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Modelling and Simulation of a four-stroke Compression Ignition engine

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ABSTRACT

An attempt to create a realistic simulation of the processes that are observed inside a compression ignition engine using various thermodynamic and fluid dynamic models. Observing the differences and shortcomings of the developed code for fluid dynamic simulation with a more real life scenario using volumetric efficiency over a range of RPM and how fuel injection timing factors in.

Contents

1 Four-stroke CI Engine Fundamentals

The compression ignition engine was originally developed by Randolf Diesel in Germany in 1892. The basic power-developing components of a typical reciprocating CI engine are the:

1. Cylinder
2. Piston
3. Connecting rod
4. Crankshaft

These are show in in Fig.1.

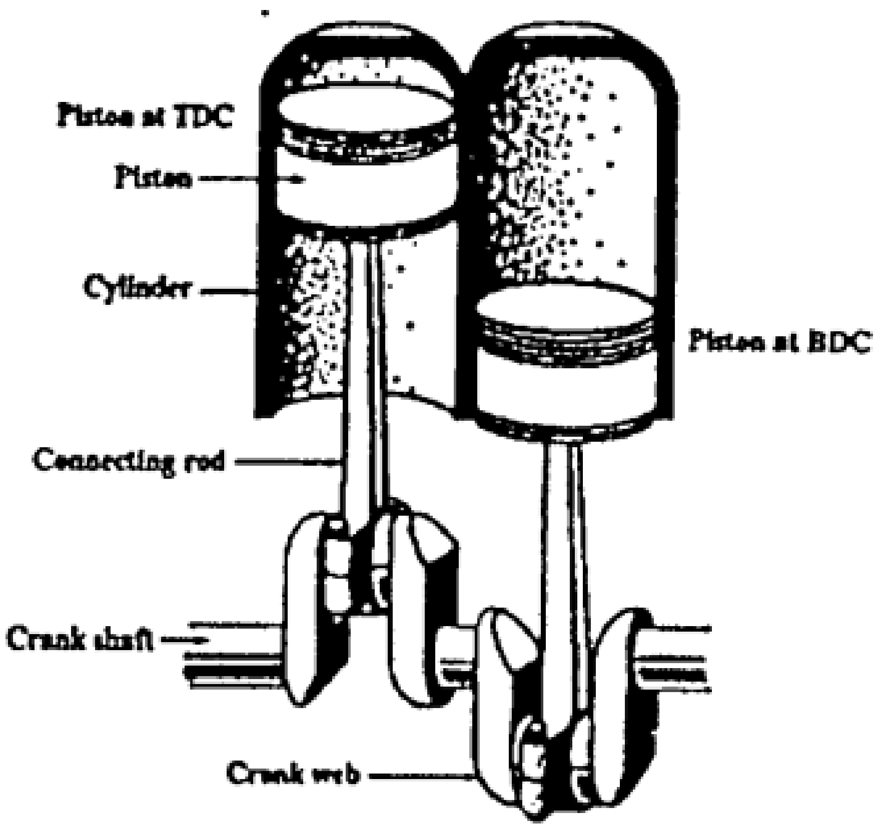

Figure 1: Basic power delivering components of a CI engine [3]

The gap between the cylinder walls and the piston is sealed with the help of piston rings and lubricant so that no gas escapes between this gap. The piston is connecting to the crankshaft by the means of a connecting rod so that the reciprocating motion is converted to rotary motion.
The position of the piston and the moving parts at the moment when the direction of its motion is reversed is called the dead centre. There are two dead centres - top dead centre (TDC) and bottom dead centre (BDC). The distance between TDC and BDC is called the **stroke (S)**. **L** is the length of the connecting rod. **Bore (B)** is the inner diameter of the cylinder. TDC and BDC are used as

reference points for various events throughout the working of the engine.
The volume swept while the piston travels from BDC to TDC is called the **displacement volume** V_{disp} and the volume of the space above the piston when it is at TDC is called the **clearance volume** V_{cc}. The compression ratio is defined as the ratio of volume of space above the piston when it is at BDC to when it is at TDC. In other words

$$\text{compression ratio } r = \frac{V_{bdc}}{V_{tdc}} = \frac{V_{disp}+V_{cc}}{V_{cc}}$$

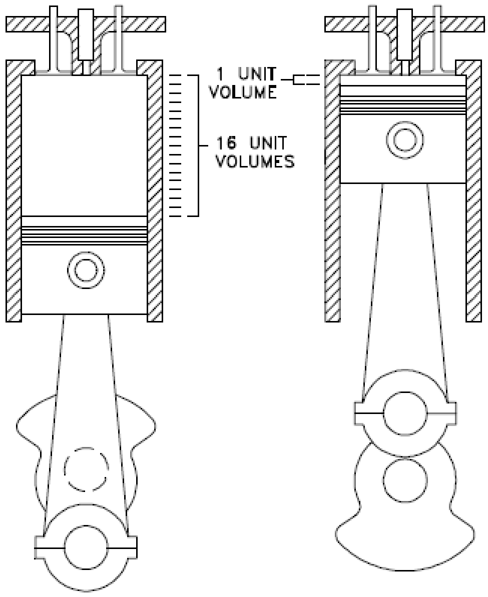

Figure 2: Piston is at BDC on the left and at TDC on the right. V_{disp}=15 unit, V_{cc}=1 unit, r=16

A four-stroke compression ignition engine works based on the four-stroke, five-event principle. The four strokes are:

Suction/Intake Starts with the piston at TDC and ends with the piston at BDC. The cylinder is mostly evacuated of gases after the exhaust stroke that occurred just before, hence the expansion of volume available in the cylinder creates a suction inducting air. To increase the mass of air inducted the inlet valve stays open shortly before the stroke starts and is open shortly after it ends too.

Compression Starts with the piston at BDC and ends with it at TDC. The air inside the cylinder is compressed due to the upwards motion up the piston while both inlet and exhaust valves are closed. The compression ratios for CI engines are in the range 12-24 which is much higher compared to spark ignition engines. The reason for this may be guessed from the name itself and will be discussed underneath.

Combustion is initiated towards the end of the compression stroke by injecting fuel into the chamber. The extremely high pressure and temperature developed due to the high compression ratio along with the inducted O_2 provides sufficient conditions for the fuel to ignite further increasing the pressure and temperature inside rapidly.

Expansion/Power Starts with the piston at TDC and ends with the piston at BDC. The high temperature and pressure gases push the piston down and force the crank to rotate. Nearly five times as much work is done on the piston during the power stroke as the piston had to do during compression. As the piston approaches BDC the exhaust valve opens.

Exhaust Starts with the piston at BDC and ends with it at TDC. The burnt gases exit the cylinder through the exhaust valve due to the cylinder pressure usually being higher than the exhaust manifold pressure and due to the upward movement of the piston. As the piston approaches TDC the inlet valve opens and a little after TDC the exhaust valve closes. The cycle starts again.

The five events include the above mentioned four along with Combustion which takes places between the Compression and Expansion stroke as described.

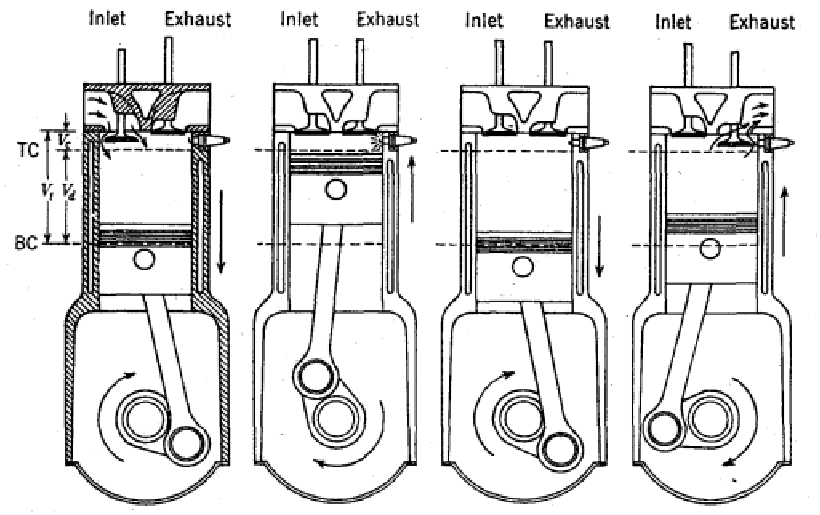

Figure 3: The four-stroke operating cycle [1]

2 Mathematical Modelling

2.1 Some necessary relations

***Available volume** as a function of crank angle [2]:*

$$V_{cyl}(\theta) = V_{disp}\left[\frac{r}{r-1} - \frac{1-cos(\theta-\pi)}{2} + \frac{L}{S} - \frac{1}{2}\sqrt{\left(\frac{2L}{S}\right)^2 - sin^2(\theta-\pi)}\right]$$

$\theta = 0$ is at TDC

***Area of cylinder walls** exposed to gases inside as a function of crank angle [**?**]:*

$$A_{cyl}(\theta) = \frac{\pi B^2}{2} + \frac{\pi BS}{2}\left[1 - cos\theta + \frac{2L}{S} - \sqrt{\left(\frac{2L}{S}\right)^2 - sin^2\theta}\right]$$

***Heat Transfer** to cylinder walls [5]:*

$$\frac{dQ_{ht}}{dt} = A_{cyl}C_{ht}(T_{cyl} - T_{wall})$$

The heat transfer coefficient C_{ht} is given by the Woschini correlation:

$$C_{ht} = 3.26P_{cyl}^{0.8}B^{-0.2}T_{cyl}^{-0.53}m^{0.8}$$

where m is the average cylinder gas velocity which can be approximated as 2.28 times the mean piston speed

$$m = 2.28\frac{\text{RPM} \times S}{30}$$

Van der Waals gas equation:

$$\left(P + \frac{an^2}{V^2}\right)(V - nb) = nRT$$

First Law of Thermodynamics:

$$\frac{dQ}{dt} = P_{cyl}\frac{dV_{cyl}}{dt} + nC_v\frac{dT_{cyl}}{dt}$$

$$\frac{dQ}{dt} = \frac{dQ_{comb}}{dt} - \frac{dQ_{ht}}{dt} + \frac{dQ_{m_{in}}}{dt} - \frac{dQ_{m_{out}}}{dt}$$

There is only heat addition due to combustion during the combustion event and gas exchange during intake and exhaust strokes. Rate of heat addition due to combustion will be discussed later on.

Variable heat constants: From Fig4 and for our given operational temperatures and pressures the heat constant at constant volume is approximated as

$$c_v = 0.2T_{cyl} + 700$$

$$c_p - c_v = R$$

$$c_p = 0.2T_{cyl} + 987$$

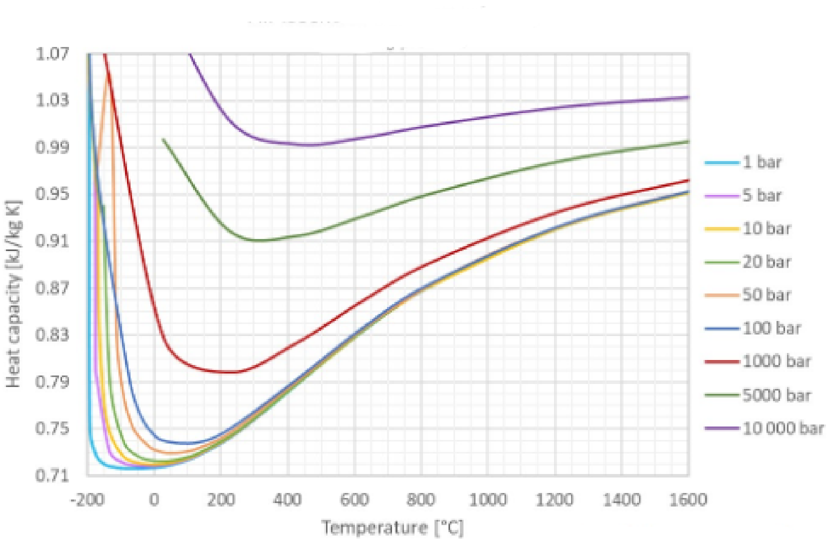

Figure 4: Air isochoric heat capacity c_v [6]

2.2 Intake and Exhaust

Quasi-steady model is used to describe gas flow across the intake and exhaust valve. This model is based on the assumption of one dimensional steady flow for a compressible ideal gas [5].

$$\frac{dm_{iv}}{dt} = A_{iv}\left[l_{iv}(\theta)\right]\varphi\left(P_{cyl}, P_{im}, T_{cyl}, T_{im}\right)$$

$$\frac{dm_{ev}}{dt} = A_{ev}\left[l_{ev}(\theta)\right]\varphi\left(P_{em}, P_{cyl}, T_{em}, T_{cyl}\right)$$

iv - intake valve *ev* - exhaust valve
im - intake manifold *em* - exhaust manifold

A is the effective orifice area which is a function of the lift of the valves which is in turn a function of the crank angle. $l(\theta)$ is manufacture specific.

$$A = \left(\frac{\pi D^2}{4}\right)\left(107.78\left(\frac{l(\theta)}{D}\right)^4 - 77.204\left(\frac{l(\theta)}{D}\right)^3 + 14.1\left(\frac{l(\theta)}{D}\right)^2 - 1.01\left(\frac{l(\theta)}{D}\right) + 0.6687\right)$$

D is the diameter of the valves.
In the following definition of the function φ, u and d refer to upstream and downstream variables [1]:

$$\varphi\left(P_d, P_u, T_d, T_u\right) = \begin{cases} \frac{P_u}{\sqrt{RT_u}}\varphi^+\left(\frac{P_d}{P_u}\right) & P_d \leq P_u \\ \frac{P_d}{\sqrt{RT_d}}\varphi^+\left(\frac{P_u}{P_d}\right) & P_d > P_u \end{cases}$$

$$\varphi^+(x) = \begin{cases} \gamma^{\frac{1}{2}}\left(\frac{2\gamma}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & x \leq c_r \\ x^{\frac{1}{\gamma}}\sqrt{\frac{2\gamma}{\gamma-1}\left(1 - x^{\frac{\gamma-1}{\gamma}}\right)} & x > c_r \end{cases}$$

$$\textit{critical ratio: } c_r = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

The flow is assumed to be subsonic when the pressure ratio is lower than the critical ratio c_r and supersonic otherwise.

The amount of air inducted at any instant can be calculated from the mass flow rate with the help of the above set of equations. Since we know how the volume changes with time or θ ($d\theta = dt \times \omega$) we can find how the Pressure evolves from the first law of thermodynamics which will be shown in the next section.
Also [5],

$$\frac{dQ_m}{dt} = c_p\frac{dm}{dt}T_{cyl}$$

2.3 Compression and Expansion

Combining the Van der Waals gas equation and the first law of thermodynamics

$$\frac{dP_{cyl}}{dt} = \frac{R}{c_v} \times \frac{\frac{dQ}{dt} - \frac{dV_{cyl}}{dt}\left(P_{cyl}\left(1 + \frac{c_v}{R}\right) + 2abn^3\frac{c_v}{RV_{cyl}^3} - an^2\frac{c_v}{RV_{cyl}^2}\right)}{V - nb}$$

$\frac{dQ}{dt}$ for compression and expansion only involves Heat transfer to the cylinder walls. For intake and exhaust it'll have a $\frac{dQ_m}{dt}$ term as well. For combustion $\frac{dQ_{comb}}{dt}$ will have to be considered.

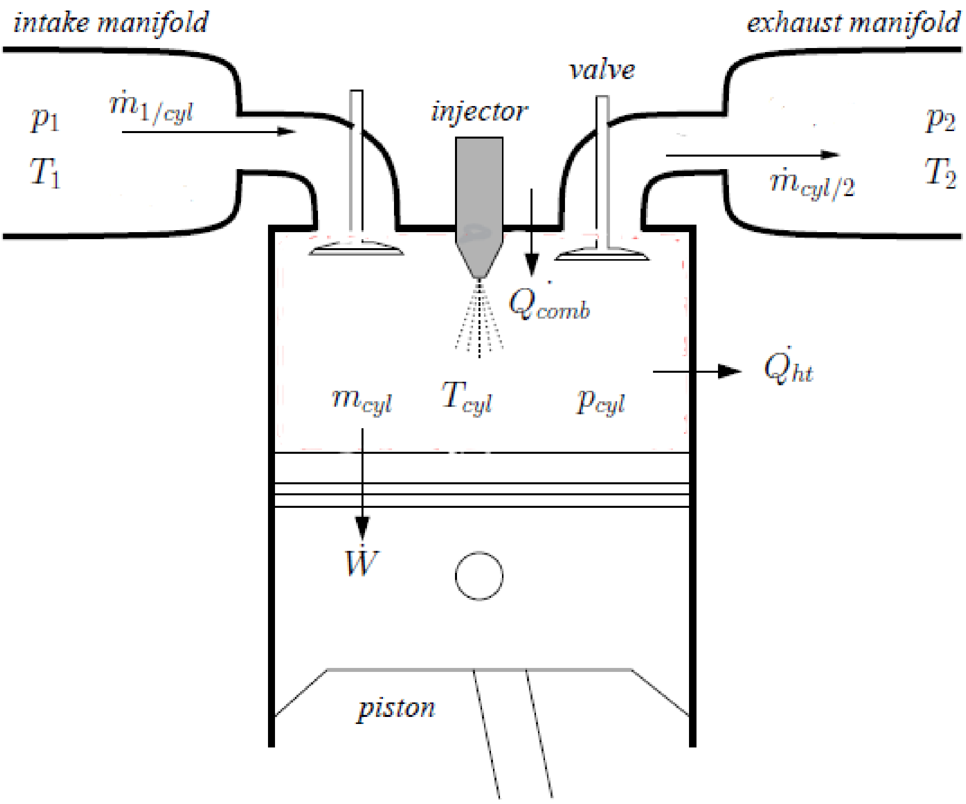

Figure 5: Diagram showing gas exchange and heat addition and losses

2.4 Combustion

We can determine the rate of heat addition to the cylinder from ***Wiebe Heat release model*** [2] in the form of $J/°CA$

$$\frac{dQ_{comb}}{dt} = a(m+1)\left(\frac{Q_{av}}{\Delta\theta_c}\right)^m\left(\frac{\theta-\theta_i}{\Delta\theta_c}\right)^m exp\left[-a\left(\frac{\theta-\theta_i}{\Delta\theta_c}\right)^{m+1}\right]$$

where,
$\Delta\theta_c$ duration of combustion in °CA
θ crank angle at any instant
θ_i crank angle at start of combustion
Q_{av} heat released per cycle in J
a parameter that determines the completeness of combustion
m parameter that determines the rate of combustion

The ignition delay can be determined using a relation derived by *Jensen Samuel and Ramesh* [4]:

$$\tau_{id}(ms) = k * P_{cyl}^{-n} * exp\left(\frac{E}{RT_{cyl}}\right)$$

3 Engine and Modelling Parameters

3.1 Engine dimensions

- Stroke $S = 90mm$
- Length of connecting rod $L = S \times 1.75 = 157.5mm$
- Compression ratio $r = 18.5$
- Bore $B = 80mm$
- Maximum Lift of valves = $4.5mm$
- Valve diameter = $30mm$
- Displacement volume $V_d \approx 497.3cc$
- Clearance volume $V_c \approx 28.4cc$

3.2 Operational conditions

- RPM = 3500↔5000
- Intake valve open before TDC = 15° (IVO)
- Intake valve close after BDC = 20° (IVC)
- Exhaust valve open before BDC = 20° (EVO)
- Exhaust valve close after TDC = 15° (EVC)
- Fuel injection starts before TDC = 5° (FIS)
- Fuel injection stops after TDC = 25° (FIC)
- Spray duration = 30°
- Lower Heating Value of fuel (Diesel) = $LHV = 42.588MJ/kg$
- Stoichiometric Air-Fuel ratio = $14.5M/M$, Operating AF ratio = $20.3M/M$

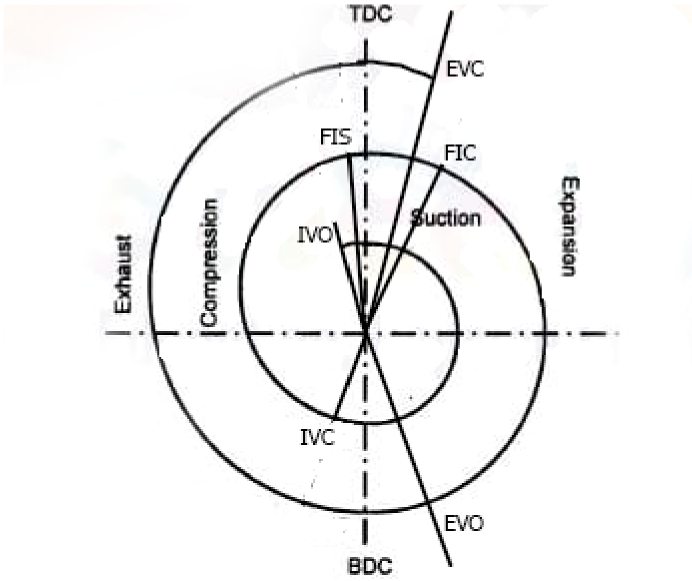

Figure 6: Valve and injector timing diagram

3.3 Modelling Parameters

- *van der Waals constants:*
  - $a = 0.137$
  - $b = 0.0387 \times 10^{-3}$
- $l(\theta)$ = max lift = 4.5mm
- *Wiebe heat release model:*
  - $a = 6.908$ which is equivalent to 99% of the fuel burning
  - $m = 2.36$
  - $Q_{av} = m_{fuel} \times LHV$
- τ_{id} calculation:
  - $k = 2.644$
  - $n = 0.248$
  - $E/R = 2825$
- Molecular weight of air = 29.87amu

4 Simulation and Results

4.1 Simulation according to model

4.1.1 Intake

```python
theta_i = np.arange(-IVO,pi + IVC + dtheta, dtheta)
N_i = len(theta_i)
P_i = np.zeros(N_i)
P_i=np.zeros(N_i)
dP_i=np.zeros(N_i)
T_i=np.zeros(N_i)
V_i=Vd*((r/(r-1)) - ((1-np.cos(theta_i - pi))/2) + (L/S) - 0.5*np.sqrt((2*L/S)**2 - (np.sin(theta_i - pi))**2))
dV_i=np.gradient(V_i,dtheta)
m_i=np.zeros(N_i)
avg_cyl_gas_velocity=2.28*CM
dm_i=np.zeros(N_i)
Twall=400
T_im=300
P_m=20000
P_im=101325
A_i= (pi*Bore**2)/2 + (pi*Bore*S/2)*(1 - np.cos(theta_i) + (L/Rad) - np.sqrt((L/Rad)**2 - (np.sin(theta_i))**2))
P_i[0]=P_m
T_i[0]=T_im
m_i[0]=(P_i[0]*V_i[0])/(T_i[0]*(Ru/MWair))
for i in range(0,N_i-1):
    cv = 0.2*T_i[i] + 700
    cp = R+cv
    gamma=cp/cv
    CD=(107.78*((LIFT_MAX/D)**4))-(77.204*((LIFT_MAX/D)**3))+(14.1*((LIFT_MAX/D)**2))-(1.01*(LIFT_MAX/D))+0.6687
    CURTAREA=(pi/4)*D**2
    Cht=3.26*(10**-3)*(P_i[i]**(0.8))*(T_i[i]**(-0.55))*(Bore**(-0.2))*(avg_cyl_gas_velocity**(0.8))
    if P_i[i]<=P_im:
        if (P_i[i]/P_im)<=(2/(gamma+1))**(gamma/(gamma-1)):
            dm_i[i]=(P_im/(np.sqrt(R*T_im)))*((np.sqrt(gamma))*((2*gamma/(gamma+1))**((gamma+1)/2*(gamma-1))))*CD*CURTAREA/W
            m_i[i+1]=(dtheta*dm_i[i])+m_i[i]
        else:
            dm_i[i]=(P_im/(np.sqrt(R*T_im)))*(((P_i[i]/P_im)**(1/gamma))*np.sqrt((2*gamma/(gamma-1))*(1 - (P_i[i]/P_im)**((gamma-1)/gamma))))*CD*CURTAREA/W
            m_i[i+1]=(dtheta*dm_i[i])+m_i[i]
        n=m_i[i]/MWair
        dP_i[i] = (W*dm_i[i]*cp*T_i[i]-A_i[i]*Cht*(T_i[i]-Twall) - W*dV_i[i]*(P_i[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_i[i])**3)) - a*n*n*cv/(R*((V_i[i])**2))))/(W*(V_i[i]*cv/(R) - cv*b*n/R))
        P_i[i+1]=(dtheta*dP_i[i])+P_i[i]
        T_i[i+1]=(P_i[i+1]*V_i[i+1])/(m_i[i+1]*(Ru/MWair))
    else:
        if (P_im/P_i[i])<=(2/(gamma+1))**(gamma/(gamma-1)):
            dm_i[i]=-(P_i[i]/np.sqrt(R*T_i[i]))*((np.sqrt(gamma))*((2*gamma/(gamma+1))**((gamma+1)/2*(gamma-1))))*CD*CURTAREA/W
            m_i[i+1]=(dtheta*dm_i[i])+m_i[i]
        else:
            dm_i[i]=(P_i[i]/np.sqrt(R*T_i[i]))*(((P_im/P_i[i])**(1/gamma))*np.sqrt((2*gamma/(gamma-1))*(1 - (P_im/P_i[i])**((gamma-1)/gamma))))*CD*CURTAREA/W
            m_i[i+1]=(dtheta*dm_i[i])+m_i[i]
        n=m_i[i]/MWair
        dP_i[i] = (W*dm_i[i]*cp*T_i[i] - A_i[i]*Cht*(T_i[i]-Twall)- W*dV_i[i]*(P_i[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_i[i])**3)) - a*n*n*cv/(R*((V_i[i])**2))))/(W*(V_i[i]*cv/(R) - cv*b*n/R))
        P_i[i+1]=(dtheta*dP_i[i])+P_i[i]
        T_i[i+1]=(P_i[i+1]*V_i[i+1])/(m_i[i+1]*(Ru/MWair))
mass_of_air_inducted = m_i[N_i-1]
```

Figure 7: Python code for simulating the intake procedure

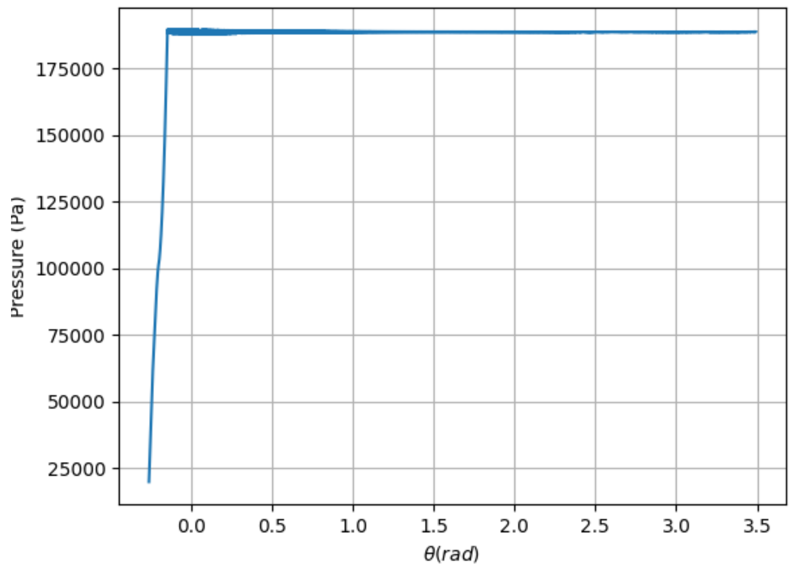

Figure 8: Pressure(Pa) vs crank angle (rad) during intake stroke

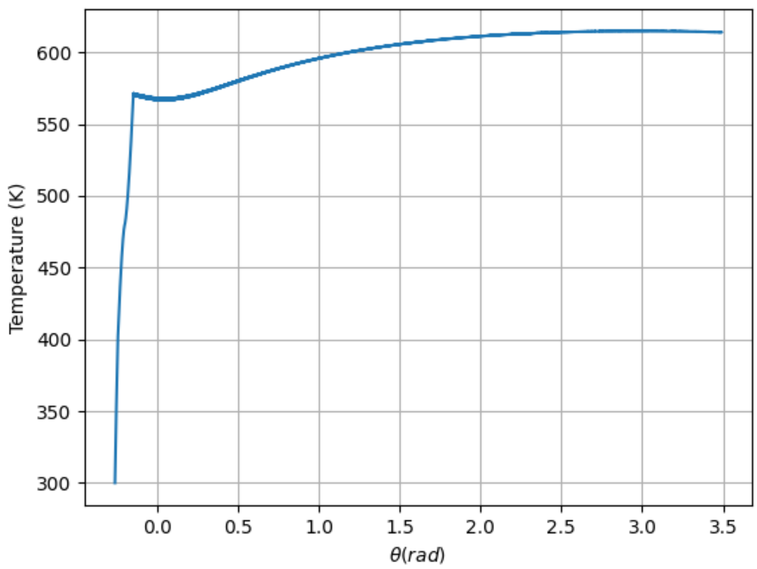

Figure 9: Temperature(K) vs crank angle (rad) during intake stroke

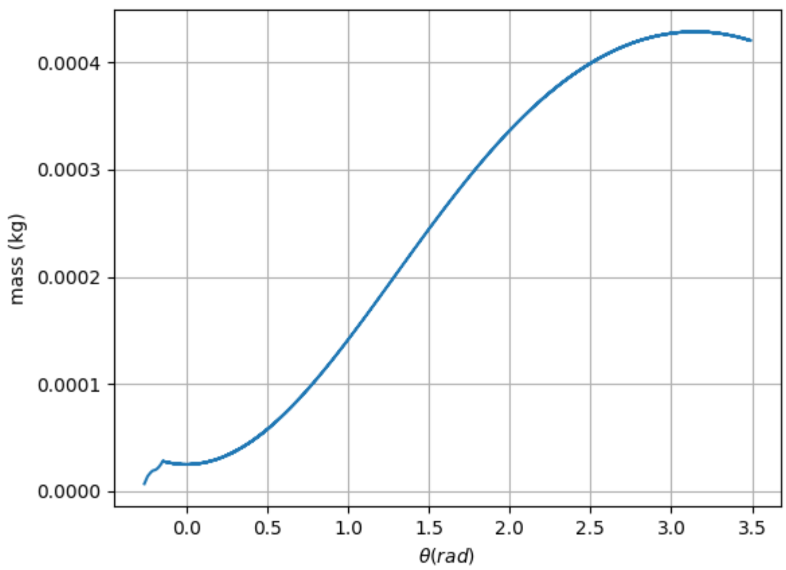

Figure 10: Mass of air inducted (kg) vs crank angle (rad) during intake stroke

At 5000 RPM:

$$\text{Final pressure} = 188658.768\ Pa = 1.886\ bar$$
$$\text{Final temperature} = 614.021\ K$$
$$\text{mass of air inducted} = 0.4202\ g$$

4.1.2 Compression

```python
theta_c = np.arange(pi + IVC,2*pi - FIS + dtheta, dtheta)
N_c = len(theta_c)
P_c = np.zeros(N_c)
P_c=np.zeros(N_c)
dP_c=np.zeros(N_c)
T_c=np.zeros(N_c)
V_c=Vd*((r/(r-1)) - ((1-np.cos(theta_c - pi))/2) + (L/S) - 0.5*np.sqrt((2*L/S)**2 - (np.sin(theta_c - pi))**2))
dV_c=np.gradient(V_c,dtheta)
A_c= (pi*Bore**2)/2 + (pi*Bore*S/2)*(1 - np.cos(theta_c) + (L/Rad) - np.sqrt((L/Rad)**2 - (np.sin(theta_c))**2))
Twall=400
avg_cyl_gas_velocity=2.28*CM
n = m_i[N_i-1]/MWair
T_c[0]=T_i[N_i-1]
P_c[0]=P_i[N_i-1]
for i in range(0,N_c-1):
    cv = 0.2*T_c[i] + 700
    Cht=3.26*(10**-3)*(P_c[i]**(0.8))*(T_c[i]**(-0.55))*(Bore**(-0.2))*(avg_cyl_gas_velocity**(0.8))
    dP_c[i] = -(A_c[i]*Cht*(T_c[i]-Twall) + W*dV_c[i]*(P_c[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_c[i])**3)) - a*n*n*cv/(R*((V_c[i])**2))))/(W*(V_c[i]*cv/(R) - cv*b*n/R))
    P_c[i+1]=(dtheta*dP_c[i])+P_c[i]
    T_c[i+1]=(P_c[i+1]*V_c[i+1])/(Ru*n)
```

Figure 11: Python code for simulating the compression procedure

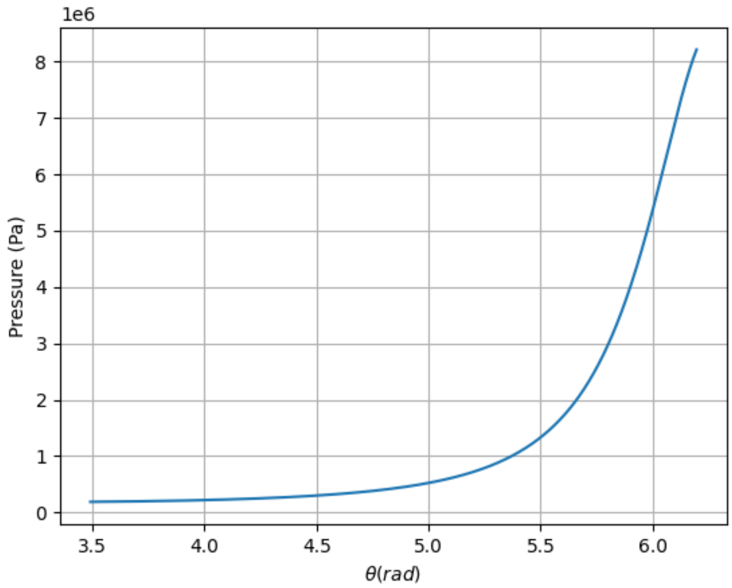

Figure 12: Pressure(Pa) vs crank angle (rad) during compression stroke

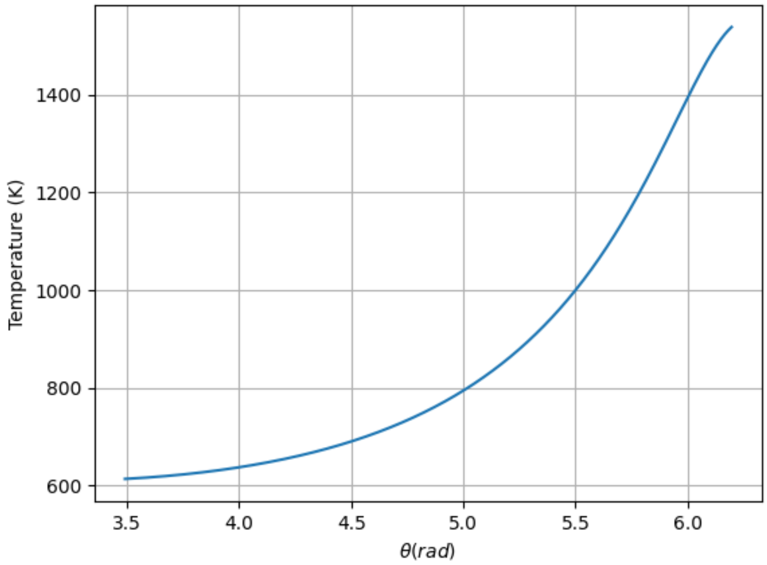

Figure 13: Temperature(K) vs crank angle (rad) during compression stroke

At 5000 RPM:

$$\text{Final pressure} = 8211302.538\ Pa = 82.113\ bar$$
$$\text{Final temperature} = 1537.926\ K$$
$$\text{Ignition delay} = 0.32\ ms = 0.1675\ rad$$

4.1.3 Combustion

```python
theta_b=np.arange(2*pi - FIS,2*pi + FIC +tidr + dtheta,dtheta)
N_b = len(theta_b)
P_b=np.zeros(N_b)
dP_b=np.zeros(N_b)
T_b=np.zeros(N_b)
dqc=np.zeros(N_b)
V_b=Vd*((r/(r-1)) - ((1-np.cos(theta_b - pi))/2) + (L/S) - 0.5*np.sqrt((2*L/S)**2 - (np.sin(theta_b - pi))**2))
dV_b=np.gradient(V_b,dtheta)
A_b= (pi*Bore**2)/2 + (pi*Bore*S/2)*(1 - np.cos(theta_b) + (L/Rad) - np.sqrt((L/Rad)**2 - (np.sin(theta_b))**2))
Twall=1200
avg_cyl_gas_velocity=2.28*CM
n = m_i[N_i-1]/MWair
lam=1.4 #equivalence air-fuel ratio
mfuel=m_i[N_i-1]/(stoichiometric_af_ratio*lam)
qav = LHV*mfuel
ab=6.908
comb_dur=FIC+FIS+tidr
mb=2.36
T_b[0]=T_c[N_c-1]
P_b[0]=P_c[N_c-1]
for i in range(0,N_b-1):
    cv = 0.2*T_b[i] + 700
    Cht=3.26*(10**-3)*(P_b[i]**(0.8))*(T_b[i]**(-0.55))*(Bore**(-0.2))*(avg_cyl_gas_velocity**(0.8))
    dqc[i] = ab*W*(mb+1)*((qav*pi/(180*comb_dur))**mb)*(((theta_b[i] - (2*pi-FIS))/comb_dur)**mb)*np.exp(-ab*(((theta_b[i] - (2*pi-FIS))/comb_dur)**(mb+1)))
    dP_b[i] = (dqc[i] - A_b[i]*Cht*(T_b[i]-Twall) - W*dV_b[i]*(P_b[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_b[i])**3)) - a*n*n*cv/(R*((V_b[i])**2))))/(W*(V_b[i]*cv/(R) - cv*b*n/R))
    P_b[i+1]=(dtheta*dP_b[i])+P_b[i]
    T_b[i+1]=(P_b[i+1]*V_b[i+1])/(Ru*n)
```

Figure 14: Python code for simulating the combustion procedure

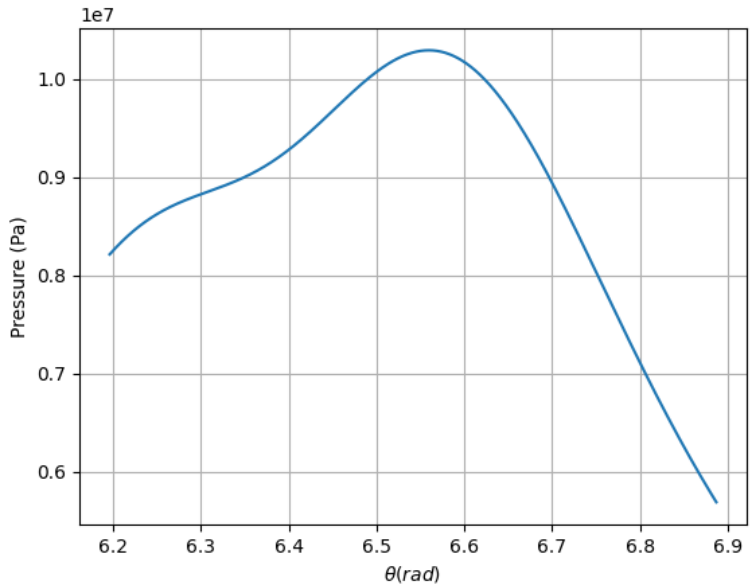

Figure 15: Pressure(Pa) vs crank angle (rad) during combustion stroke

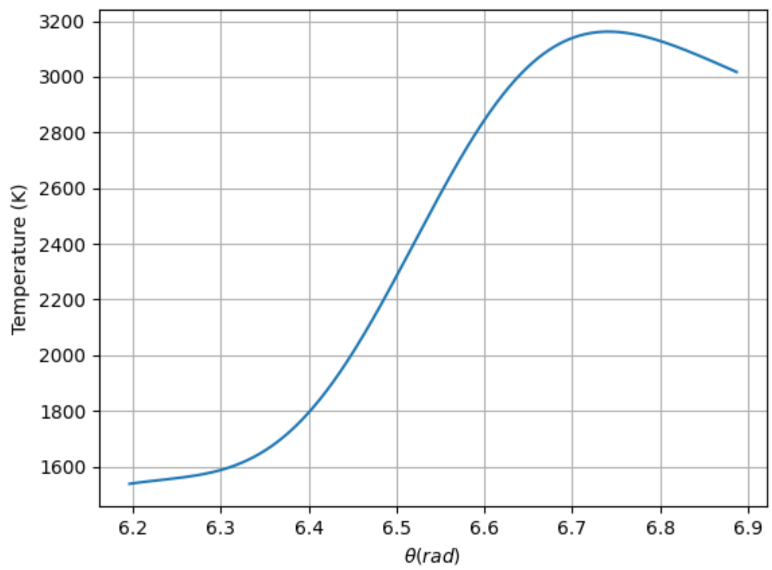

Figure 16: Temperature(K) vs crank angle (rad) during combustion stroke

At 5000 RPM:

$$\text{Final pressure} = 5687039.275\ Pa = 56.870\ bar$$
$$\text{Peak pressure} = 10290136.224\ Pa = 102.901\ bar$$
$$\text{Final temperature} = 3017.321\ K$$
$$\text{Peak temperature} = 3162.172\ K$$

4.1.4 Expansion

The python code is very similar to the compression process, refer to Fig11.

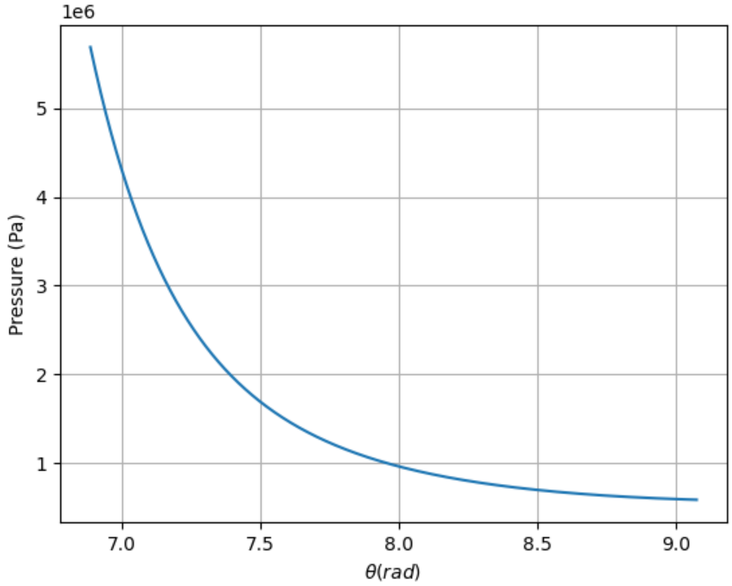

Figure 17: Pressure(Pa) vs crank angle (rad) during expansion stroke

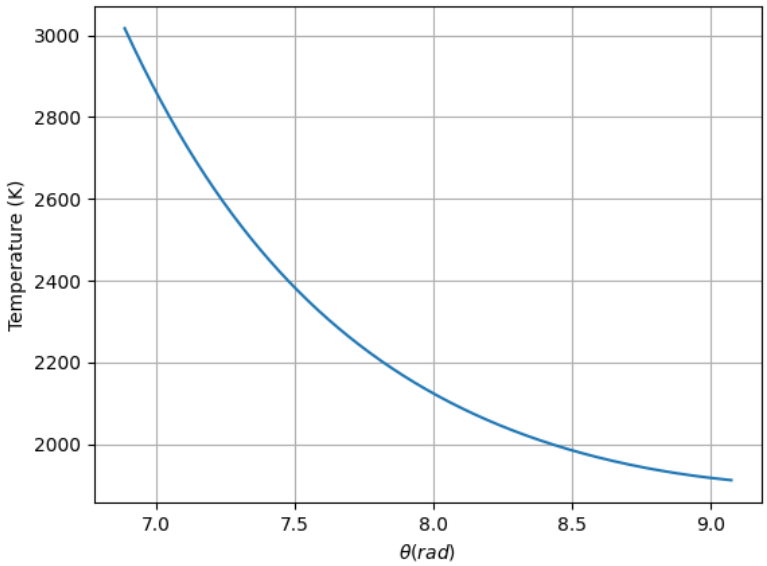

Figure 18: Temperature(K) vs crank angle (rad) during expansion stroke

At 5000 RPM:

$$\text{Final pressure} = 587429.444\ Pa = 5.874\ bar$$
$$\text{Final temperature} = 1911.963\ K$$

4.1.5 Exhaust

```python
theta_ex=np.arange(3*pi - EVO,dtheta+ 4*pi + EVC,dtheta)
N_ex = len(theta_ex)
P_ex=np.zeros(N_ex)
dP_ex=np.zeros(N_ex)
T_ex=np.zeros(N_ex)
V_ex=Vd*((r/(r-1)) - ((1-np.cos(theta_ex - pi))/2) + (L/S) - 0.5*np.sqrt((2*L/S)**2 - (np.sin(theta_ex - pi))**2))
dV_ex=np.gradient(V_ex,dtheta)
m_ex=np.zeros(N_ex)
avg_cyl_gas_velocity=6.18*CM
dm_ex=np.zeros(N_ex)
Twall=700
T_exm=1000
P_exm=101325
A_ex= (pi*Bore**2)/2 + (pi*Bore*S/2)*(1 - np.cos(theta_ex) + (L/Rad) - np.sqrt((L/Rad)**2 - (np.sin(theta_ex))**2))
P_ex[0]=P_e[N_e-1]
T_ex[0]=T_e[N_e-1]
m_ex[0]=m_i[N_i-1]

for i in range(0,N_e-1):
    cv = 0.2*T_ex[i] + 700
    cp = R+cv
    gamma=cp/cv
    CD=(107.78*((LIFT_MAX/D)**4))-(77.204*((LIFT_MAX/D)**3))+(14.1*((LIFT_MAX/D)**2))-(1.01*(LIFT_MAX/D))+0.6687
    CURTAREA=(pi/4)*D**2
    Cht=3.26*(10**-3)*(P_ex[i]**(0.8))*(T_ex[i]**(-0.55))*(Bore**(-0.2))*(avg_cyl_gas_velocity**(0.8))
    if P_exm<=P_ex[i]:
        if (P_exm/P_ex[i])<=(2/(gamma+1))**(gamma/(gamma-1)):
            dm_ex[i]=-(P_ex[i]/(np.sqrt(R*T_ex[i])))*((np.sqrt(gamma))*((2*gamma/(gamma+1))**((gamma+1)/2*(gamma-1))))*CD*CURTAREA/W
            m_ex[i+1]=(dtheta*dm_ex[i])+m_ex[i]
        else:
            dm_ex[i]=(P_ex[i]/(np.sqrt(R*T_ex[i])))*(((P_exm/P_ex[i])**(1/gamma))*np.sqrt((2*gamma/(gamma-1))*(1 - (P_exm/P_ex[i])**((gamma-1)/gamma))))*CD*CURTAREA/W
            m_ex[i+1]=(dtheta*dm_ex[i])+m_ex[i]
        n=m_ex[i]/MWair
        dP_ex[i] = ((W*dm_ex[i]*cp*T_ex[i])-A_ex[i]*Cht*(T_ex[i]-Twall)-W*dV_ex[i]*(P_ex[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_ex[i])**3)) - a*n*n*cv/(R*((V_ex[i])**2))))/(W*(V_ex[i]*cv/(R) - cv*b*n/R))
        P_ex[i+1]=(dtheta*dP_ex[i])+P_ex[i]
        T_ex[i+1]=(P_ex[i+1]*V_ex[i+1])/(m_ex[i+1]*(Ru/MWair))
    else:
        if (P_ex[i]/P_exm)<=(2/(gamma+1))**(gamma/(gamma-1)):
            dm_ex[i]=(P_exm/np.sqrt(R*T_exm))*((np.sqrt(gamma))*((2*gamma/(gamma+1))**((gamma+1)/2*(gamma-1))))*CD*CURTAREA/W
            m_ex[i+1]=(dtheta*dm_ex[i])+m_ex[i]
        else:
            dm_ex[i]=(P_exm/np.sqrt(R*T_exm))*(((P_ex[i]/P_exm)**(1/gamma))*np.sqrt((2*gamma/(gamma-1))*(1 - (P_ex[i]/P_exm)**((gamma-1)/gamma))))*CD*CURTAREA/W
            m_ex[i+1]=(dtheta*dm_ex[i])+m_ex[i]
        n=m_ex[i]/MWair
        dP_ex[i] = (W*dm_ex[i]*cp*T_ex[i]-W*dV_ex[i]*(P_ex[i]*(1 + cv/(R)) + 2*a*b*n*n*n*cv/(R*((V_ex[i])**3)) - a*n*n*cv/(R*((V_ex[i])**2))))/(W*(V_ex[i]*cv/(R) - cv*b*n/R))
        P_ex[i+1]=(dtheta*dP_ex[i])+P_ex[i]
        T_ex[i+1]=(P_ex[i+1]*V_ex[i+1])/(m_ex[i+1]*(Ru/MWair))
```

Figure 19: Python code for simulating the exhaust procedure

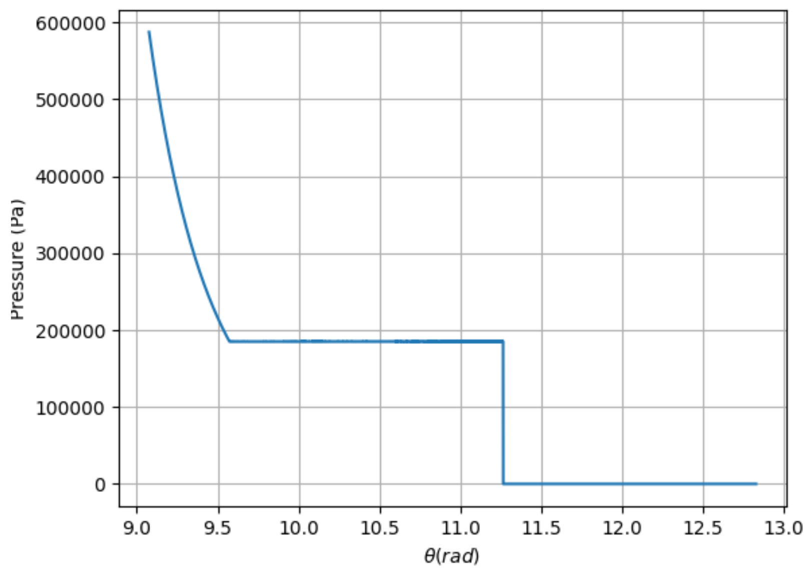

Figure 20: Pressure(Pa) vs crank angle (rad) during exhaust stroke

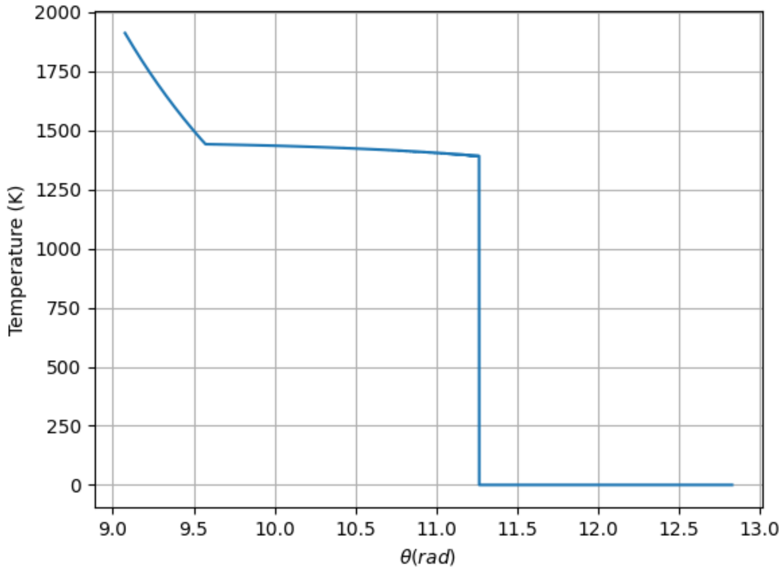

Figure 21: Temperature(K) vs crank angle (rad) during exhaust stroke

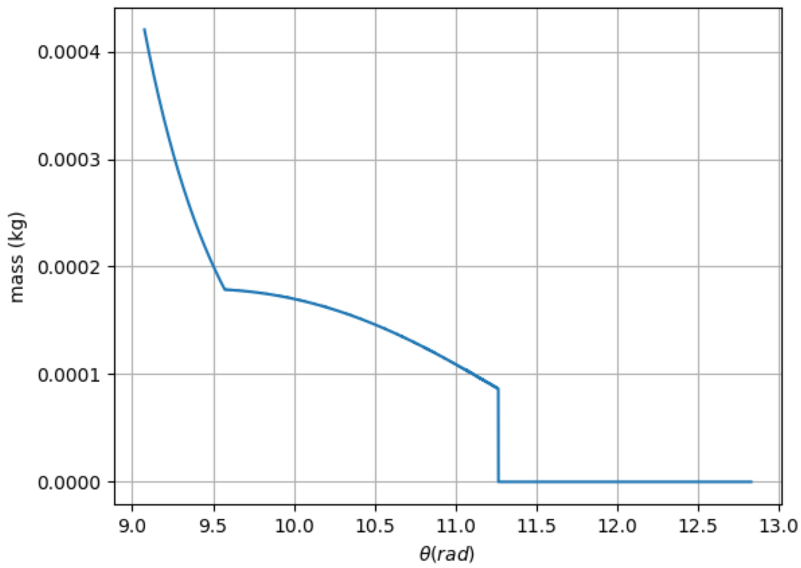

Figure 22: Mass of air inducted (kg) vs crank angle (rad) during exhaust stroke

4.1.6 Indicator Diagrams

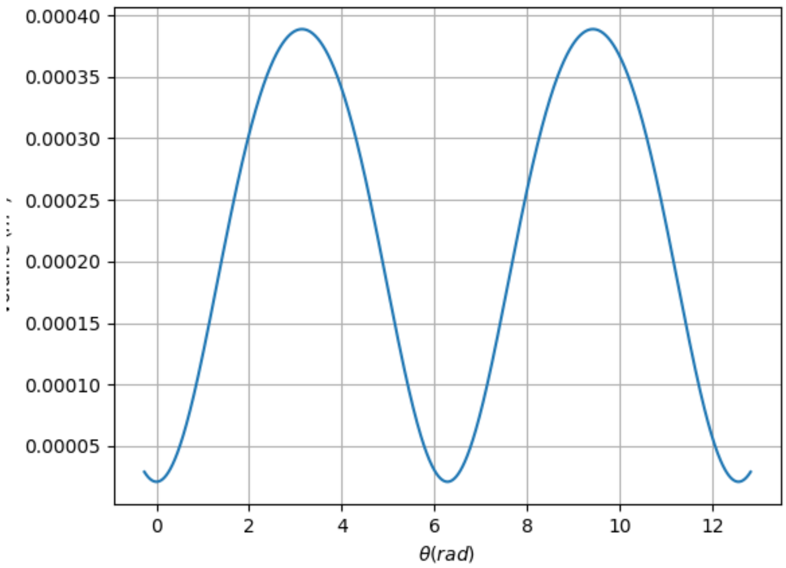

Figure 23: Volume (m^3) vs crank angle (rad) during one cycle

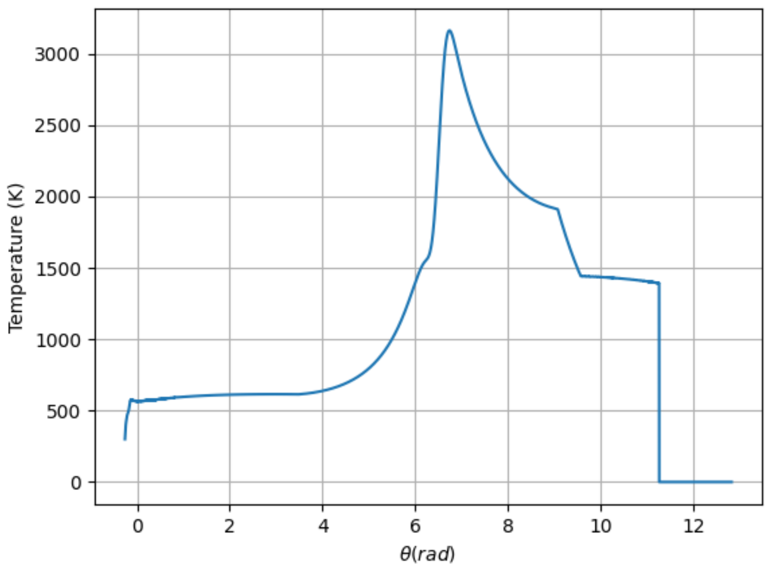

Figure 24: Temperature(K) vs crank angle (rad) during one cycle

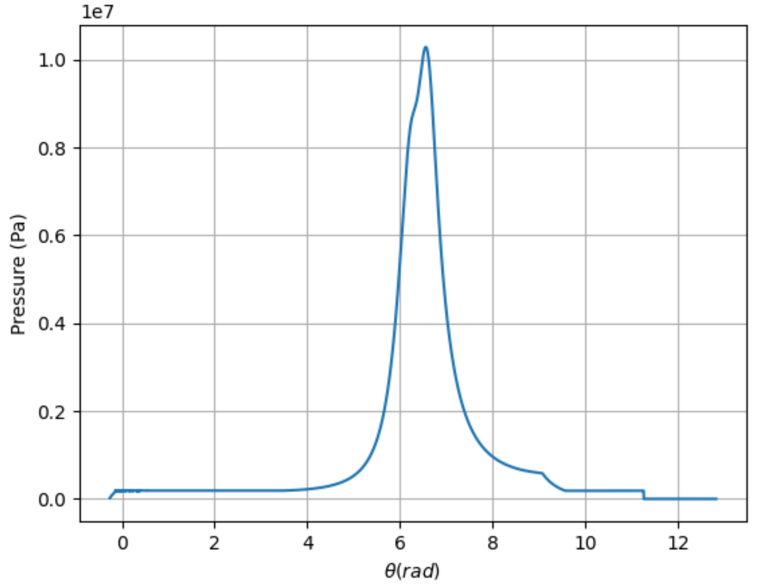

Figure 25: Pressure (Pa) vs crank angle (rad) during one cycle

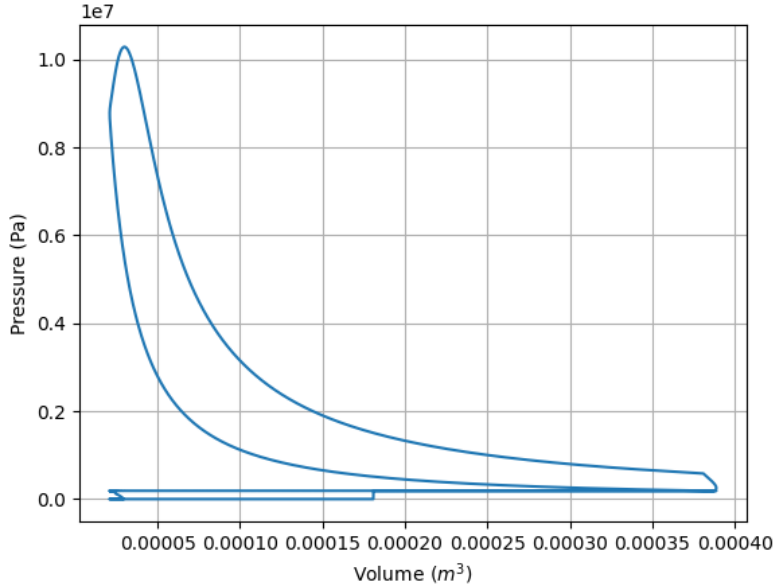

Figure 26: Pressure (Pa) vs Volume (m^3) during one cycle

At 5000 RPM:

$$\text{Total Work done (TWD)} = 549.126\ J$$
$$\text{Indicated Mean Effective Pressure (IMEP)} = \frac{W}{V_d} = 1493376.852\ Pa = 14.933\ bar$$
$$\text{Indicated Power (IP)} = 22880.257\ W = 30.68\ \text{HP}$$
$$\text{Indicated Torque (IT)} = 43.703\ Nm$$

4.1.7 Without Combustion

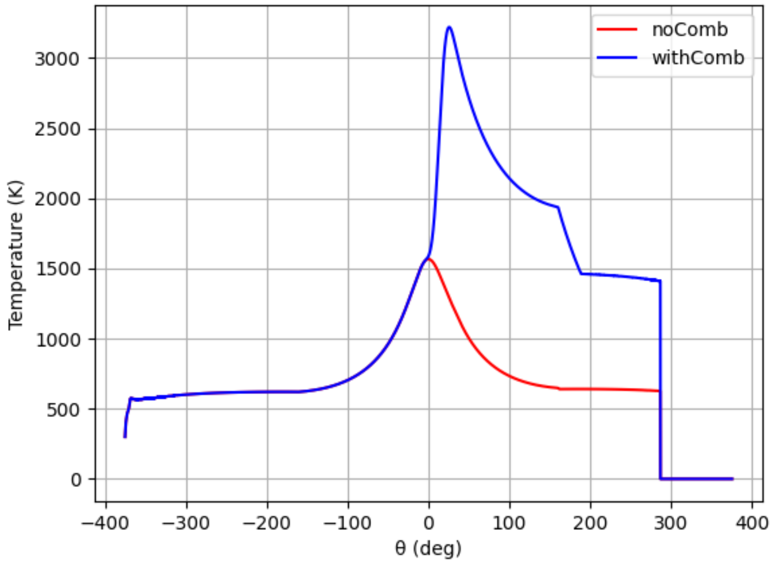

Figure 27: Temperature(K) vs crank angle (rad) during one cycle

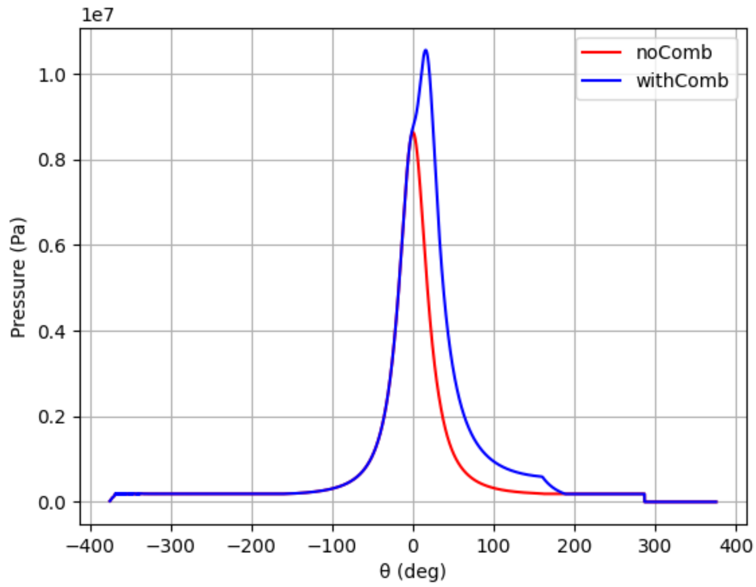

Figure 28: Pressure (Pa) vs crank angle (rad) during one cycle

0° is TDC only in this subsection.

4.2 Simulation using Volumetric Efficiency

The amount of air inducted into the cylinder is a major deciding factor on multiple outputs. It is also the very first event to occur which like a chain reaction effects the subsequent events. The ”breath-ability” or Volumetric Efficiency in Compression Ignition is heavily dependent on operating RPM and valve timings. The typical volumetric efficiency of CI engine looks somewhat as depicted in Fig29

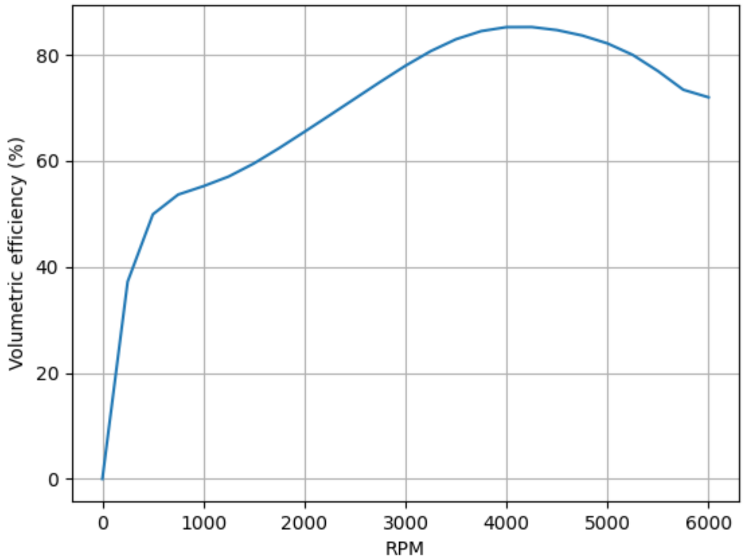

Figure 29: Volumetric Efficiency vs RPM

The following polynomial almost perfectly fits the above curve (N=RPM):

$$VolEff = 9.27\times10^{-30}N^9 - 2.69\times10^{-25}N^8 + 3.33\times10^{-21}N^7 - 2.29\times10^{-17}N^6 + 9.68\times10^{-14}N^5 - 2.57\times10^{-10}N^4 + 4.28\times10^{-07}N^3 - 4.25\times10^{-04}N^2 + 0.231N + 7.85\times10^{-03}$$

Disregarding the intake method using the mentioned fluid dynamic models and calculating the mass of air inducted from the volumetric efficiency method, here are the various results obtained.

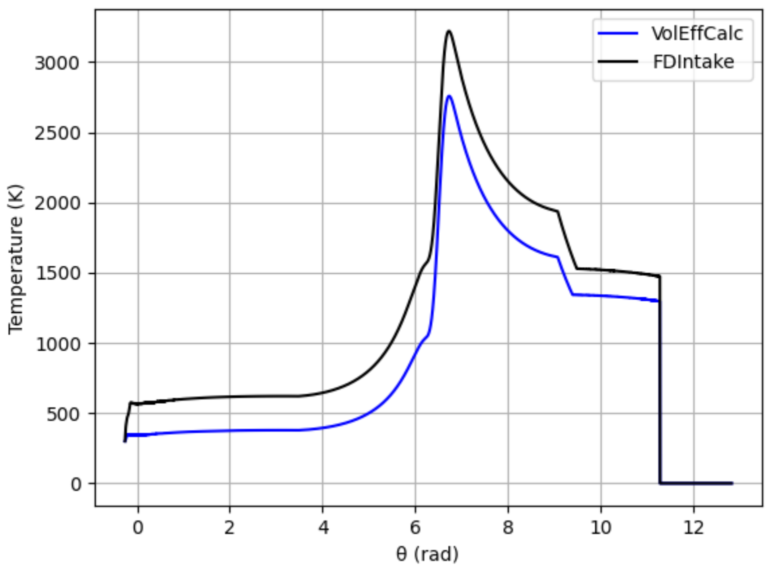

Figure 30: Temperature(K) vs crank angle (rad)

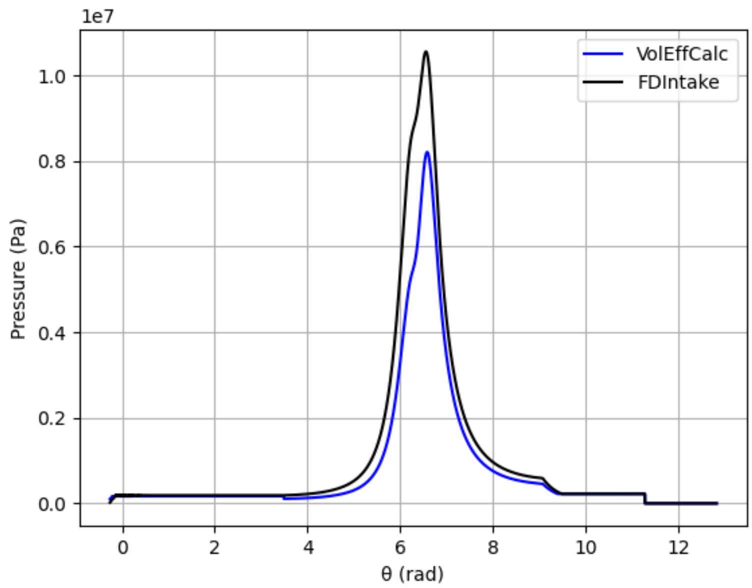

Figure 31: Pressure(Pa) vs crank angle (rad)

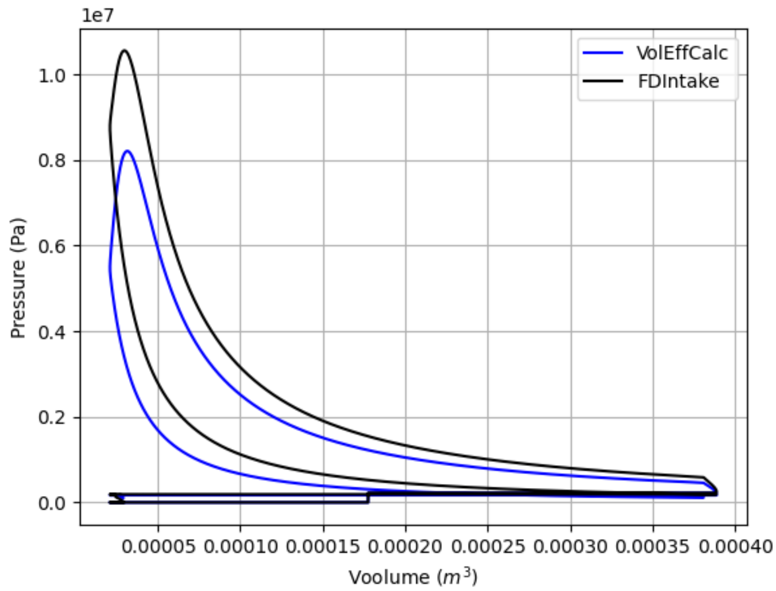

Figure 32: Pressure(Pa) vs Volume (m^3)

@5kRPM , AFR=20.3	air inducted(g)	fuel injected(g)	Peak Pressure(bar)	Peak Temp(K)
FD Intake	0.4202	0.0206	102.901	3162.172
VolEff	0.3904	0.0192	82.07	2758.348

@5kRPM , AFR=20.3	TWD(J)	IMEP(bar)
FD Intake	549.126	14.933
VolEff	492.398	13.391

The shortcomings of the implemented intake mechanism become much more evident when performance outputs are checked over a range of RPM. My reason for mentioning the RPM to be set between 3500 and 5000 also becomes clear here.

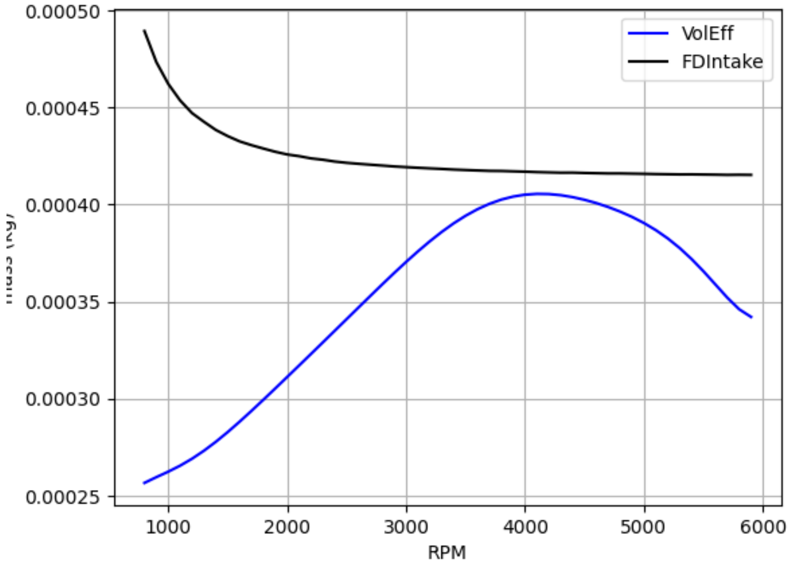

Figure 33: mass of air inducted vs RPM for both the methods. The curves are closest to each other in the RPM range 3500↔5000. This graph is analogous to the volumetric efficiency graphs.

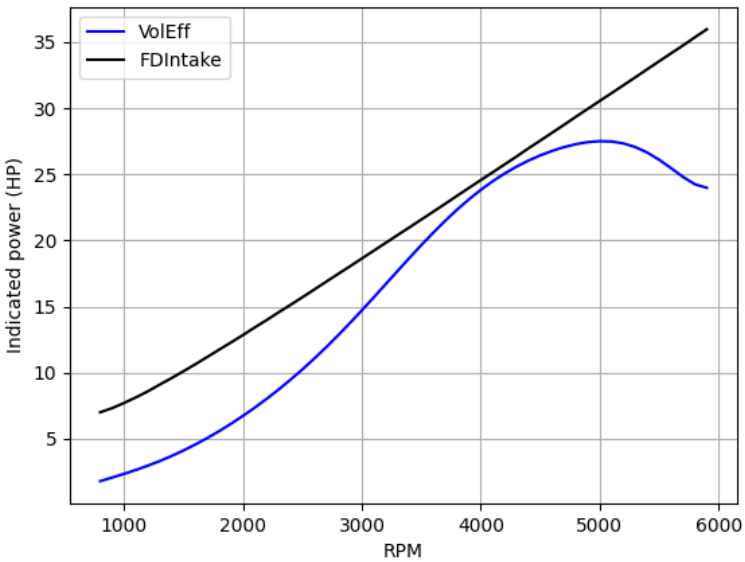

Figure 34: Indicated power (HP) vs RPM

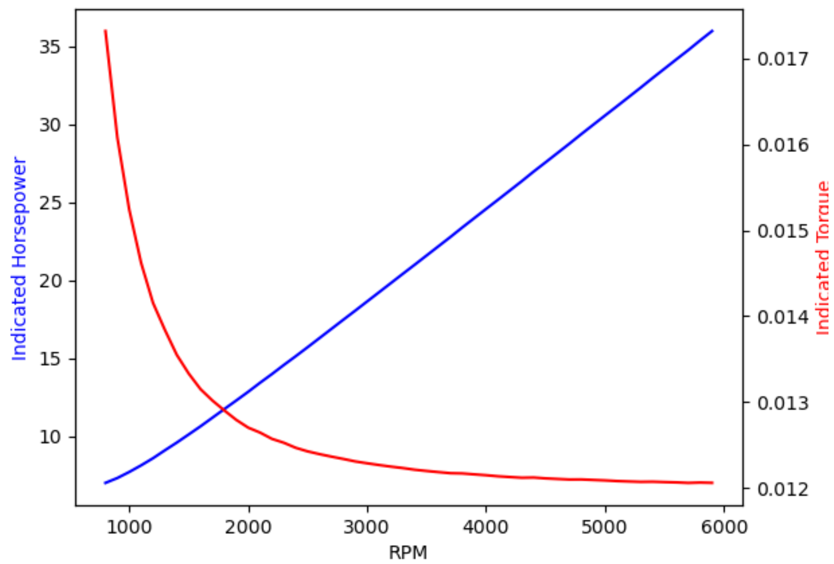

Figure 35: Indicated power and torque for the FD Intake method. Notice how the torque curve has the same shape as the volumetric efficiency curve in Fig33.

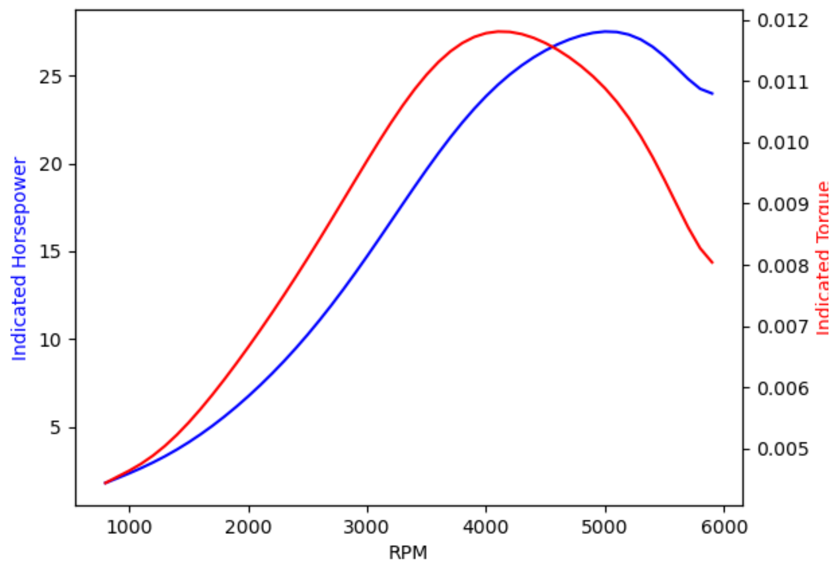

Figure 36: Indicated power and torque for the Volumetric Efficiency method. Once again notice how the torque curve has the same shape as the volumetric efficiency curve in Fig33.

4.3 Varying Fuel Injection Timing

By varying the fuel injection timing we can change at what crank angle the peak pressure is attained. For maximum torque the peak must be attained at around 5° ATDC. This is due to the fact that the volume inside the chamber does not change much between 5° before and after TDC and 5° after TDC the piston starts moving down. Hence, we time our fuel injection accordingly. Here are the results varying injection timing from 20° to 35° BTDC, keeping the spray duration (30°) constant.

FIS BTDC	Peak Pressure (bar)	at °CA
20	1.416	366.092
25	1.578	363.464
30	1.692	361.455
35	1.764	360.311
40	1.818	359.961

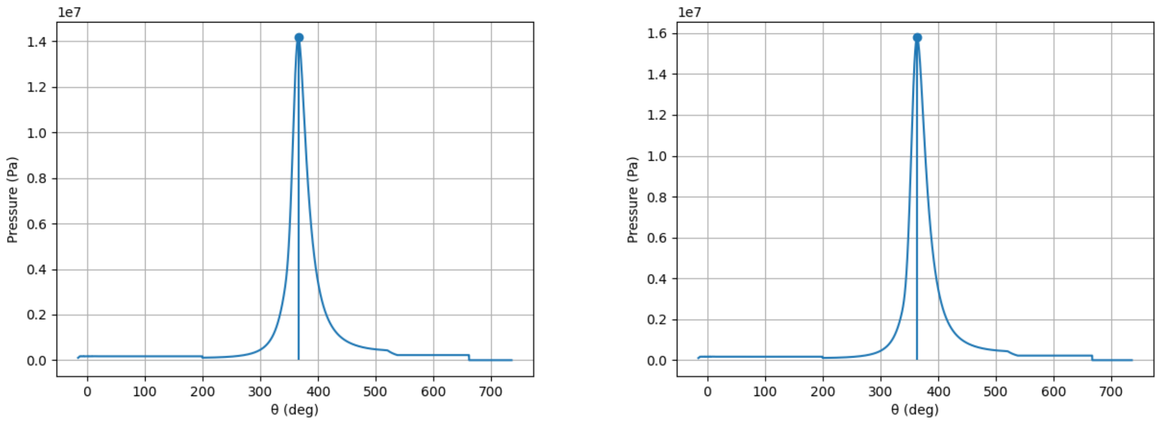

Figure 37: FIS 20° (left) and 25° (right) BTDC

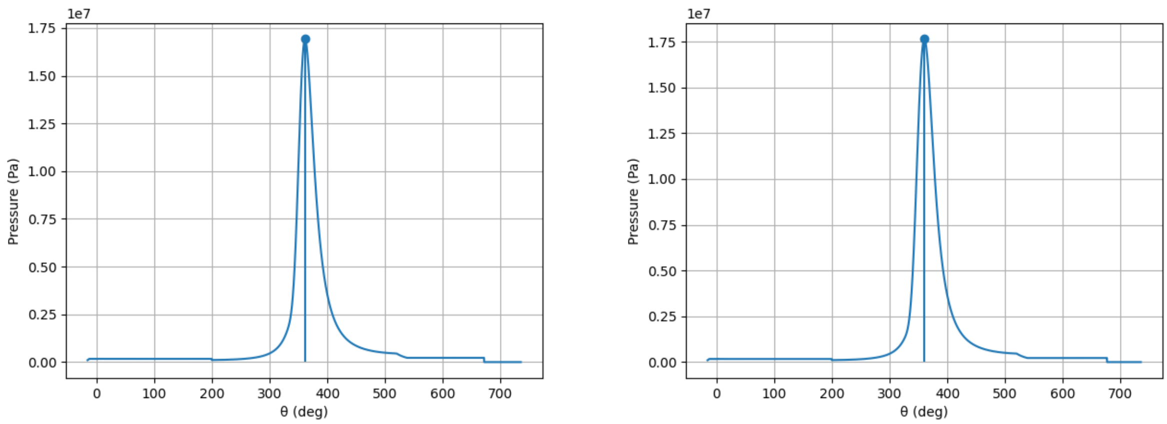

Figure 38: FIS 30° (left) and 35° (right) BTDC

5 Conclusion

The Fluid Dynamic model implemented through code seems to only be closely accurate in the RPM region of 3500-5000. It fails to properly simulate the choking an engine suffers from at extremely low and high rpm.
Due to this the mass of air inducted is higher than expected, resulting in more fuel being injected and different (but fairly similar) graphs when compared to the volumetric efficiency method. This is clearly visible when the engine is simulated over a range of rpm from 1000-6000 to calculate indicated power and torque.
The torque curves have the same shape as volumetric efficiency which can be due to the fact that the indicated mean effective pressure is directly proportional to the amount of air inducted.
The sudden dip in mass of air present inside, pressure and temperature to zero during the exhaust stroke is a clear indicator of a bug in the code which may lead to lower values of total work done and indicated mean effective pressure.
From the no combustion Pressure-theta indicator diagram the ignition delay is visible as the slope of the of graph begins to rise a little after TDC with a fuel injection advance of 5° instead of dropping due to the increasing cylinder volume after TDC.
By increasing the fuel injection advance the crank angle at which the peak pressure was attained was shifted. However it seems that at a 40° advance the peak pressure is attained before TDC. This is an indicator of knocking. Knocking would've been clearly visible in the indicator diagrams if the force on the piston was calculated at every instance, but the rate of change of volume inside the cylinder was hard coded for a given RPM which could also lead to incorrect calculations of torque and power in case of extreme knocking.

All figures and codes used in this report can be found at

https://github.com/adityagangula/CI_Engine_Simulation

References

[1] John B. Heywood, Internal Combustion Engine Fundamentals

[2] V Ganesan, Computer Simulation of Compression-Ignition Processes

[3] V Ganesan, Internal Combustion Engines

[4] Lucian Miron, Radu Chiriac, Marek Brabec, Viorel Bădescu, Ignition delay and its influence on the performance of a Diesel engine operating with different Diesel–biodiesel fuels

[5] Olivier Grondin, J. Maquet, R. Stobart, Houcine Chafouk, Compression ignition engine simulator for instantaneous pressure and torque generation

[6] https://www.engineeringtoolbox.com/air-specific-heat-capacity-d_705.html