DESIGN AND PERFORMANCE EVALUATION OF A NOVEL SELF-ROTATING FUEL INJECTOR USING COMPUTATIONAL FLUID DYNAMICS – A PRELIMINARY STUDY

by

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Improving the performance and reducing emissions in a Diesel engine is the single most objective in current research. Various methods of approach have been studied and presented in literature. A novel but not so pursued study is on the performance of a rotating diesel injector. To date, there has been very little study by implementing a rotating injector. Studies have shown an improvement on the performance of an engine, but with a complicated external rotating mechanism. In the present research, a novel self-rotating fuel injector is designed and developed that is expected to improve the performance without the need for a complicated rotating mechanism. The design procedure, CFD simulation along with 3-D printing of a prototype is presented. Numerical modelling and simulation are performed to study the combustion characteristics of the rotating injector viz-a-viz a standard static injector. Comparison based on heat release, efficiency, and emissions are presented. While the proposed 9-hole injector had slight loss in thermal efficiency, the modified 5-hole had a slight increase in thermal efficiency when compared to the static baseline readings. The NO_x reduced by 13% and CO increased by 14% compared baseline emissions for the 5-hole version.

Key words: *rotating nozzle, combustion, CFD, emission*

Introduction

There is a continuous effort to improve the performance of an engine in terms of power, fuel consumption, and emissions characteristics. Improvement of anyone of the aforementioned parameters is detrimental to the other hence; an optimized design is generally arrived with a trade-off. The effort therefore is to minimize this trade-off to ensure higher levels of performance under all categories. The performance, emission and combustion characteristics have been studied for Diesel engine employing various methods to improve performance, and a few are discussed here.

The piston bowl geometry is modified and studied for improvement in efficiency, swirl induction, atomization and emission. It was presented by Kumar [1] that a toroidal geometry performs better than that of a hemispherical one, for B20 diary scum fuel. Channappagoudra, *et al.* [2] showed that modifying piston bowl geometry aids in improving the per-

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formance and emissions except for NO_x. Abd El-Sabor *et al.* [3] performed a CFD analysis to study the effect of valve shrouding and orientation angle. They observed that though both influences swirl motion, it has negligible effect on the tumble inside the chamber.

Li *et al.* [4] experimentally studied that the effect of split injections coupled with swirl in a Diesel engine and found that performance was achieved due to improved combustion process. The soot concentration was also lower. Similarly, both experimental and numerical study was done by Wang *et al.* [5] to study influence of intake swirl on fuel spray characteristics. They observed that with increase in swirl ratio resulted in higher NO and lower soot formation.

A novel proposal of an auto rotating injector was first proposed and patented by Klomp *et al.* [6]. However, there was no information on the physical development and experimentation on the said device. Sjoberg [7] further studied the performance of a rotating injector with an external power. He found that the rotating spray has a significant effect on spray formation, dispersion and penetration when compared to a normal static injection. The NO_x emission was also found to be higher. Contrary to this observation Senkonttaiyan *et al.* [8] observed a lower pressure and heat release rate (HRR) attribute to a lower NO_x. This probably could have been due to inefficient sealing during rotation. In this proposed research a self-rotating nozzle is designed and simulated numerically for combustion and emission characteristics. The rotating rpm depends on the torque produced by the ejected fuel.

Numerical methodology

The hypothesis for implementing a rotating nozzle is that, the fuel atomized and dispersed is superior to a static injector. The current design is self-rotating by harnessing the thrust developed from the ejected fuel and different from that of Klomp *et al.* [6]. The simulation approach is:

- Design of a self-rotating injector.
- Test the self-rotating feature by nozzle hole numbers, nozzle hole diameter for a given flow rate.
- Optimize the minimum flow rate, hole diameter and hole numbers for a rotating speed of 1500 rpm.
- Use the previous optimized values to test the combustion parameters and compare with an equivalent static injector.

Design of rotating nozzle

The model of the injector with a rotating ring is shown in fig. 1. The cross-sectional view of the injector is shown in fig. 1(a), and the self-rotating nozzle in fig. 1(b). Here, L represents the ring thickness and designed as 2.5 mm. In this design, fuel ejects out of two holes from the sac and gets distributed to the holes in the ring. Thus, as the ring rotates, the fuel through each of these holes will vary in flow rate from a minimum to a maximum value as it sees the inlet orifice. The engine studied is a standard 839 cc rated at 1500 rpm. Thus, the nozzle rotation within the cylinder should be at least 1500 rpm for fuel dispersion. The mass flow pattern will be sinusoidal in nature and the pattern will have a phase lag from one hole to another. This pattern can be seen in fig. 2.

The actual dimensions have not been presented for confidentiality. The calculations provided below are for a nozzle ring of 0.5 g as an example and not to be construed as actual.

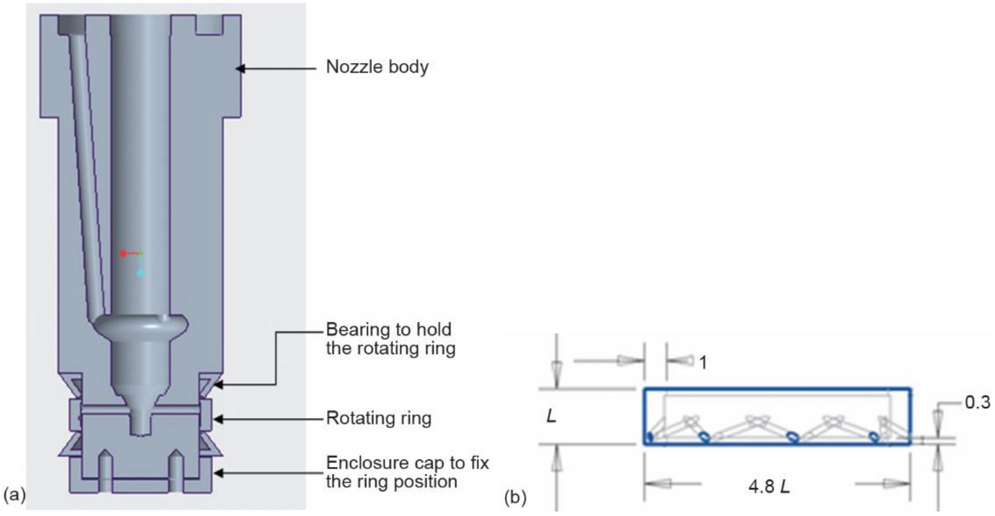

Figure 1. (a) Cut section, (b) nozzle ring

Moment of inertia, $I = mR^2$ [kgm^{-2}]	Power, $P = T\omega$, [W]
Angular acceleration, $\alpha = \omega/t$ [rads^{-2}]	Torque, $T = F_t r\cos 20$ [Nm]
Angular force, $\tau = I\alpha$ [Nm]	Fluid thrust force, $F_t = \rho A V^2 \cos\theta C_t$ [N]

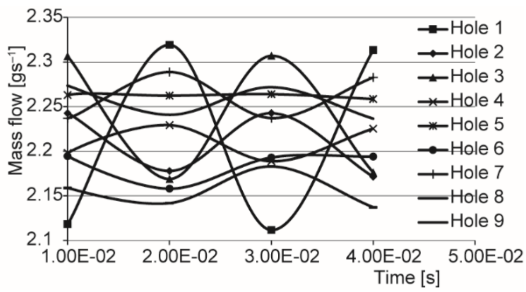

Figure 2. Orifice 0.02 kg/s and 0.1 mm, for 9 holes at 1500 rpm

With this configuration the injection velocity is calculated to achieve the desired rpm for the nozzle ring. The torque generated by the injected fuel through the 9-hole is thus calculated:
– the mass of the ring is calculated from the CAD design as $m = 5e^{-4}$ kg,
– ring radius, $R = 2.4\ L = 6$ mm,

where L is the ring thickness is designed as 2.5 mm and labeled in fig. 1(b).

Initially the torque required to rotate the nozzle ring to 1500 rpm is calculated. This torque is then used to work backwards to estimate the velocity of the injected fuel.

The moment of inertia of the ring is computed from eq. (1).

$$I = m \times R^2, \tag{1}$$

The desired rotation is 1500 rpm or angular velocity 157 rad/s. It is assumed that the ring will take 1 ms to achieve this rotational speed. Thus, the angular acceleration, α, is computed from eq. (2):

$$\alpha = \frac{\omega}{t} \tag{2}$$

Therefore, the desired torque is obtained from eq. (3);

$$\tau = I\,\alpha \tag{3}$$

This torque is now equated using eq. (4) with the thrust generated by the diesel injected through the 9-hole to calculate the desired velocity;

$$\tau = \rho A v^2 R \cos\theta C_t \tag{4}$$

where $\theta = 20°$ is the ejected angle with respect to the tangent.

The minimum velocity required for initiation of self-rotation at 1500 rpm is thus found to be 320 m/s.

The velocity of the fuel as it ejects out of the nozzle is computed from eq. (5):

$$\dot{m} = \rho A v \tag{5}$$

$$\rho = 846 \text{ kg/m}^3, \; A = \frac{\pi}{4} D^2 \times 9\text{-hole}$$

The mass-flow rate required for an 839 cc engine is 0.02 kg/s. Therefore, the velocity at which the fuel ejects is 335 m/s. As this velocity (335 m/s) is greater than the minimum calculated velocity of the designed ring (320 m/s), will safely rotate at 1500 rpm. Similarly the same procedure is used to estimate the self-rotating rpm of a 5-hole injector for the same mass flow rate. The 5-hole nozzle was found to have a higher velocity at 360 m/s than 9-hole. This observation is logical due to decreased flow area. However, the reduced number of holes yields a lower thrust and rotates at 1000 rpm. It is also necessary to verify if 5-hole would be better than the 9-hole counterpart in terms of combustion and emission characteristics, and the reason for the same is discussed later.

From the calculation it is seen that the minimum injection velocity desired 320 m/s. This can be achieved with a minimum mass flow rate of 0.02 kg/s. A fluid structure interaction was also performed to test the flow rates to obtain a minimum rotation of 1500 rpm using 9-hole and 1000 rpm for 5-hole version using STAR CCM+ [9].

Numerical model

The designed nozzle is incorporated inside an 839 cc Diesel engine. Simulation is performed using STAR CD software for static as well as rotating fuel injector having 5-hole and 9-hole configurations. The engine specification is provided in tab. 1. The domain is discretizing with polyhedral cells represented in fig. 3. The piston has a translation motion and the nozzle has a rotational moving reference frame (MRF). The present computational domains couple these two motions with combustion.

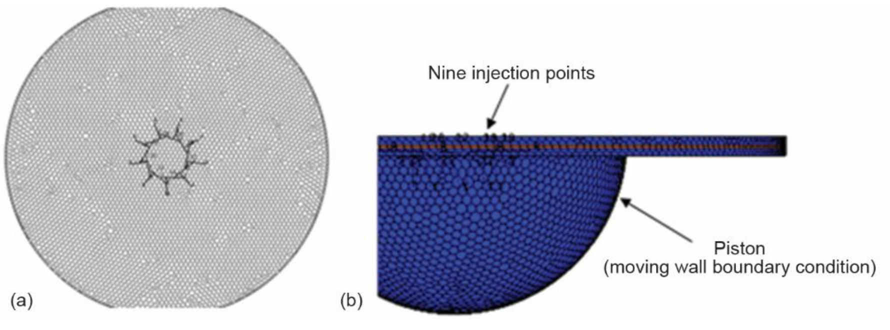

Figure 3. Computational mesh setup (plan and elevation)

Table 1. Engine specification and condition

Engine	Boat engine	Engine	Boat engine
Cylinder bore × stroke [mm]	96 × 116	Fuel used	$C_{12}H_{23}$ (Jet – A)
Compression ratio	17.5: 1	Static injector	3-hole
Cubic capacity [cc]	839	Rotating injector	5-hole and 9-hole
Connecting rod length [mm]	245	Nozzle hole diameter [mm]	0.1
Combustion chamber geometry	Hemispherical piston bowl type	Mass of fuel injected [mg]	40
Engine speed [rpm]	1500	Fuel injection rate [kgs^{-1}]	0.0051, 0.0105, 0.02
Injection system	Point injection	Start of injection [°CA]	697, 708, 714
Injection method	Static and rotating		

Boundary conditions

The simulation was performed for 839 cc engine and the total simulation time is 40 ms. The time step is varied at different duration of crank angle and the same is graphically presented in fig. 4. The detailed boundary conditions are tabulated in tab. 2. The self-rotating rpm depends on the thrust developed for the given number of holes as explained in chapter *Design of rotating nozzle*.

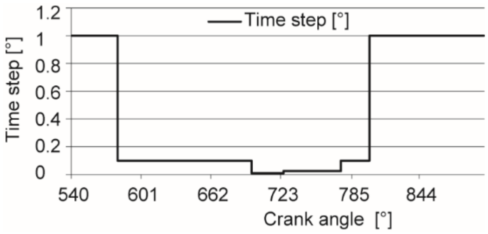

Figure 4. Simulation time step *vs.* crank angle

Table 2. Summary of boundary conditions and numerical value

Description	Boundary conditions
Computational geometry Orifice diameter [mm]	 0.1
Boundary conditions Inlet Piston Cylinder wall and cylinder head Nozzle 5-hole at 360 m/s 9-hole at 335 m/s	Mass-flow 0.02 kg/s Moving wall with adiabatic boundary conditions Static wall with adiabatic boundary conditions Rotating nozzle (MRF) 1000 rpm 1500 rpm
Mesh count	5,50,000 cells
Solver settings Turbulence Atomization and spray break-up Combustion	 k-ε model Reitz Diwakar model Extended Coherent Flame Model-3Z

Atomization and spray break-up model

The atomization process can be done by a combination of three reasons: turbulence within the liquid phase, disintegration of cavitation particles, and aerodynamic forces acting on the liquid jet Stiesch [10]. The standard atomization models are in the commercial code: Huh's model [11], Reitz Diwakar (RD) model [12], MPI model [13], and modified MPI model [14]. In this study, RD model has been used for atomization characteristics and secondary break up was considered as was presented in Margot *et al.* [15].

The STAR – CD tool [16] has some specific models for analysing the droplet behaviour. The following standard models are available for simulation, such as; RD, Plich and Erdman, Hsiang and Faeth and Kelvin-Helmholtz Rayleigh Taylor (KHRT) model. Detailed discussion on the various models in out of the scope of the present research and suffice to say that the RD and KHRT models are the most popular. In the present simulation the RD atomization and RD break-up model is utilized. The break-up rate is given:

$$\frac{\mathrm{d}D_\mathrm{d}}{\mathrm{d_t}} = \frac{D_\mathrm{d} - D_\mathrm{d,stable}}{\tau_\mathrm{b}} \tag{7}$$

More detailed information is available in Hossainpour *et al.* [17], Margot *et al.* [15] and Fujimoto *et al.* [18].

Combustion model

Owing to expensive physical experiments, CFD modelling of internal combustion engine plays a very important role in improving the engine design, reduction in emission and fuel consumption. The CFD simulation provide results such as turbulent mixing of air and fuel, the combustion chemistry, knock occurrence, formation of pollutants like NO_x and soot. The STAR– CD [16] offers a set of models to simulate the above processes. The following models are available for combustion simulation in the software such as, extended coherent flame model (ECFM), 3 zone – extended coherent flame model (ECFM – 3Z), equilibrium limited ECFM (ECFM – CLEH), and progress variable model – multi fuel (PVM – MF). In this models ECFM – 3Z and ECFM – CLEH can be used for all type of combustion regimes of non-homogeneous engine (CI engine) whereas, ECFM model is used for SI engines and PVM – MF model is used for multi fuel CI engines. In this study the combustion the ECFM – 3Z model was used.

The ECFM – 3Z model simulates mixing, auto ignition, flame propagation, and emission. The 3Z stands for three zones namely, unmixed fuel zone, mixed zone, and unmixed air + EGR zone.

Utilizing the mass flow of diesel as given in fig. 2, the simulation was performed for one cycle. A cold flow simulation was first performed to match with the pressure obtained from a theoretical adiabatic compression-expansion process to ensure mesh independence. Further combustion simulation was performed to optimize the time steps by comparing with available results in open literature. The current simulation is performed on an 839 cc Diesel engine with a static injector and compared with 5-hole and 9-hole rotating injector. The same quantity of fuel is injected into the cylinder for all the three case by varying the start of injection from 697, 708, and 714 CA for static, 5-hole, and 9-hole cases, respectively. The reason for these changes in timing are to ensure that adequate velocity is obtained at the nozzle holes for both atomization and desired thrust for self-rotation. The angular representation point for the three cases will be the geometric centre of the rotating nozzle ring.

Results and discussions

Cylinder pressure

The variation of cylinder pressure *vs*. CA for static 3-hole compared with rotating 5-hole and 9-hole is presented in fig. 5. The zero value in the *x*-axis represents the TDC. It is generally accepted that the magnitude and occurrence of peak pressure affects engine power and emissions. The peak pressure for the static case is higher among the three cases. The 9-hole 714 CA injection causes a lower peak. The peak cylinder pressure mainly depends on ignition delay period and distribution of fuel droplets, Ganesan [19], the current simulation captures this phenomena. Higher cylinder pressure affects the engine cylinder and piston and releases high temperature exhaust that is directly related to the NO_x emission. The cylinder pressure for rotating nozzles of 5-hole and 9-hole is 82.39 bar, and 73.26 bar respectively, which is about 4% and 15% lower than the static case.

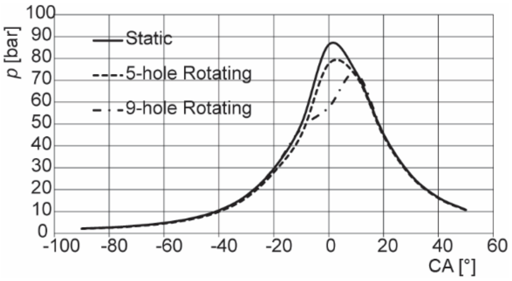

Figure 5. Cylinder pressure *vs.* CA

Apparent heat release rate

The energy released is calculated using the first law of thermodynamics assuming the engine cylinder as a closed system and presented in eq. (8);

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \dot{Q} - p\frac{\mathrm{d}V}{\mathrm{d}t}\dot{m}_\mathrm{f} h_\mathrm{f} \tag{8}$$

The heat release rate *vs*. CA for standard 3-hole nozzle is compared with 5-hole and 9-hole rotating nozzle and presented in fig. 6. The heat release rate can be used to identify the start of combustion, the fraction of fuel burned in the premixed mode, and the differences in combustion rates because of changes in injection system or air swirl levels. The heat release per cycle is computed by integrating the HRR profile over CA. The computed values are 575, 605.27, and 549.01 Joules per cycle for static, 5-hole and 9-hole cases respectively. From the figure it is seen that the heat release peaks for the 5-hole nozzle before the TDC but peaks after the TDC for the 9-hole version. As computed earlier, the 5-hole version has higher velocity (360 m/s) and longer injection duration (12°) compared to the 9-hole nozzle (335 m/s and 6°). This ensures better atomization and combustion although the self-rotation is lesser at 1000 rpm. It is found that the thermal efficiency is 29.05, 30.56, and 27.72 for static, 5-hole, and 9-hole nozzles, respectively. Thus, a rotating nozzle

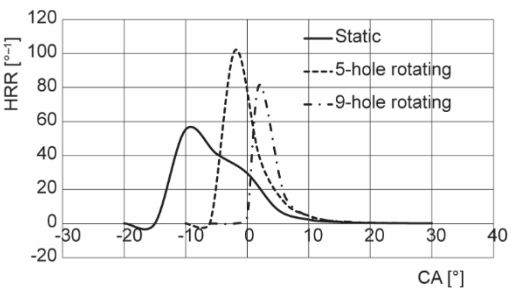

Figure 6. Heat release rate *vs.* CA

with better atomization and dispersion gives higher efficiency. In this case, a 5-hole version performs better than static due to rotation and superior to 9-hole due to higher injection velocity and duration.

Oxides of nitrogen

The difference between standard 3-hole nozzle as compared with 5-hole and 9-hole rotating nozzles in terms of NO_x emission against CA is shown in fig. 7. The main causes of NO_x emission are the higher combustion chamber temperature and presence of oxygen in diesel engine [2]. The NO_x emission was found to be lower for 9-hole rotation when compared to static and 5-hole rotation. Since the injection starts at 714° of crank angle the cylinder pressure, heat release rate and NO_x emission start rising after 720° of CA. Even though fig 7 shows that near the end of combustion process the 9-hole has more NO_x emission than others, the total discharged per cycle is less by 16.7% compared to static nozzle and 8.5% for 5-hole. The NO_x for static, 5-hole, and 9-hole cases are 14, 12.8, and 11.5 g/kWh, respectively. Factoring for the efficiency gain or loss with static case as benchmark, the apparent NO_x released would be 14, 12.16, and 12.07 for the static, 5-hole, and 9-hole cases and presented in tab 3. Clearly simultaneous increase in efficiency with corresponding decrease in NO_x is obtained in the 5-hole case.

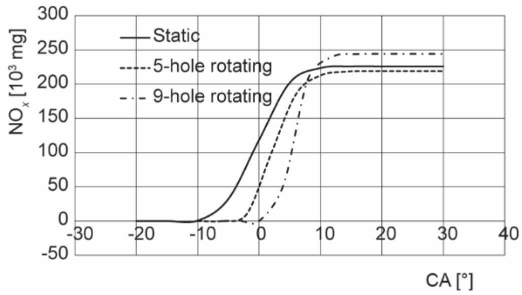

Figure 7. NO_x *vs.* CA

Table 3. Performance parameters

	697 Static Baseline	708-5 hole	714-9 hole	708-5 hole Factor for η	714-9 hole Factor for η
Efficiency, η, [%]	29.07	30.56	27.72	29.07/30.56 (0.95)	29.07/27.72 (1.05)
NO_x, [gkW^{-1}h^{-1}]	14	12.8	11.5	12.16	12.07
CO, [gkW^{-1}h^{-1}]	47.6	57.5	44	54.6	46.2
p, [bar]	86.26	82.39	73.26	–	–
HRR, [Joule cycle^{-1}]	575	605	549	–	–

Carbon monoxide

The CO is a poisonous gas generated by fuel-rich equivalence ratio. Figure 8 shows that a comparison between static 3-hole, and that of 5 and 9-hole rotating nozzle. When there is enough oxygen, most of the carbon in the fuel is converted into CO_2 however, when less oxygen is present, the combustion process leads to more CO. The CO emission is 47.6, 57.5, and 44 g/kWh for the static 3-hole and that of, 5 and 9-hole rotating cases. Factoring for the gain/loss in efficiency, the apparent CO emission would be 47.6, 54.6, and 46.2 g/kWh. It is found that the 5-hole rotating injector has higher CO emission than static and 9-hole rotation, tab. 3. However, from the perspective of optimization it would be safer to assume a 5-hole injector is better of the other two cases studied here.

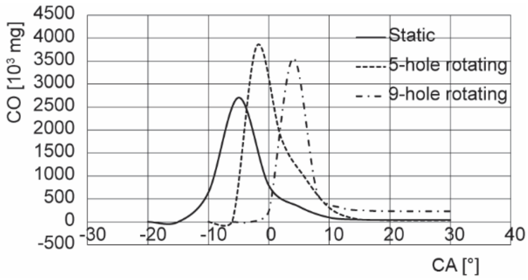

Figure 8. CO *vs.* CA

Conclusions

A novel self-rotating nozzle having 5-holeand 9-hole was designed and developed. The numerical model was studied for both rotating capabilities using a fluid structure interaction model as well as combustion modeling. Existing information in open literature on rotating nozzle concludes that the efficiency is higher with higher emissions. This was done without varying the number of nozzle hole and rpm.

In the present research the self-rotating nozzle was designed and numerically simulated and optimized. Also, the nozzle hole numbers and rpm were varied for reducing emissions with simultaneous increase in efficiency. It was found that the 5-hole nozzle was more efficient in terms of increase in thermal efficiency and decrease in NO_x albeit with a slight increase in CO though the 9-hole has lesser CO than the rest. However, from a practical point of view it is concluded that the 5-holeself-rotating is better of the other two. More modifications in terms of hole size, nozzle ring ejection angle, rounds per minute, and flow rate is required to see if simultaneous increase in efficiency with reduction in emission can be obtained.

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Nomenclature

C_t – coefficient of thrust
F_t – fluid thrust force, [N]
h_f – fuel enthalpy
I – moment of inertia, [kgm^{-2}]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
m – mass, [kg]
$\dot{m}_f$ – fuel addition rate
p – cylinder pressure, [bar]
P – power, [W]
$\dot{Q}$ – heat transfer rate
R – ring radius, [m]
r – radius of nozzle
t – time, [s]
U – internal energy of the cylinder contents
V – cylinder volume

Greek symbols

α – angular acceleration, [rads^{-2}]
τ – angular force, [Nm]
ω – angular velocity of the nozzle

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