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Influence of nozzle hole diameter on the first and second law balance in a DI Diesel engine

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Abstract

In the present work, the influence of nozzle hole diameter is studied on the first and second law balance in a DI Diesel engine. To this end, first law analysis is done by using the results of a three dimensional CFD model. The results show good agreement with the previous experimental data. For second law analysis, a computational code developed in house is applied. Behaviors of the results have good accordance with the literature. The results show that an increase in nozzle hole diameter increases both the indicated work and heat loss to walls. Also with regard to second law terms, the results show that an increase in nozzle hole diameter leads to an increase in indicated work availability, heat loss availability, and entropy generation per cycle and a decrease in combustion irreversibility and exhaust gas availability.

Keywords: Diesel engine, Nozzle hole diameter, First and second law analysis, Availability, Irreversibility

1. Introduction

The compression ignition (Diesel) engine is now the preferred prime mover in many applications, owing to its reliability and excellent fuel efficiency [1]. Diesel engine simulation is a very effective tool to approximate and improve engine performance. First law analysis is a good method for predicting engine performance and the influence of its various operative parameters [2–4]. On the other hand it has long been understood that traditional first law analysis, which is needed for modeling engine processes, often fails to give the engineer the best insight into the engine's operation. In order to analyze engine performance, that is to evaluate the inefficiencies associated with the various processes, second law analysis must be applied [5, 6]. For second law analysis, the key concept is "availability or exergy".

The availability of a matter refers its potential to produce useful work. Unlike energy, availability can be destroyed, which is a result of some phenomena such as combustion, friction, mixing, throttling etc. The destruction of availability is a source for insufficient use of fuel availability to produce useful mechanical work in an internal combustion engine. The reduction of irreversibilities can lead to better engine performance through a more efficient exploitation of fuel [1, 7]. Many studies have been published in the past few decades (the majority in the last 20 years) concerning the application of second law to internal combustion engines.

The first studies in to the operation of internal combustion engines that included exergy balance in the

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calculations were done around 1960, the works of Traupel [8] and Patterson and Van Wylen [9]. Flynn et al [10], explained a new observation in internal combustion engine studies. They applied a method based on the second law of thermodynamics for engine analysis. A review paper written by Caton [11] explains previous studies into engine operation from the perspective of second law observation. One of the most important sections in Diesel engines is the injection system. Injection system characteristics such as injection pressure, number of nozzle hole and nozzle holes diameter directly affect on combustion process and thereby directly impact the engine performance. In the case of some processes dependent on the injection system such as mixture formation, the numerical simulation is becoming increasingly important. One advantage of using simulation models is that, in contrast to experiments, results can often be achieved faster and more cheaply. More importantly still despite the higher uncertainty compared to experiments, the numerical simulation of mixture formation and combustion processes can give much more extensive information about complex in-cylinder processes than experiments could ever provide. Using numerical simulations, it is possible to calculate the temporal behavior of every variable of interest at any place inside the computational domain. This makes it possible to obtain detailed knowledge of the relevant processes and is a prerequisite for improvement. Furthermore, numerical simulation can be used to investigate processes that take place on time and length scales or in places that are not accessible and thus cannot be investigated using experimental techniques. In the case of high-pressure diesel injection for example, the spray break-up near the nozzle is mainly influenced by the flow conditions inside the injection holes. Due to the small hole diameters (less than 200 μm for passenger cars) and high flow velocities (about 600 m/s and more), the three-dimensional turbulent and cavitating two-phase flow is not accessible by measurement techniques. One very costly and time-consuming possibility of gaining insight into these processes is to manufacture a glass nozzle in real-size geometry and use laser-optical measurement techniques. Outside the nozzle, in the very dense spray measurements of the three-dimensional spray structure (droplet sizes, velocities etc.) become even more complicated, because the dense spray does not allow sufficient optical access of the inner spray core. In these and other similar cases numerical simulations can give valuable information and help to improve and optimize the processes of interest [12]. A. Abassi and et al. [13] explained the influence of the injection system characteristics on first and second law terms in high speed DI Diesel engines. They carried out an overall study, not going into detail. Most previous investigations relevant to nozzle hole (nozzle hole diameter and number of nozzle holes) restrict themselves to focusing on first law analysis and emission formation. For example, set against this background, Akira Numata and Yashinori Nagae [14] studied the simultaneous effect of nozzle hole diameter and injection pressure on engine performance and emissions to increase thermal efficiency and reduce NO_x emission in a stationary Diesel engine. Byong Seok Kim et al. [15] explained the effect of the injector nozzle hole diameter and number of holes on combustion performance in a medium speed marine Diesel engine. They applied an experimental scheme and a numerical simulation to investigate NO_x formation and fuel consumption according to various fuel injection nozzle geometries in medium speed marine Diesel engines. Gao Jian et al [16] worked on the effect of injection pressure and nozzle hole diameter on mixture properties of DI Diesel spray. They applied the laser absorption scattering (LAS) technique to quantitatively and simultaneously measure the concentration distributions of both the liquid and vapor phases in the fuel spray injected by the common rail hole-type injector for DI Diesel engines. N.Tamaki et al. [17] utilized an experimental scheme and explained the influence of nozzle hole diameter, number of nozzle holes and pitch circle diameter of nozzle holes on atomization of spray and break up length to invent high efficiency atomization enhancement nozzle. However, it is exploring the effect of nozzle parameters on first and second law terms together that gives the engineer the best insight into the engine's operation and thereby leads to the best engine design.

The present work attempts to analyze the influence of nozzle hole diameter on first and second law balance in a DI Diesel engine. First law analysis is done by using the results of a three dimensional AVL Fire software. For second law analysis, a computational code developed in house is applied. Second law terms such as burnt fuel availability, indicated work availability, availability loss associated with heat transfer to walls (heat loss availability), combustion irreversibility, exhaust gas availability and entropy generation per cycle are evaluated with this code.

Eight nozzle hole diameter are studied and their results are compared to gauge how first and second law terms change when increasing or decreasing the nozzle hole diameter.

The results of the analysis are given in a series of diagrams, which plot interesting first and second law values.

2. Model description

In the present study, Diesel engine operation is analyzed from perspective of the first and second laws of thermodynamics and the various terms of the first and second laws are evaluated.

2.1. First law analysis

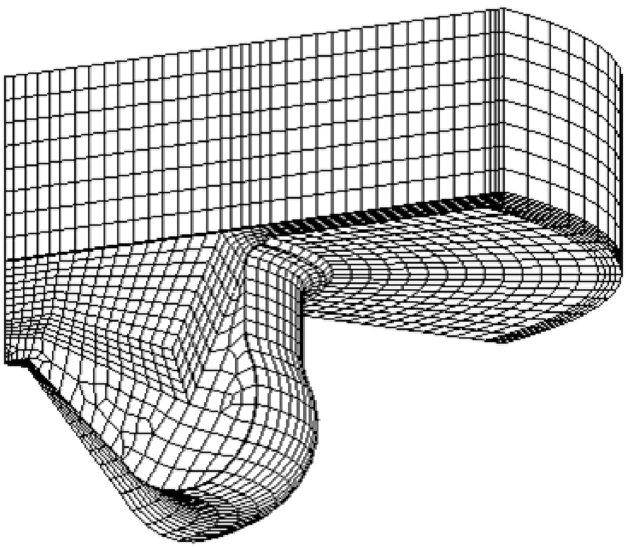

Figure 1: 3D moving mesh at 40° BTDC

In this study, first law analysis is done by using AVL Fire software which uses a 3D moving mesh to simulate combustion chamber. A scheme of 3D moving mesh at 40° CAD before TDC is shown in Fig. 1.

With the applied code, the compressible, turbulent, three dimensional transient conservation equations are solved for reacting multi-component gas mixtures with the flow dynamics of an evaporating liquid spray by Amsden A. A et al, [18]. The turbulent flows within the combustion chamber are simulated using the RNG $\kappa - \varepsilon$ turbulence model which is presented by Han, Z and Reitz, R. D [19], modified for variable-density engine flows. The combustion process is modeled by the Eddy Breakup model. The Eddy Breakup model was developed as a combustion model with active turbulence controlled combustion model based on the Magnussen [20] formulation. This model assumes that in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from eddies containing hot combustion products. The rate of dissipation of these eddies determines the rate of combustion as follows:

$$\overline{\rho \dot{r}_{fu}} = \frac{C_{fu}}{\tau_R} \overline{\rho} \min\left(\overline{y}_{fu}, \frac{\overline{y}_{ox}}{S}, \frac{C_{pr} \cdot \overline{y}_{pr}}{1+S}\right) \tag{1}$$

The first two terms of the "minimum value of" operator determine whether fuel or oxygen is present in limiting quantity, and the third term is a reaction probability which ensures that the flame is not spread in the absence of hot products. The above equation includes three constant coefficients (C_{fu}, τ_R, C_{pr}) and C_{fu} ranges from 3 to 25 in diesel engines. An optimum value was selected according to the experimental data.

Spray breakup and droplet distribution is modeled by the advanced Wave Standard, developed by Reitz [21]. In this model the growth of an initial perturbation on a liquid surface is linked to its wave length and to other physical and dynamic parameters of the injected fuel and the domain fluid. Drop parcels are injected with a characteristic size equal to the nozzle exit diameter (blob injection). The Dukowicz model [22] is used for treating the heat-up and evaporation of the droplets.

2.2. Effect of nozzle hole diameter on the combustion process

When fuel is injected into the combustion chamber, the droplets of fuel enter into direct contact with the

gas which is in the combustion chamber. The interaction between the droplets and the gas creates and conducts the combustion process. Droplet size which is directly related to the nozzle hole diameter, plays a significant role in interaction between the droplets and the in cylinder gas. Small droplets lead to better mixture formation and reduce ignition delay and vice versa. On the other hand, to produce small size droplets a small nozzle hole diameter is required. However, reducing the diameter of nozzle holes (when injection pressure is kept constant) causes the injecting fuel mass flow rate to decrease and injection duration to increase, which reduces the intensity of the combustion process and vice versa. Therefore, as regards the combustion process, reducing nozzle hole diameter has a desired effect (to produce better mixture formation and to decrease ignition delay) and an undesired effect (to reduce the intensity of the combustion process).
The differential equations for the momentum and evaporation of spray particles are as follows:

2.2.1. Momentum
Momentum equation of a droplet is given as follow:

$$m_d \frac{du_d}{dt} = F_{dr} + F_g + F_P + F_b \tag{2}$$

Where F_{dr} is the drag force, given by:

$$F_{dr} = D_P \cdot u_{rel} \tag{3}$$

D_P is the drag function, defined as:

$$D_P = \frac{1}{2} \rho_g \frac{\pi \cdot D_d^2}{4} C_D \left| u_{rel} \right| \tag{4}$$

C_D is the drag coefficient which generally is a function of the droplet Reynolds number Re_d.
From the various formulations in the literature for the drag coefficient of a single sphere, FIRE uses the following formulation from Schiller and Naumann [23]:

$$C_D = \begin{cases} \frac{24}{Re_d}(1 + 0.15\, Re_d^{0.687}) & Re_d < 10^3 \\ 0.44 & Re_d \geq 10^3 \end{cases} \tag{5}$$

The droplet Reynolds Number is shown in the following equation where μ_g is the domain fluid viscosity.

$$Re_d = \frac{\rho_g \left| u_{rel} \right| D_d}{\mu_g} \tag{6}$$

F_g is a force including the effects of gravity and buoyancy:

$$F_g = V_d \left(\rho_d - \rho_g \right) g \tag{7}$$

F_P is the pressure force, given by

$$F_P = V_d \cdot \nabla P \tag{8}$$

F_b summarizes other external forces like the so-called virtual mass force, magnetic or electrostatic forces, Magnus force or others.

2.2.2. Evaporation
In this study, the Dukowicz model [22] is applied for treating the heat-up and evaporation of the droplets during the interaction between the droplets and the gas. This model assumes a uniform droplet temperature. In addition, the rate of droplet temperature change is determined by the heat balance, which states that the heat convection from the gas to the droplet either heats up the droplet or supplies heat for vaporization.

$$m_d\, c_{pd} \frac{dT_d}{dt} = l \frac{dm_d}{dt} + \dot{Q} \tag{9}$$

The convective heat flux $\dot{Q}$ supplied from the gas to the droplet surface is:

$$\dot{Q} = \alpha A_s (T_\infty - T_s) \tag{10}$$

Where α is the convective heat transfer coefficient through the film surrounding the droplet in the absence of mass-transfer and A_s is the droplet surface area. T_∞ is gas temperature and T_s is the droplet temperature.
With the convective heat transfer coefficient in equation (10) replaced by the Nusselt number the convective heat flux $\dot{Q}$ supplied from the gas to the droplet surface is:

$$\dot{Q} = D_d \pi \lambda Nu \left(T_\infty - T_s \right) \tag{11}$$

For spherical droplets the heat flux $\dot{Q}$ can be obtained from heat transfer coefficient correlations.

The Nusselt number Nu is obtained from the following correlation proposed by Ranz and Marshall [24] for single droplets and confirmed to apply to certain types of sprays by Bose and Pei [25].

$$Nu = 2 + 0.6\,Re_d^{1/2}Pr^{1/3} \tag{12}$$

Here, the droplet Reynolds number (Re_d) is obtained from equation (6) and Prandtl (Pr) numbers is evaluated from the usual expressions. The reference temperature for evaluating transport properties like vapor viscosity, specific heat, thermal conductivity etc, is the average temperature between the local domain fluid temperature and the droplet surface temperature as follows:

$$\bar{T} = \frac{T_\infty + T_s}{2} \tag{13}$$

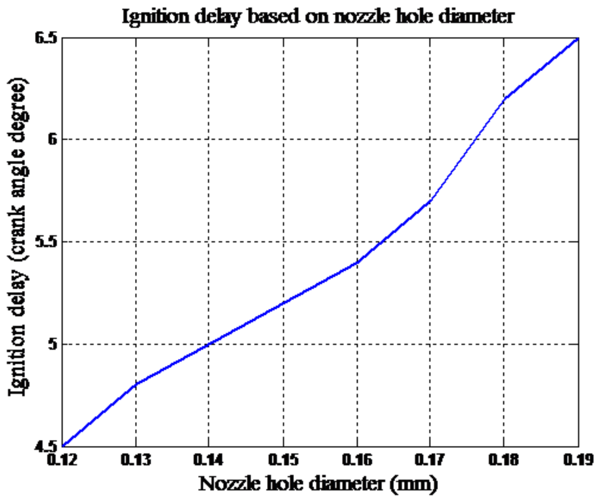

Figure 2: Increasing ignition delay by increasing the nozzle hole diameter

With the above explanations in sections 2.2.1 and 2.2.2, it is revealed that the nozzle hole diameter which is equal to the primary diameter of droplets, affects the momentum equations of droplets and the heat transfer between droplets and the gas when the droplets are injected in to the combustion chamber. Hence nozzle hole diameter directly influences the motion and evaporation of spray droplets and consequently affects the combustion process.

Fig. 2 depicts an increase in nozzle hole diameter disturbing good mixture formation and leading to increased ignition delay. Indeed, increasing the nozzle hole diameter increases the primary diameter of droplets which must mix with gas until ignition happens. Big droplets can not mix with gas as well as small droplets and increasing the nozzle hole diameter postpones the initiation of ignition.

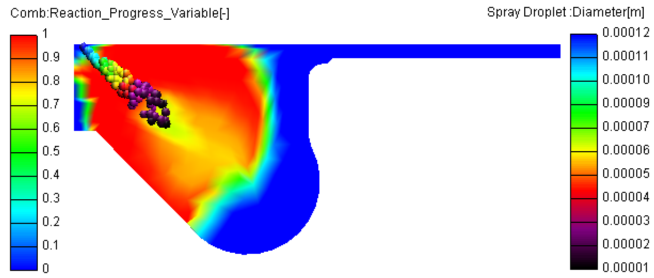

Figure 3: Combustion progress and spray penetration at +13 CAD after start of injection at nozzle hole diameter 0.12 mm

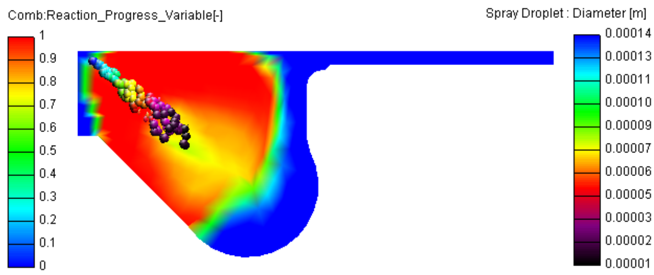

Figure 4: Combustion progress and spray penetration at +13 CAD after start of injection at nozzle hole diameter 0.14 mm

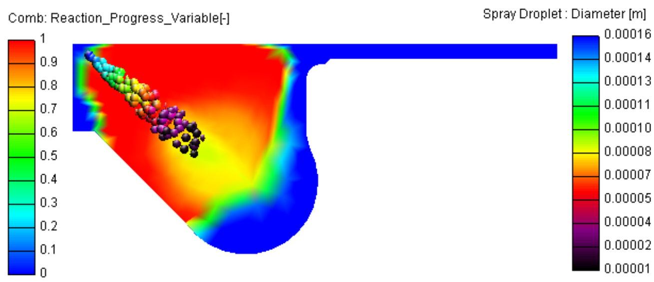

Figure 5: Combustion progress and spray penetration at +13 CAD after start of injection at nozzle hole diameter 0.16 mm

On the other hand increasing the diameter of nozzle holes (when injection pressure is kept constant) causes the injecting fuel mass flow rate to increase and injection duration to decrease, which increases the intensity of the combustion process. Figs. 3...6 confirm the above explanation and illustrate that increasing nozzle hole diameter, increases the intensity

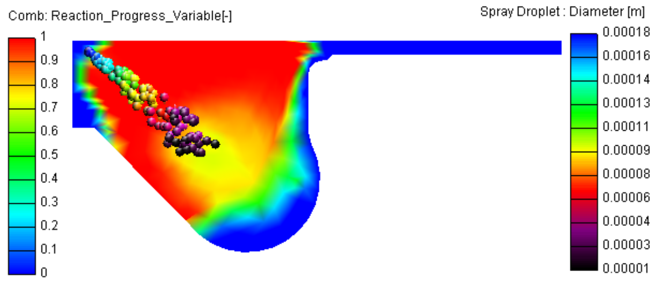

Figure 6: Combustion progress and spray penetration at +13 CAD after start of injection at nozzle hole diameter 0.18 mm

of the combustion process and causes the combustion process to happen in a more intense fashion and to complete faster.

2.3. Second law analysis

The cylinder availability balance can be formulated as follows [2]:

$$\frac{dA_{cyl}}{d\Phi} = \frac{\dot{m}_{in}b_{in} - \dot{m}_{out}b_{out}}{N} - \frac{dA_w}{d\Phi} - \frac{dA_l}{d\Phi} + \frac{dA_f}{d\Phi} - \frac{dI}{d\Phi} \tag{14}$$

Where $\dot{m}_{in}$ is the inlet mass flow from the inlet manifold, $\dot{m}_{out}$ is the outlet mass flow from the outlet manifold and b represents flow availability as follows [2]:

$$b = (h - h_o) - T_o\,(s - s_o) \tag{15}$$

${}^{dA_w}/_{d\Phi}$ represents indicated work transfer. In fact it can be known as the value of output availability from the cylinder associated with the indicated work [2]:

$$\frac{dA_w}{d\Phi} = \left(P_{cyl} - P_o\right)\frac{dV}{d\Phi} \tag{16}$$

Where ${}^{dV}/_{d\Phi}$ states the rate of change of cylinder volume based on crank angle degree and P_{cyl} is the instantaneous cylinder pressure, which are both computed by first law analysis about the engine processes. P_o represents the ambient pressure.

${}^{dA_l}/_{d\Phi}$ is the rate of availability loss associated with heat transfer to the cylinder walls on the basis of the crank angle degree. It can be given as follows [2]:

$$\frac{dA_l}{d\Phi} = \frac{dQ_l}{d\Phi}\left(1 - \frac{T_o}{T_{cyl}}\right) \tag{17}$$

${}^{dQ_l}/_{d\Phi}$ is the rate of heat transfer to the cylinder walls on the basis of the crank angle degree and T_{cyl} is the instantaneous temperature of the cylinder gases, which are both available from the first law analysis. T_o represents the ambient temperature.

${}^{dA_f}/_{d\Phi}$ is the burned fuel availability based on the crank angle degree, which can be given as follows [2]:

$$\frac{dA_f}{d\Phi} = \frac{dm_{fb}}{d\Phi}a_{fch} \tag{18}$$

a_{fch} represents the chemical fuel availability. The chemical exergy (availability) for a substance not present in the environment (e.g. fuel, sulfur, combustion products such as NO or OH, etc) can be evaluated by considering an idealized reaction of the substance with other substances for which the chemical exergies are known [26]. This chemical exergy of the fuel can be expressed as follows on a molar basis [6, 26]:

$$\bar{a}_{fch}(T_o, P_o) = \bar{g}_f(T_o, P_o) - \left(\sum_P x_P\bar{\mu}_P^o - \sum_r x_r\bar{\mu}_r^o\right) \tag{19}$$

Where index P denotes products (CO_2, H_2O, CO, etc) and index r the reactants (fuel and O_2) of the (stoichiometric) combustion process, T_o and P_o are the dead state temperature and pressure, and the over bar denotes properties on a per mole basis.

For hydrocarbon fuels of the type C_zH_y which are of special interest to internal combustion engines applications, Eq. (19) becomes [26]:

$$\begin{aligned}\bar{a}_{fch} &= \overline{HHV}(T_o, P_o) \\ &-T_o\left[\bar{s}_f + \left(z + \tfrac{y}{4}\right)\bar{s}_{o_2} - z\,\bar{s}_{co_2} - \tfrac{y}{2}\,\bar{s}_{H_2O(l)}\right](T_o, P_o) \\ &+\left\{z\,\bar{a}_{CO_2,ch} + \tfrac{y}{2}\bar{a}_{H_2O\,(l),ch} - \left(z + \tfrac{y}{2}\right)\bar{a}_{O_2,ch}\right\}\end{aligned} \tag{20}$$

The above relation is often approximated for liquid fuels (on a kg basis now) by (Ref. [6]):

$$a_{fch} = LHV\left(1.04224 + 0.011925\frac{y}{z}\right) - \frac{0.042}{z} \tag{21}$$

${}^{dI}/_{d\Phi}$ states the rate of irreversibility production in the cylinder which involves combustion, viscous loss, turbulence, mixing and etc. Since combustion has the maximum contribution (more than 90%) [7, 27–29], in the present work, only the combustion irreversibility is taken into account to evaluate the in cylinder irreversibilities. However $^{dI}/_{d\Phi}$ can be evaluated as follows [30]:

$$\frac{dI}{d\Phi} = -\frac{T_o}{T_{cyl}} \sum_j \mu_j \frac{dm_j}{d\Phi} \tag{22}$$

The subscript j involves all the reactants and products. For ideal gasses $\mu_j = g_j$and for fuel $\mu_f = a_{fch}$. S_{gen} is the entropy generation per a cycle which can be given as follows [2]:

$$S_{gen} = (S_{out} - S_{in}) + \frac{Q_{loss}}{T_o} + \frac{I_{total}}{T_o} \tag{23}$$

Where S states gas entropy, Q_{loss} states the heat transfer to the cylinder walls, I_{total} is the total irreversibilities and T_o is the ambient temperature.
The above equations can be solved by numerical methods so the second law terms can be evaluated during a cycle. In this study, a computational code developed in house is applied to evaluate the second law terms.

3. Results and discussion

Table 1: Engine characteristics

Bore×Stroke	82.5×82 mm
Conrod	130 mm
Compression ratio	19.5
Swirl ratio IVC	3
Number of injection nozzle holes	5×0.15 mm
Rail pressure	540 to 1255 bar, dependent on speed
Engine speed	3500 rev/min

In the present investigation, a Ford Diesel 1.8 L engine was used to evaluate the first and second law terms. Table 1 shows the engine characteristics.
To show model validation from the first law perspective, diagrams of the cylinder pressure and cumulative heat release are compared with the previous experimental data [31] at two engine speeds (Figs. 7 and 8) and a good agreement is seen.

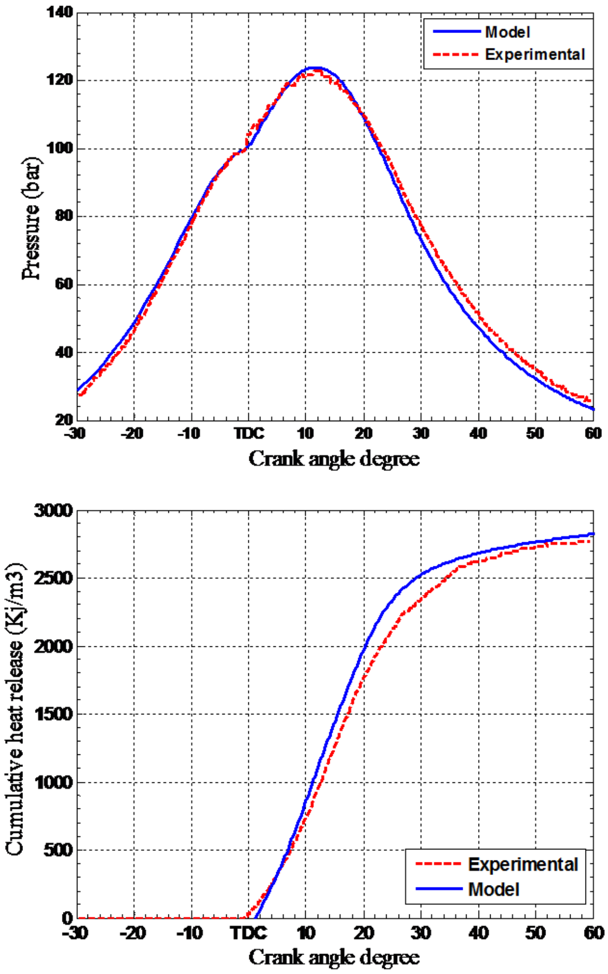

Figure 7: Model validation at engine speed 3000 rpm

To show model validation from the second law perspective, diagrams of heat loss availability Fig. 9, combustion irreversibility Fig. 10 and exhaust gas availability Fig. 11, which are computed by the developed in house code, are compared with the work of C. D. Rakopoulos and E. G. Giakoumis [7]. It is necessary to note that the operating speeds of the engine, which are used by the latter authors, are much lower than the operating speeds of the engine in the present study. Nevertheless, the behaviors of the diagrams confirm each other.
Figs. 9, 10 and 11 also state that an increase in speed causes a decrease in cylinder heat loss and an increase in combustion irreversibilities (although at a lower rate after the mid engine speed range) and exhaust gas availability, (all reduced to the injected fuel availability). A detailed study done by C. D. Rakopoulos and E. G. Giakoumis [32] clearly

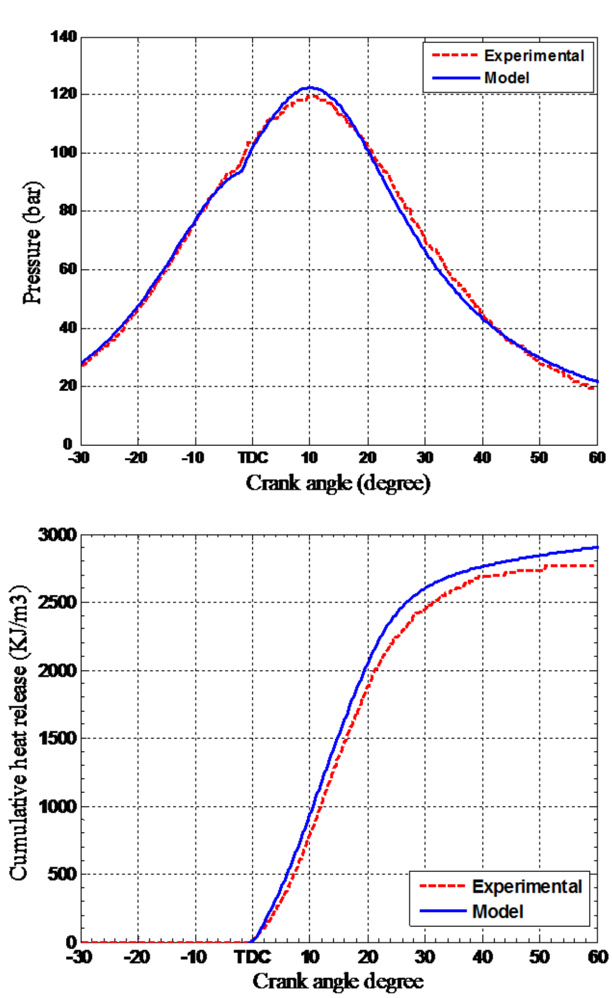

Figure 8: Model validation at engine speed 3500 rpm

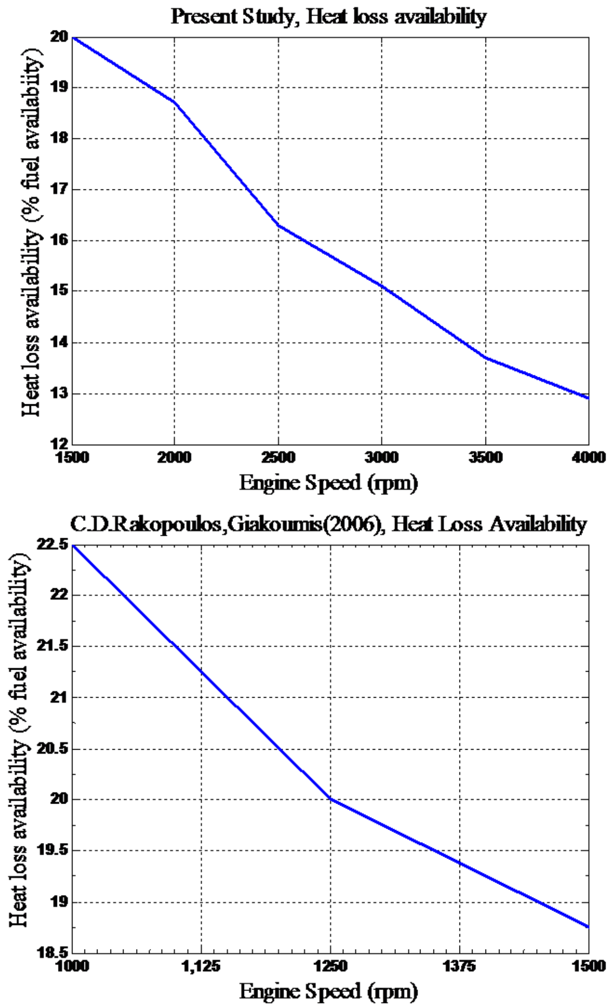

Figure 9: Model validation, Heat loss availability

explains engine speed effects on the availability balance in the Diesel engine. For more on the reasons of the above figures procedure, see Ref. [32].

The results of this study are given in a series of diagrams which show various first and second law terms based on eight nozzle hole diameters as presented in the Figs. 9...11.

The first law terms are presented in the Figs. 12...14.

Second law terms are shown in Figs. 15...17. It is known from the work of W. R. Dunbar and N. Lior [33], that about 80% of the combustion irreversibilities occur during the heat transfer process between the reacting gas and the as yet unburned mixture and it is also known that increasing the gas temperature reduces the relative amount of heat transfer from the reacting gas to the as yet unburned mixture. Since an increase in nozzle hole diameter leads to an increase in combustion temperature (or gas temperature), this eventually decreases the combustion irreversibilities reduced to the fuel availability.

Fig. 18 shows that increasing the nozzle hole diameter decreases the exhaust gas availability reduced to the injected fuel availability. Increasing the nozzle hole diameter leads to increased the indicated work availability (see Fig. 15), increased heat loss availability (see Fig. 16) and decreased combustion irreversibilities (see Fig. 17). However the ensemble of the increase in indicated work availability and the increase in heat loss availability is bigger than the decrease in combustion irreversibilities, therefore the part of the injected fuel availability which can be consumed by in cylinder processes (such as indicated work, heat loss availability and combustion

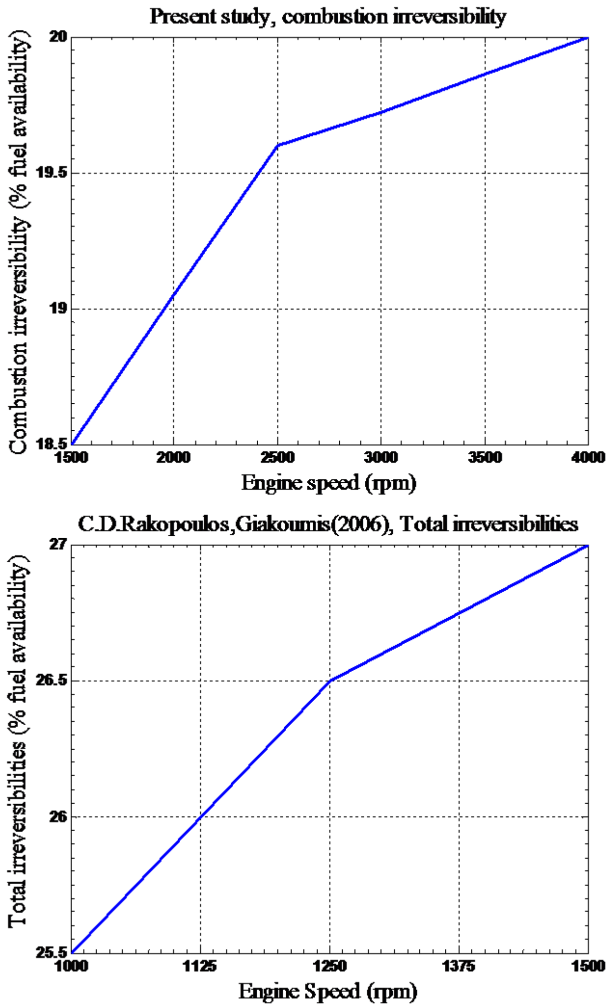

Figure 10: Model validation, Combustion irreversibility

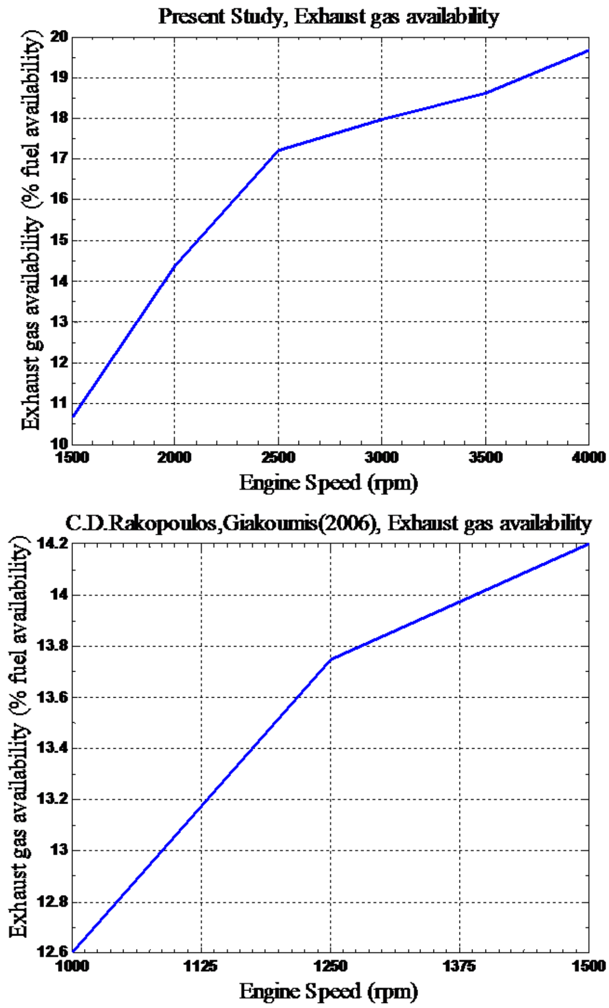

Figure 11: Model validation, Exhaust gas availability

irreversibilities) is increased and hence the amount of injected fuel availability which goes out in the exhaust gas from the cylinder, is reduced by increasing the nozzle hole diameter.

The above interpretation shows that an increase in nozzle hole diameter reduces exhaust gas availability.

Fig. 19**:** This figure reveals that an increase in the nozzle hole diameter leads to an increase in the entropy generation per cycle. This result can be explained by referring to equation (23). When the nozzle hole diameter is increased, the temperature level of the combustion process increases which raises the entropy of the outlet gas (S_{out}) and the heat loss to walls (refer to Fig. 14) and reduces the combustion irreversibilities (see Fig. 17). However, the ensemble of the increase in entropy of the outlet gas and the increase of heat loss to walls, which are both due to the increase in the nozzle hole diameter, is greater than the reduction in combustion irreversibilities. Hence by referring to equation (23), it is explained that an increase in the nozzle hole diameter leads to a decrease in the amount of entropy generation per cycle.

4. Conclusions

A numerical investigation was used to study the influence of the nozzle hole diameter on the first and second law balance in a DI Diesel engine. The results show that an increase in the nozzle hole diameter causes an increase in the indicated work and heat loss to walls. With regard to second law terms, the results show that an increase in nozzle holes diameter leads to an increase in indicated work availability, heat loss availability and entropy generation per cy-

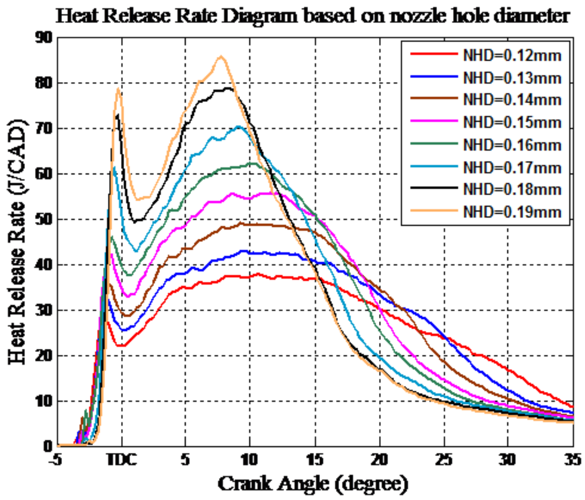

Figure 12: Increase in intensity of combustion and decrease in combustion duration by increasing the nozzle hole diameter. This figure reveals that increasing the nozzle hole diameter leads to increased combustion intensity and a shortened combustion process. It is owing to this fact that increasing the diameter of nozzle holes (when injection pressure is constant) causes the injecting fuel mass flow rate to increase and injection duration to decrease, which increase combustion intensity and reduce combustion duration

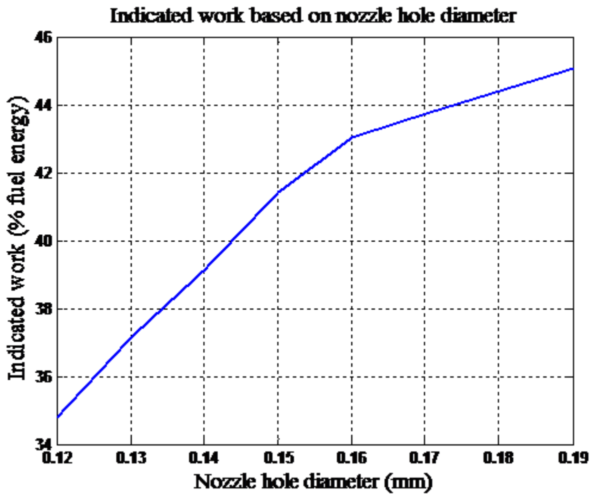

Figure 13: Increase in indicated work by increasing the nozzle hole diameter. This figure shows that increasing the nozzle hole diameter leads to an increase in the indicated work reduced to the fuel energy. Indeed increasing the nozzle hole diameter increases the intensity of combustion, which causes the mean pressure level of gas in the cylinder to rise. This raises the indicated work

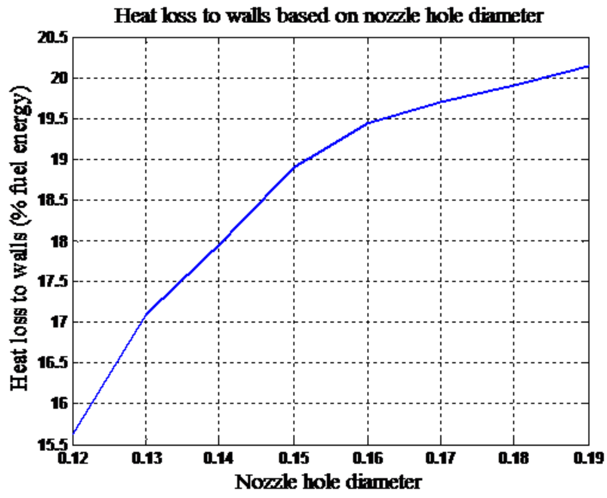

Figure 14: Increase in heat loss to walls by an increase in nozzle hole diameter. This figure shows that increasing the nozzle hole diameter causes an increase in heat transfer to walls. The increased combustion intensity caused by increased fuel mass flow rate due to increased the nozzle hole diameter, raises the mean temperature level of cylinder gas, which leads to an increase in heat transfer to walls

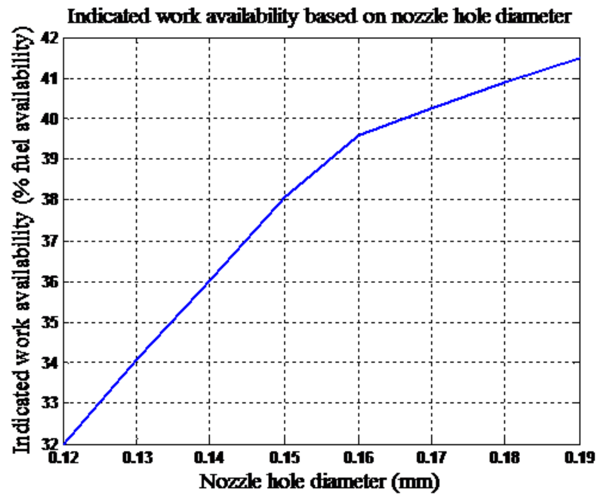

Figure 15: Increase in indicated work availability by an increase in nozzle hole diameter. This figure reveals that an increase in the nozzle hole diameter leads to an increase in the indicated work availability reduced to the injected fuel availability. It is understood from Figs. 12...14 that an increase in nozzle hole diameter increases the indicated work (see Figs. 12...14), which increases indicated work availability

cle and a decrease in combustion irreversibility and exhaust gas availability.

The results reveal that changes in indicated work availability and exhaust gas availability are more severe than changes in heat loss availability and combustion irreversibility when the nozzle hole diameter is altered. The combustion irreversibilities exhibit the smallest rate of change. Therefore exhaust gas availability and indicated work availability have

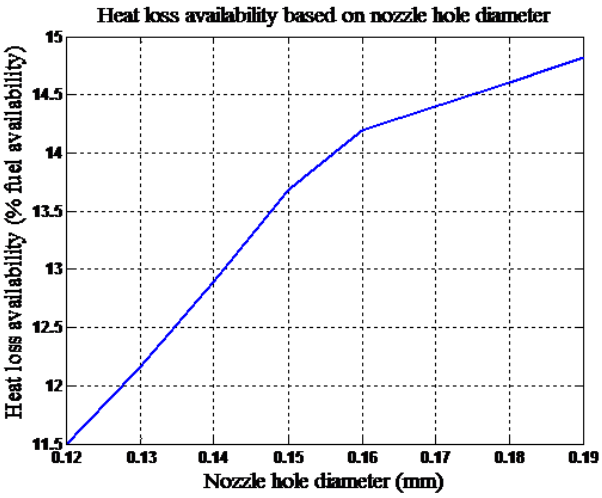

Figure 16: Increase in heat loss availability by an increase in nozzle hole diameter. This figure illustrates that an increase in the nozzle hole diameter increases the heat loss availability reduced to the injected fuel availability. It is known from Figs. 12...14 that an increase in nozzle hole diameter raises the mean temperature level of cylinder gas (see Figs. 12...14), which leads to an increase in heat loss availability

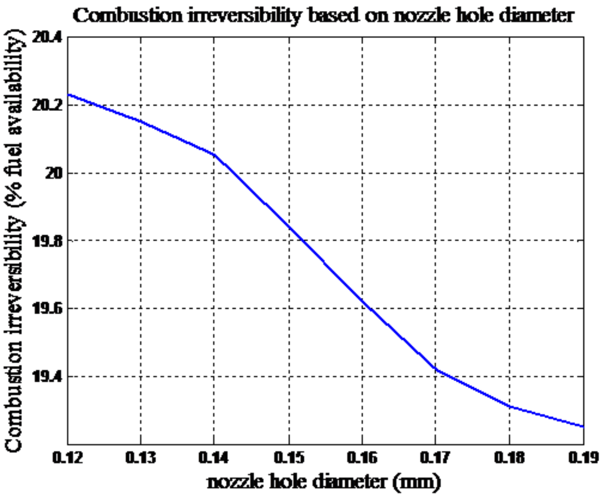

Figure 17: Decrease in combustion irreversibility by an increase in nozzle hole diameter. In this figure, it is shown that an increase in the nozzle hole diameter reduces the amount of combustion irreversibilities reduced to the injected fuel availability

the biggest sensitivity and combustion irreversibilities have smallest sensitivity to changes in the nozzle hole diameter.

The above results show that by finding the optimum nozzle hole diameter it is possible to achieve a good balance between the first and second law terms, which leads to better engine design.

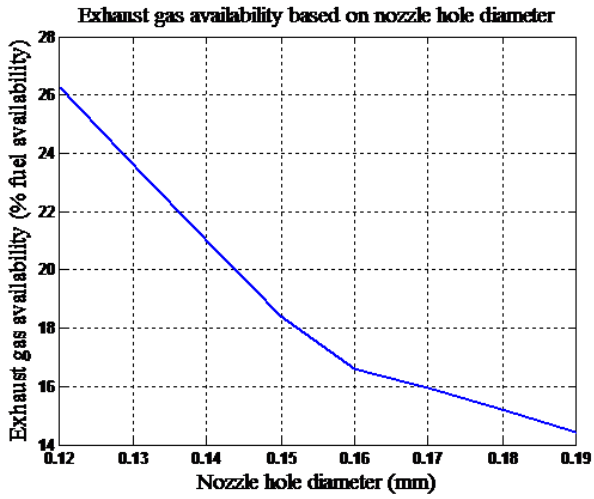

Figure 18: Decrease in exhaust gas availability by an increase in nozzle hole diameter

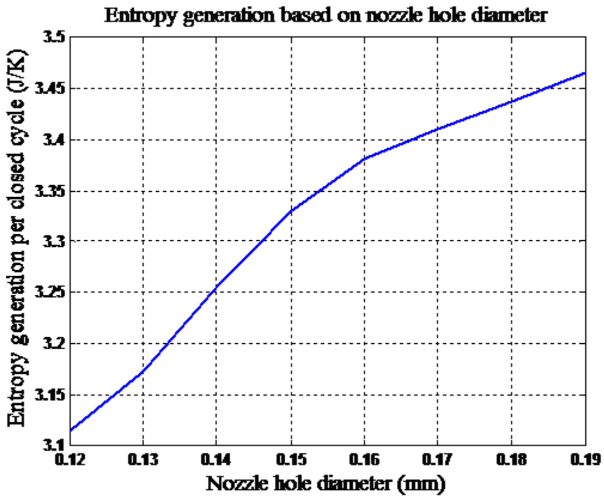

Figure 19: Increase in entropy generation by an increase in nozzle hole diameter

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Nomenclature

α heat transfer coefficient, W/(m^{2}K)

$\dot{m}$ mass flow rate, kg/s

$\dot{r}$ fuel consumption rate

∞ droplet far - field condition

λ thermal conductivity, W/(m·K)

μ chemical potential, J/kg

Φ crank angle, deg

ρ density, kg/m^3

τ_r time scale of turbulent mixing, s

A availability/ exergy, J

a specific availability/ exergy, J/kg

A_s droplet surface area, m^2

b flow availability (flow exergy), J/kg

C carbon

C_D drag coefficient, -

C_p specific heat, J/(kg·K)

ch chemical

cyl cylinder

D diameter, m

d droplet

D_p drag function, N·s/m

DI direct injection

dr drag

F force, N

f fuel

fu fuel

g gas

g gravity

g specific Gibbs free enthalpy, J/kg

gen generation

H hydrogen

h specific enthalpy, J/kg

HHV higher heating value

I irreversibility, J

$IMEP$ indicated mean effective pressure

in inlet

$ISFC$ indicated specific fuel consumption

j all reactants and products

l latent heat of evaporation, J/kg

l loss

LHV lower heating value

m_d droplet mass, kg

N engine speed, rpm

Nu Nusselt number

o ambient condition

P pressure, Pa

P pressure

P product

Pr Prandtl number

Q heat, J

r reactant

Re Reynolds number

rel relative

rpm revolutions per minute

S entropy

S stoichiometric oxygen requirement, source term

s droplet surface

s specific entropy, J/kg·K

u velocity, m/s

V volume, m^3

w work

y number of hydrogen atom

z number of carbon atom