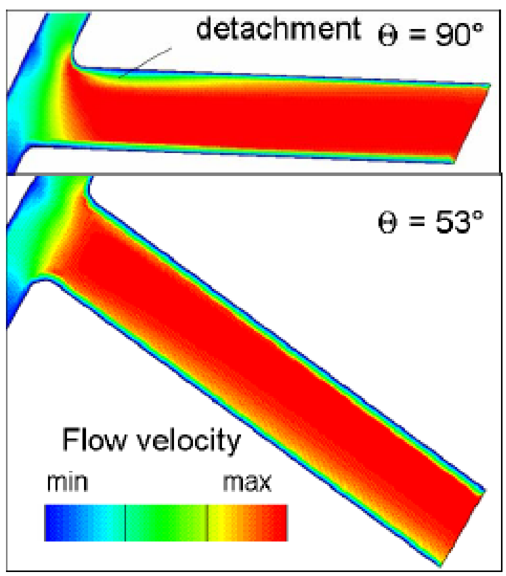

Fig 21. Simulation of nozzle flow for different inclination angles. [22]

Figure 21 shows that a larger inclination angle leads to more detachment at the sharper upper inlet corner. This flow behavior has an influence on hole to hole variations in inclined installation nozzles. Kull and Krüger have measured the mass flow, momentum, and fuel spray penetration of the inclined VCO nozzle shown in Figure 20. Figure 22 shows how all the parameters vary in a "periodical" way corresponding to the variations in inclination angle.

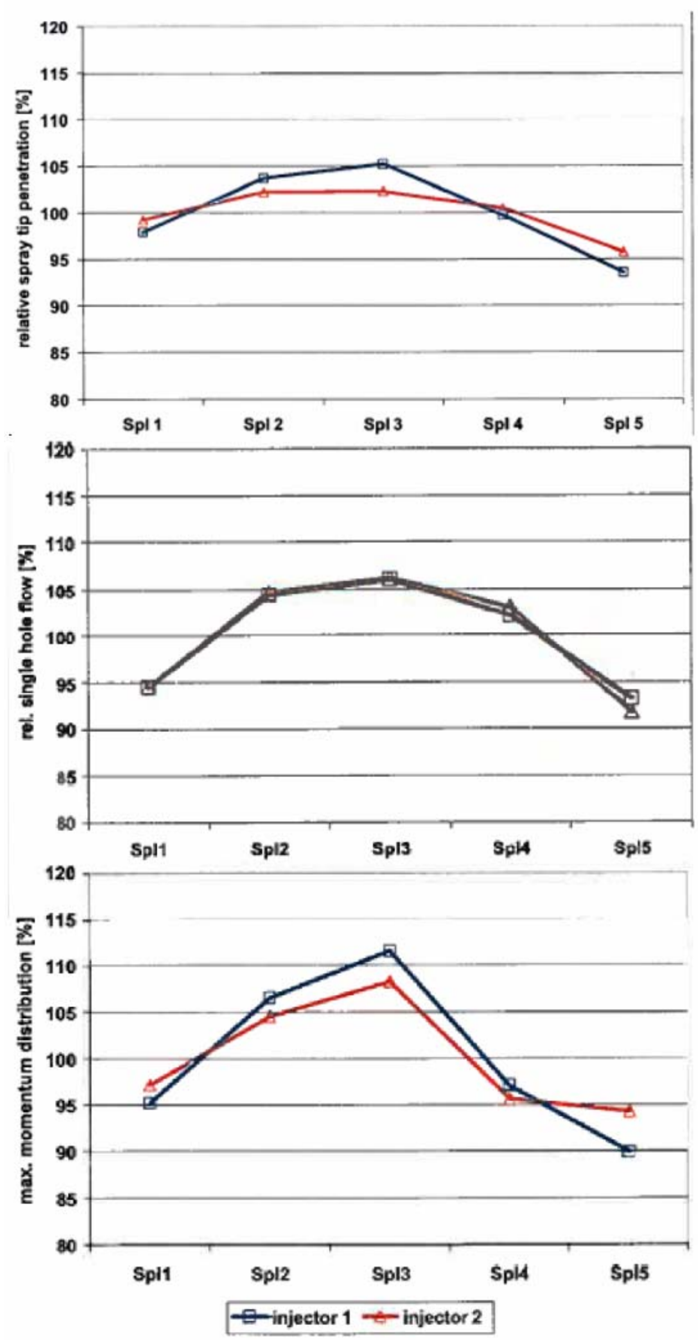

Figure 22. Hole to hole distribution of penetration, mass flow and momentum. [21]

Hole 1 and 5 have an inclination angle of 83°, hole 2 and 4 have an angle of 66°, and hole 3 has an angle of 56°. It is clear from Figure 22 that the penetration, mass flow and momentum decreases with increasing inclination angle which is also the conclusion of Kilic et al. [22].

Needle misalignment is another issue that affects fuel injection nozzles. It is primarily a concern for VCO nozzles but it may also have an influence on sac type nozzles. Figure 23 from Kilic et al. [22] shows how the flow velocity is influenced by needle mismatch.

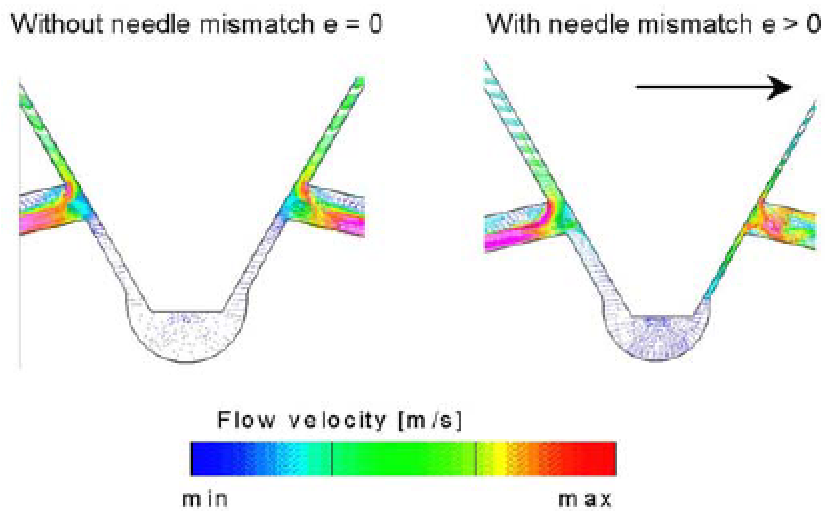

Fig 23. Flow velocity in VCO nozzles with and without needle mismatch. [22]

In Figure 23 it is obvious why needle mismatch is more serious in a VCO nozzle than in a sac type nozzle. In the VCO nozzle the mismatch directly influences the distance between the needle and hole inlet. In a sac type nozzle the influence is more indirect as the hole inlets are located in the nozzle sac.

The influence of needle mismatch depends on how large the needle lift is. A smaller lift means a bigger impact on the geometry. Kilic et al. have illustrated this by calculating the spray hole area and needle seat area as a function of needle lift for two hole sizes, see Figure 24.

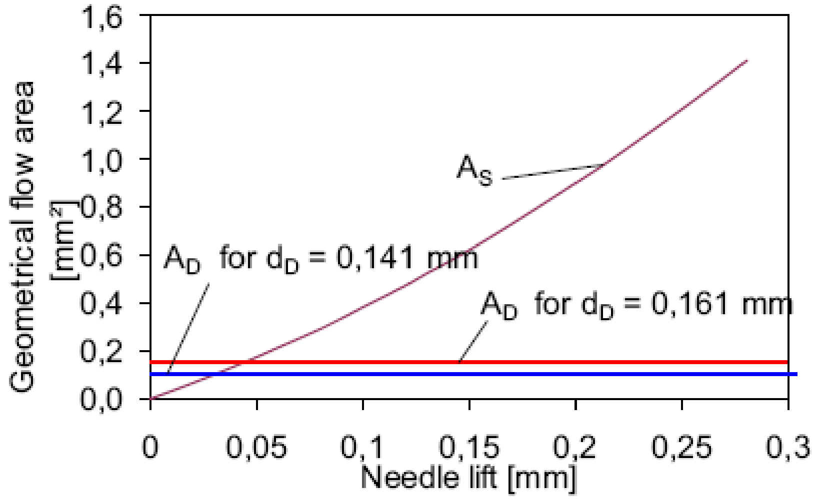

Figure 24. Nozzle seat area (A_S) and spray hole area (A_D) as a function of needle lift. [22]

For small lifts, below 0.05 mm, the needle seat area is smaller than the hole area. Therefore, the misdistribution of needle seat area that the needle mismatch leads to has a large influence on the flow. As the lift increases and the needle seat area becomes much larger than the hole area and the influence of needle mismatch becomes much smaller since it is the hole area that mainly determines the flow. Figure 25 illustrates the total effective nozzle bore area (calculated from a mass flow equation) as a function of needle lift for two hole diameters. This area refers to the area which is limiting the flow, in the beginning it is the area between the needle and the needle seat. When the needle has lifted beyond a certain point the hole area is limiting.

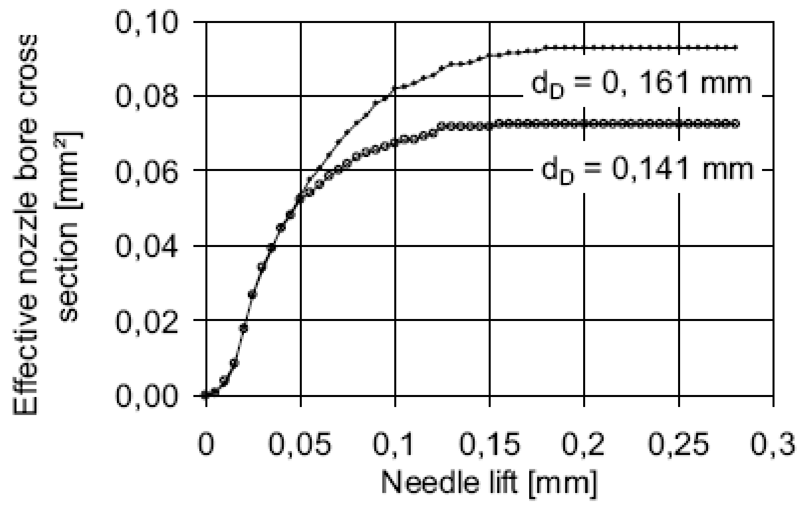

Figure 25. Effective nozzle bore cross section as a function of needle lift. [22]

Because of the major influence of the nozzle seat area at lifts below 0.05 mm there is basically no influence from the different hole sizes up to that point. After 0.05 mm the curves are different and converge to their final value at about 0.15 mm after which the increasing needle lift does not have much influence on the effective nozzle bore cross section. It can thus be concluded that needle mismatch, at least for a needle with relatively large lift, mainly influences hole to hole variations during opening and closing of the needle and not for the fully developed flow. Figure 26 shows an example of how this can cause the initial penetration of the fuel sprays from a nozzle to become very different.

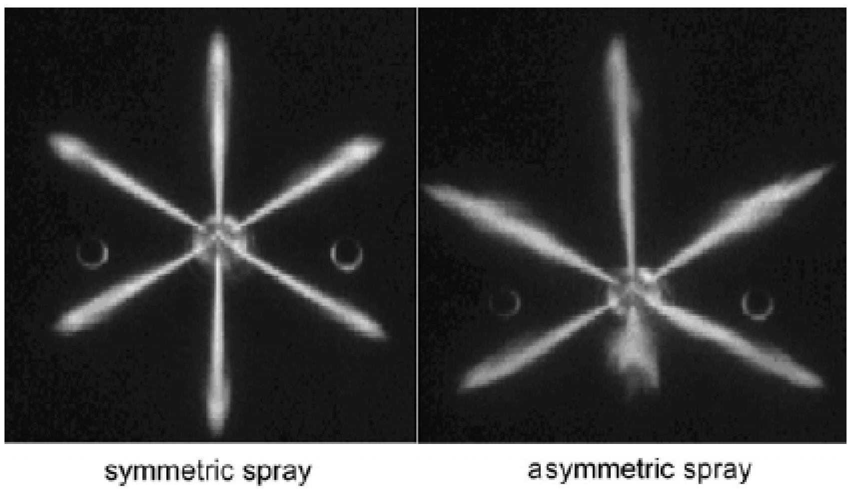

Figure 26. Nozzles with symmetric (left) and asymmetric (right) fuel sprays. [22]

De Risi et al. [23] find similar results regarding initial spray propagation. They compare spray images from some VCO nozzles with different design, a mini sac nozzle and a mini sac nozzle where the sac volume is reduced by 30 %. They report that the sac nozzles have far less initial hole to hole variation and that using a double guided needle in a VCO nozzles as opposed to the “standard” layout with a single needle guide also reduces hole to hole variations as this reduces the mismatch. De Risi et al. also used a microscopic technique to study the hole outlets. Even though deformed hole edges, variations in outlet sizes and eccentricities were found, these geometrical deviances did not correlate with the spray data. This may be because the outlet properties are not as important as for instance the inlet properties. Even though many have studied the phenomenon, it is not clear precisely what influence an initial asymmetric spray distribution such as the one in Figure 26 has on the combustion and emission formation processes.

One way of characterizing actual hole geometries is presented by Payri et al. in [24], where a silicon mould is made of the nozzle, see Figure 27.

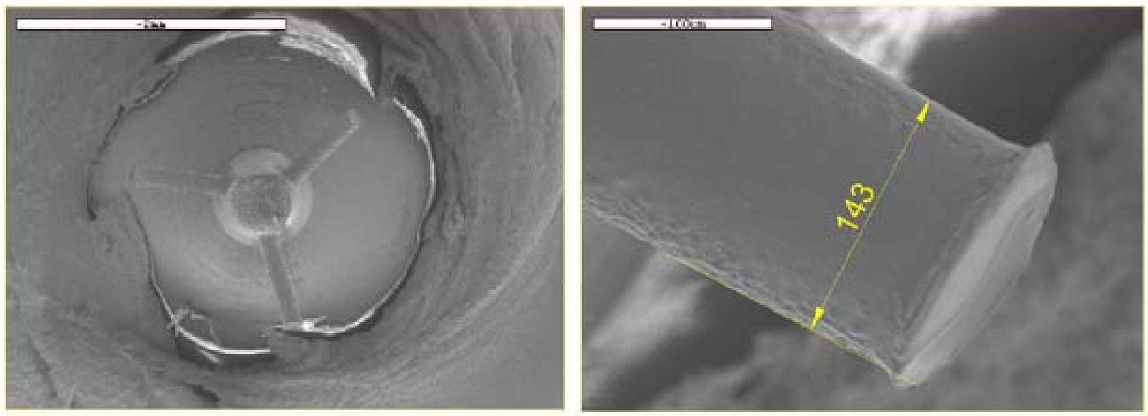

Figure 27. Silicon mould of three hole nozzle. [24]

The mould is then scanned by an electron microscope, the geometries can be loaded into a CAD software. Payri et al. compare the geometries obtained in this way with the results from three other measurement methods, mass flow measurements, spray momentum and spray penetration. Figure 28 shows how the various parameters correlate to each other. The parameters are normalized so that 1 means average for the three holes.

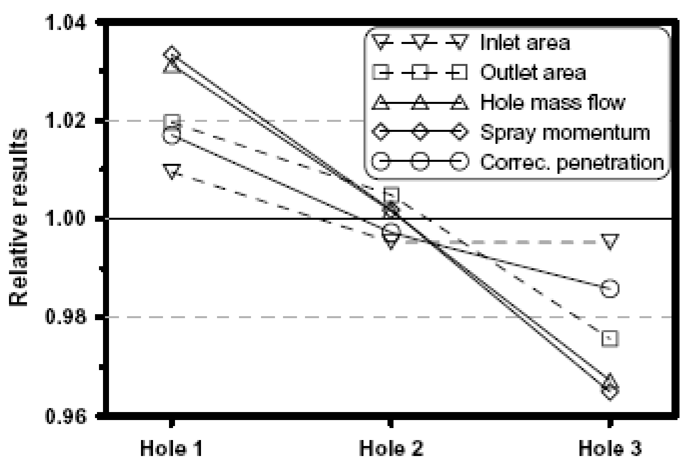

Figure 28. Comparison of results from the different measurement techniques. [24]

The studied parameters are clearly following the same trend, however these single three hole nozzle does not really provide much statistical material. The variations are also quite small.

Previously it was discussed that hole geometries influence the intensity of the cavitation and that cavitation can lead to erosive damage to the hole and thus influence the geometry. One can imagine that under certain circumstances a feedback loop may occur promoting increasing cavitation and increasing hole damage. It is very difficult to experimentally study such a phenomenon since it requires that nozzles that will behave accordingly are identified and geometrically characterized before they are used. Greif et al. [25] attempt to study the “feedback” using a simulation technique. First a flow simulation is made. Erosion damage is assumed to occur in areas with cavitation over a certain threshold. A new grid is created by adding the expected damage to the hole and a new flow simulation is made. Figure 29 shows the progression of the erosion.

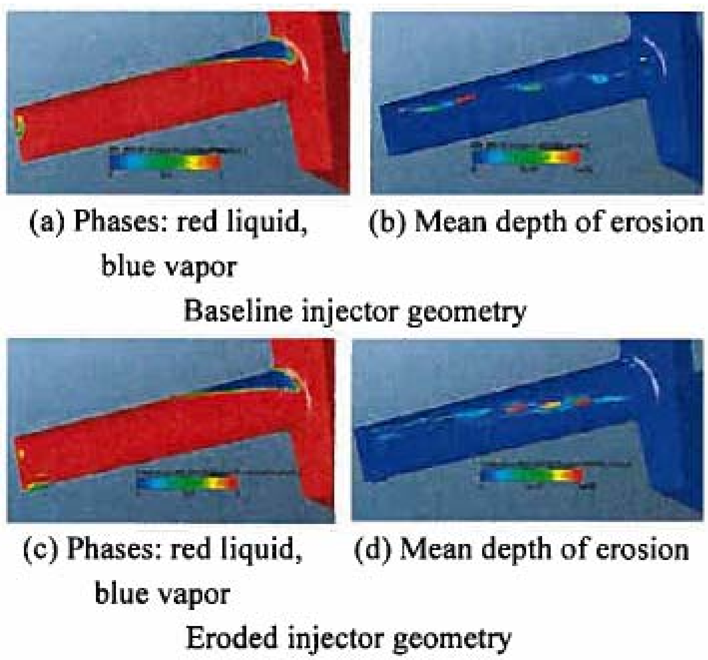

Figure 29. Progression of cavitation erosion. [25]

The simulations by Greif et al. show how a feedback between cavitation and erosion is likely to progress. This mechanism explains how small initial variations in for instance inlet radii can result in large variations in hole taper angle and diameter. Greif et al. predict that this type of erosive hole damage would result in fuel sprays with less penetration and a larger cone angle.

7 Results and discussion

Paper I deals with fuel spray impulse measurements. The so called fuel spray impingement method is a widely used method for measuring the impulse of a diesel fuel spray. The paper deals with the theoretical background to the function of such a device and goes on to describe the development of an impingement sensor. Normally, in publications dealing with this type of equipment both theoretical background as well as the description of the impingement sensor is limited. With this publication it is possible to easily construct a well functioning impingement sensor. The influence of temperature related effects are studied and a solution in the form of a sensor strike plate is presented. The issue of strike plate material strength is investigated by testing plates made of different materials and it is shown that spray induced plate deformation negatively influences the accuracy of the measurements. This is caused by the fact that the impingement method is sensitive to the lengthwise velocity component of the exiting fuel and this in turn is influenced by spray induced plate deformation. A new concept for accuracy improvement is introduced, a plate with a rotationally symmetrical curvature which guides the flow to a controlled exit direction. It is found that the commonly used flat strike plate causes an overestimation of the fuel impulse.

In paper II the issue of hole to hole variations and their influence on emissions are addressed. A set of six fuel injectors were found to give large differences in smoke emissions and fuel consumption after a 600 h running period. It was suspected that hole to hole variations could be the cause of this. This set of fuel injectors thus provided a valuable opportunity to investigate hole to hole variations for different parameters and how this influences soot emissions and fuel consumptions. Little is known about this important issue. The individual soot emissions and fuel consumption of the six injectors were measured in a single cylinder engine. A number of measurement techniques were used to characterize hole parameters for all of the eight holes on the six injectors. The mass flow was measured using a rig which collects the fuel from the individual holes. The fuel spray impulse was measured using the impingement technique. The fuel sprays were studied in a pressure vessel using a high speed camera and image analysis software. The hole geometries were measured using a computer tomography machine. All these parameters were compared to each other and to the soot emissions. Firstly, it was found that the hole to hole variations were very large, the difference between the hole with the highest and the lowest impulse was 25 % on some injectors. The difference in mass flow could be equally large or even larger. In well functioning fuel injectors, the difference can be within a few percent. Also, correlations were found between the magnitude of hole to hole variations and the soot emissions for the injectors. Especially variations in mass and impulse had a clear influence on emissions. The fuel spray study using high speed photography in the pressure vessel resulted in a number of time resolved parameters. The parameters that were considered most useful to include in the study were the penetration and the cone angle, some other parameters were available but they were either to inaccurate or they showed no variation between the different holes. By using a computer tomography machine very precise 3D-geometries of the nozzle holes were obtained. With these geometries it is possible to fit cylinders or truncated cones to the data points. It is thus possible to get an average hole diameter, hole taper angle and to obtain precise data on the actual direction of the hole in all axes. As the locations and directions of the holes, just like for the fuel sprays, were found to be very precisely spaced they were not included in the study. All the holes are more or less conical with the larger diameter at the outlet, this is a likely result of erosion damage. When all the measured parameters are plotted against each other the following pattern emerges: The mass flow, impulse and penetration increase together and show an inverted correlation to the hole taper angle, spray cone angle and hole diameter. These relation except the inverted influence by the hole diameter are what was expected from basic flow relations. One possible explanation may be that the quite small diameter difference between the holes does not have as large influence as a decreased discharge coefficient in the larger holes. If the holes are damaged by erosion the larger the hole is the more eroded it is and the smaller the discharge coefficient may be. The result from Greif et al. [25] where hole erosion leads to less penetration and higher cone angle seems to be verified.

8 Conclusions

Many factors are involved when the fuel interacts with the gas charge in a diesel combustion chamber. The mechanics and thermodynamics involved in droplet break up, fuel vaporization, combustion, NO_x-formation, soot formation and oxidation are not yet fully understood. The secret behind a high efficiency, low emission diesel combustion process is to find a “way” through the combustion process where the fuel is completely burnt without high soot emissions while avoiding excessively high temperatures. High temperature in combination with residence time promotes formation of nitrous oxides. The profile of the heat release also has to be suited for the particular engine in order to result in high efficiency. From the references and the included publications it is clear that a high performance diesel combustion process is the result of many precisely tuned parameters, thus it is very sensitive to disturbances. This explains why hole to hole variations which cause a 25 % difference in fuel mass flow between the sprays can cause smoke emission to increase by more than a factor three. When some of the fuel suddenly ends up in the wrong place, the combustion and emission formation process proceeds along another path and the result is a drastic emission increase.

Cavitation is an important phenomenon in a diesel fuel injection system. Its effects are both positive and negative. On the positive side, the cavitation promotes fuel atomization and can help to keep the nozzles free from coke deposits. On the negative side, an excessive amount of cavitation can cause erosive damage to the nozzle holes and thus deteriorate the long term performance of the fuel injection process. The process can also end up in a negative feed back loop where more erosion leads to more cavitation and vice versa. The amount of cavitation in the holes thus have to be set at an appropriate level by designing the holes with a certain inlet rounding and hole conicity. This may not be easy as the fuel system has to work in a variety of operating conditions with different injection- and backpressures. The nozzle holes are manufactured using complicated machining methods so one must pay close attention to the production quality of this very important part of the combustion system. Especially as the holes and inlet radii are very small and thus difficult to measure using any method which can be readily deployed on a production line.

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