Engine testing and HRR analysis

Master's Degree in Automotive Engineering

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Contents

1 Introduction **3**

2 Steady-state test **3**
2.1 Engine power and torque 3
2.2 Volumetric efficiency, bsfc and fuel conversion efficiency 4

3 Combustion diagnostic **5**
3.1 In-cylinder pressure 5
3.2 Start Of Injection (SOI) 7
3.3 Heat Release Rate (HRR) 8
3.4 Mass fraction burned, SOC and EOC 9
3.5 Heat release analysis 10
3.6 In-cylinder pressure comparison 10

4 Conclusions **10**

5 Appendix **12**

1 Introduction

The primary objective of this report is to evaluate the Heat Release Rate of a turbocharged compression ignition engine through experimental analysis. The study involves conducting steady-state tests to measure key engine parameters such as power, torque, volumetric efficiency, brake-specific fuel consumption and fuel conversion efficiency. A dedicated testbed, equipped with a dynamometer, multiple sensors, and a control unit, is used to collect data under controlled conditions.

Additionally, a combustion diagnostic is performed, focusing on in-cylinder pressure measurements, Start of Injection, HRR analysis, and mass fraction burned calculations. These analyses provide insights into the combustion process, ignition delay, and overall engine efficiency.

2 Steady-state test

The first step to be performed for the evaluation of the Heat Release Rate is the calculation of the characteristic parameters of the engine under study. To extrapolate the required data, some experimental tests are carried out through a testbed, featuring three main features: a dynamometer to counteract and measure the engine torque, a variety of sensors to measure variables of interest and a control unit through which the engine, the dyno and all the testbed sub-systems can be monitored and controlled.

2.1 Engine power and torque

To ensure a correct characterization of the engine parameters the measured torque and power need to be corrected to standard atmospheric conditions. This because engine performance are strongly influenced by atmospheric conditions, such as temperature, pressure and humitity of the combustion air. Therefore, to make the results obtained in different days or on different testbeds comparable, correction factor must be considered. In particular, in this report, the power correction is carried out based on the ISO 1585 standard, stating that the power and torque correction is performed by means of a correction factor μ_c that depends on the engine type. Since the engine under study is a turbocharged compression ignition engine, the formula to be used for the evaluation of the correction factor is the following:

$$\mu_c = (f_a)^{f_m} \tag{1}$$

where

$$f_a = \left(\frac{p_{0,dry}}{p_{a,dry}}\right)^{0.7} \cdot \left(\frac{T_a}{T_0}\right)^{1.2} \tag{2}$$

The saturated and dry pressure are evaluated as follows:

$$\begin{cases} p_{sat,H20} = a_0 + a_1 T_a + a_2 T_a^2 + a_3 T_a^3 + a_4 T_a^4 \\ p_{a,dry} = p_a - H_{rel} \cdot p_{sat,H20} \end{cases} \tag{3}$$

where H_{rel} is the relative humidity.

It should be noted that, according to the used standard, the test ambient conditions have to comply with the following constrains:

- 288 K $< T_a <$ 308 K for SI engines
- 283 K $< T_a <$ 313 K for CI engines
- 80 kPa $< p_{a,dry} <$ 100 kPa both for SI and CI engines

The engine's mechanical characteristics, including both the measured values from the dyno and the corrected data, are presented in the following charts.

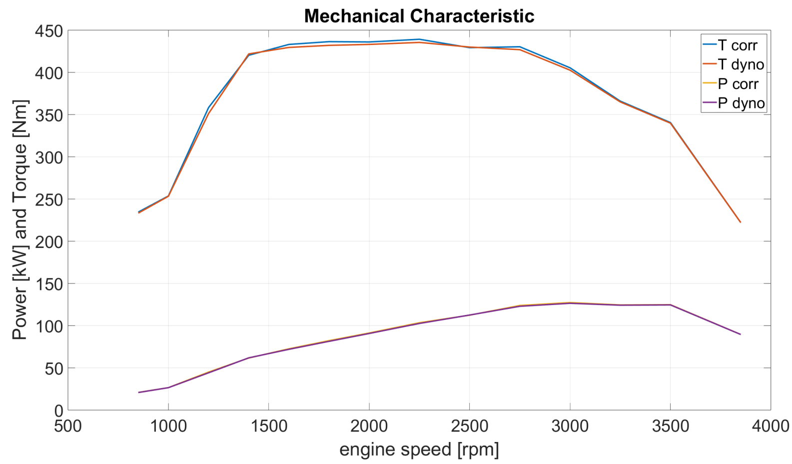

Figure 1: Mechanical characteristic as function of the engine speed

Figure 2: Correction factor as function of the engine speed

2.2 Volumetric efficiency, bsfc and fuel conversion efficiency

Once the engine brake power has been evaluated, others parameters characterizing the engine can be computed. The mentioned parameters are volumetric efficiency, brake specific fuel consumption (bsfc) and fuel conversion efficiency. They can be evaluated according to the following equations:

$$\eta_f = \frac{P_b}{\dot{m}_f \cdot Q_{LHV}} \tag{4}$$

$$bsfc = \frac{\dot{m}_f}{P_b} \tag{5}$$

$$\lambda_v = \frac{m_{int}}{m_{int,ref}} = \frac{m_a + m_{EGR}}{\rho_{int} \cdot V_d} = \frac{m_a + m_{EGR}}{V_d} \cdot \frac{R_{mix} \cdot T_{int}}{p_{int}} \tag{6}$$

where

$$R_{mix} = \frac{\dot{m}_{air} R_{air} + \dot{m}_{EGR} R_{EGR}}{\dot{m}_{air} + \dot{m}_{EGR}} \tag{7}$$

To be noticed that the expression used to compute the volumetric efficiency takes into account the EGR mass, since the engine under study is equipped with this type of system. The obtained quantities are reported as function of the engine speed in the following charts.

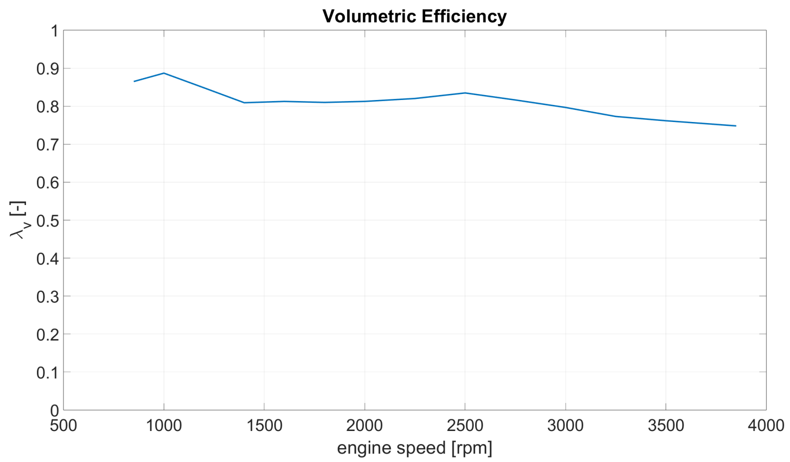

Figure 3: Volumetric efficiency plot as function of the engine speed

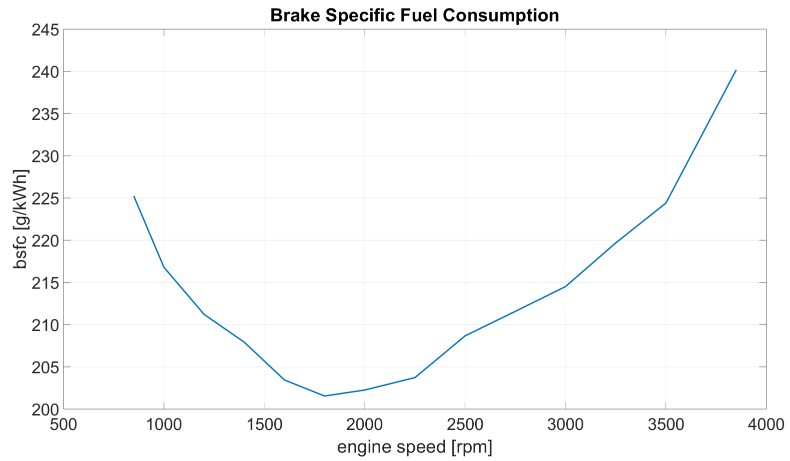

Figure 4: Brake specific fuel consumption as function of the engine speed

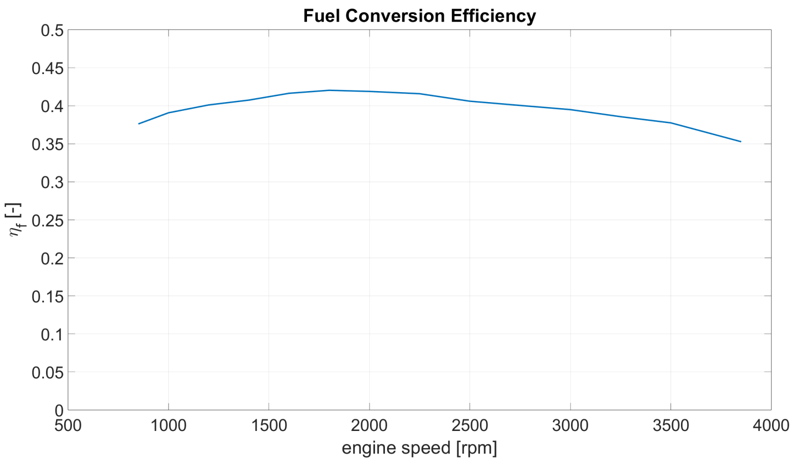

Figure 5: Fuel conversion efficiency as function of the engine speed

3 Combustion diagnostic

3.1 In-cylinder pressure

To evaluate the HRR, it is essential to assess another engine parameter: the in-cylinder pressure. This pressure varies throughout the engine cycle due to several factors, including volume changes caused by piston movement, combustion, heat transfer through the chamber walls, and mass flow through crevices. Nevertheless, in this survey, only the volume changes and the combustion process are considered. Since in-cylinder pressure cannot be determined analytically, it is measured experimentally using high-frequency piezoelectric pressure transducers. These transducers are mounted within specialized glow plug adapters, which are designed to fit into the existing mounting bore of the standard glow plug. Once the in-cylinder pressure as a function of the crank angle is measured and considering negligible the contributions of heat transfer and leakages, pressure variation can be expressed as follows:

$$\Delta p = \Delta p_c + \Delta p_v \tag{8}$$

where Δp_c is the pressure variation due to combustion and Δp_v is the pressure variation due to the piston's movement. The latter term essentially represents the pressure variation in a cycle without combustion. This can be easily evaluated by considering that, in this case, the engine undergoes compression and expansion phases, which can be described through polytropic transformations, yielding:

$$\Delta p_v = p_i \left[\left(\frac{V_i}{V_j} \right)^m - 1 \right] \tag{9}$$

Therefore, for each crank angle interval, the pressure variation caused by the combustion event can be determined as the difference between the measured in-cylinder pressure variation and the pressure variation resulting from piston movement:

$$\Delta p_c = \Delta p - \Delta p_v \tag{10}$$

The in-cylinder pressure functions over 100 measured cycles are reported in Figure 6. To be noted that pressure transducers measure a relative variation of the pressure. Therefore, to evaluate the absolute pressure inside the cylinder, the relative measurement must be pegged by adding a shift to the chart. In particular, the shifting is performed in order to achieve a matching between chamber and manifold pressures aorund the intake opening °CA:

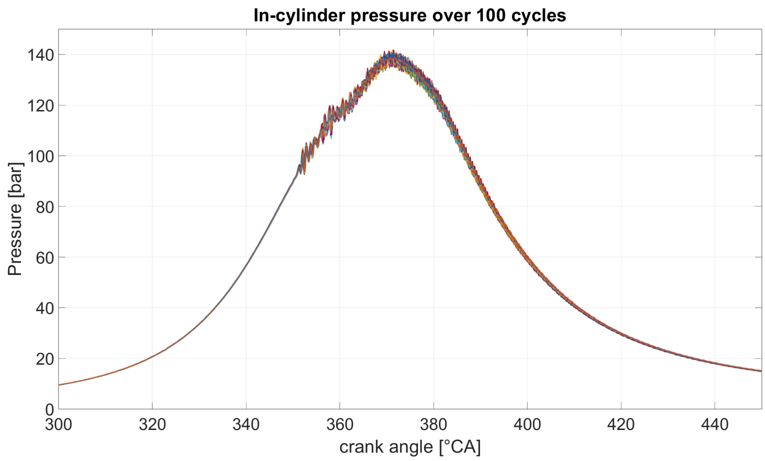

Figure 6: In-cylinder pressure plot over 100 cycles

As expected, the pressure values vary across different cycles and are not constant. Therefore, the average pressure function over 100 cycles must be calculated. The resulting function, plotted against the crank angle, is shown in Figure 7.

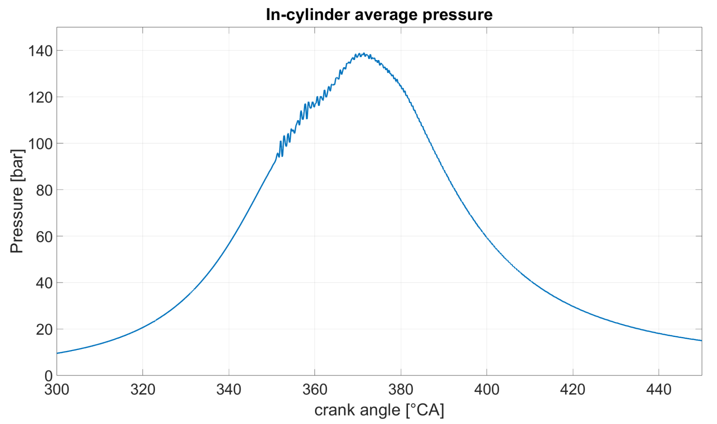

Figure 7: In-cylinder average pressure plot

The evaluation of the average in-cylinder pressure does not yet yield a smooth pressure function; instead, a fluctuating function is obtained. Therefore, a filtering method is required. Specifically, a low-pass filter is applied to attenuate high-frequency components, resulting in a smoother pressure function, as shown in Figure 8.

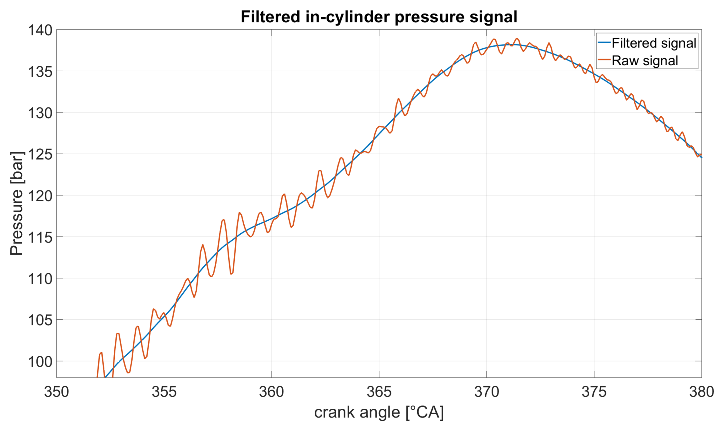

Figure 8: Filtered in-cylinder pressure plot

3.2 Start Of Injection (SOI)

An important parameter characterizing the combustion process in a CI engine is the Start of Injection (SOI). This parameter is determined by analyzing the injection current signal. Specifically, the Start of Injection (SOI) can be identified as the point at which the electrical current to the injectors begins to rise. Figure 9 shows the current signal as function of the crank angle.

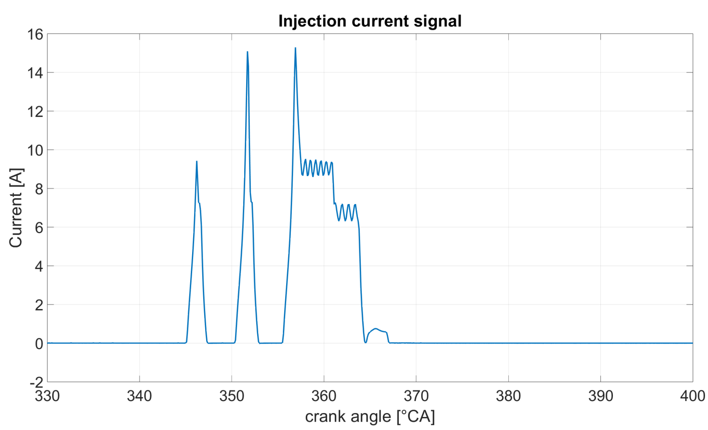

Figure 9: Injection current signal plot

From the chart, the following observations can be made:

$$SOI = 345\ ^{\circ}\mathrm{CA}$$

3.3 Heat Release Rate (HRR)

The evaluation of the heat release rate is performed by applying the First Law of Thermodynamics to the cylinder model. Moreover, the single-zone model is considered, which stands that the gas inside the chamber can be considered homogeneous. Therefore, the net heat release rate can be determined by studying the average thermodynamic state of the cylinder charge. The following equation is obtained:

$$HRR = \frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{11}$$

where γ is a coefficient varying with the temperature according to the following equation:

$$\gamma = 1.338 - 6 \cdot 10^{-5} \cdot T + 1 \cdot 10^{-8} \cdot T^2 \tag{12}$$

being T the temperature along the cycle calculated according to the ideal gas state law as follows:

$$T = \frac{pV}{m_{mix} R_{mix}} \tag{13}$$

The net heat release rate plot as function of the crank angle is reported in Figure 10. Furthermore, Figure 11 shows the relation between HRR and injection current. As expected, the majority of the heat is released during the combustion process, occurring after fuel injection and, consequently, following the rise in the injection current signal.

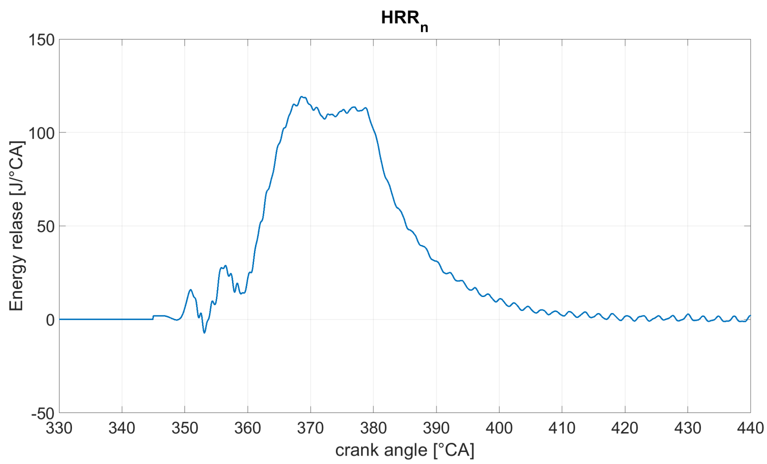

Figure 10: Net heat release rate plot

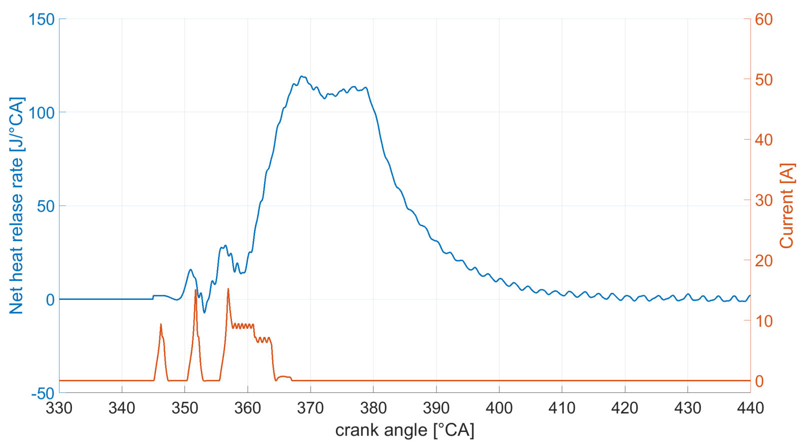

Figure 11: HRRn and injection current relationship

3.4 Mass fraction burned, SOC and EOC

Once the net heat release has been determined, the mass fraction burned (x_b) can be calculated. This parameter is defined as the ratio between the mass of fuel burned at a given crank angle and the total mass of fuel injected. The mass fraction burned is a crucial indicator of the combustion process, as it provides insights into its progression. Since the amount of fuel burned directly correlates with the energy released, x_b helps assess combustion efficiency.

Furthermore, the mass fraction burned allows for the identification of key combustion events. The Start of Combustion (SOC) is defined as the first point where x_b becomes greater than zero, while the End of Combustion (EOC) corresponds to the crank angle at which x_b reaches 1. This parameter is obtained by integrating the heat release rate over the crank angle and normalizing the result within the range of 0 to 1.

Additionally, important combustion characteristics such as MFB10, MFB50, and MFB90 can be evaluated. These values, representing the crank angles at which 10%, 50%, and 90% of the fuel mass has burned, are essential for analyzing combustion efficiency and timing. The mass fraction burned plot is shown in Figure 12.

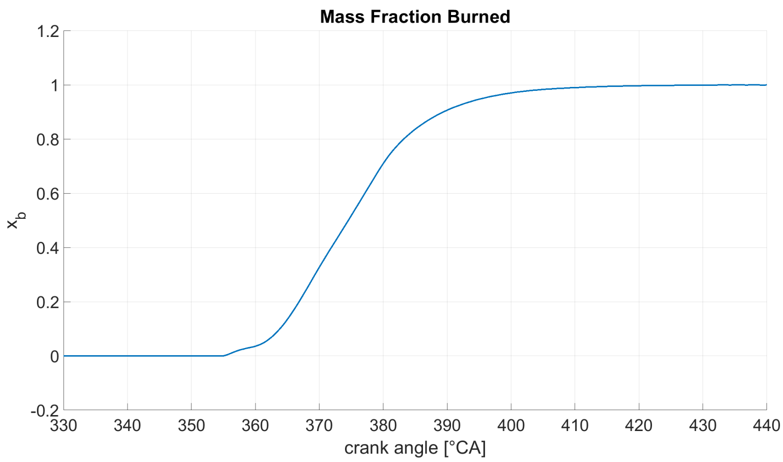

Figure 12: Mass fraction burned as function of the crank angle

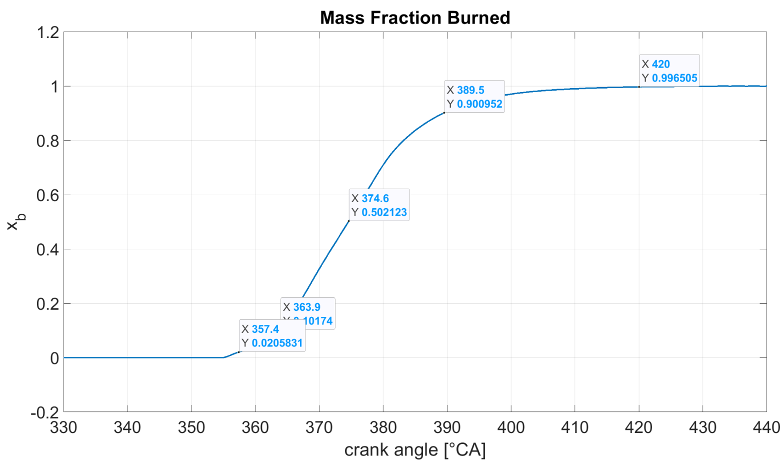

Figure 13: SOC, EOC and MFB values

As shown by the chart's labels in Figure 13, the following values are obtained:

- SOC = 357.4 °CA
- EOC = 369.3 °CA
- $MFB10$ = 374.6 °CA
- $MFB50$ = 389.5 °CA

- $MFB90 = 420$ °CA

Once SOC and SOI are known, the Ignition Delay can be computed as follows:

$$ID = SOC - SOI = 357.4 - 345 = 12.4 \text{ °CA}$$

3.5 Heat release analysis

The final analysis involves the heat release analysis, which relates the previously evaluated net heat release to the initial heat of the fuel. The result is presented in Figure 14. It is evident that not all of the fuel's energy is converted into net heat, primarily due to inefficiencies and the non-ideal nature of the combustion process.

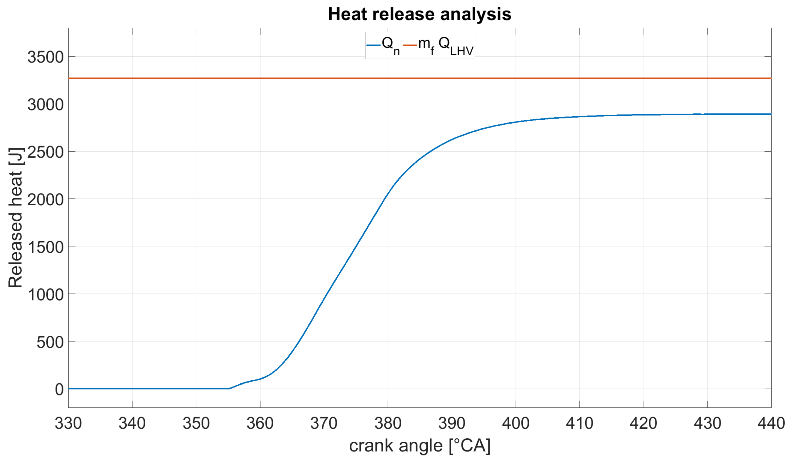

Figure 14: Heat release analysis

3.6 In-cylinder pressure comparison

In this final section, a comparison is made between the motored pressure and the measured pressure, both in their raw and filtered forms. As observed, the measured pressure is higher than the motored pressure. This is because, in addition to volume displacement, the measured pressure also accounts for the combustion process and the corresponding increase in in-cylinder absolute pressure.

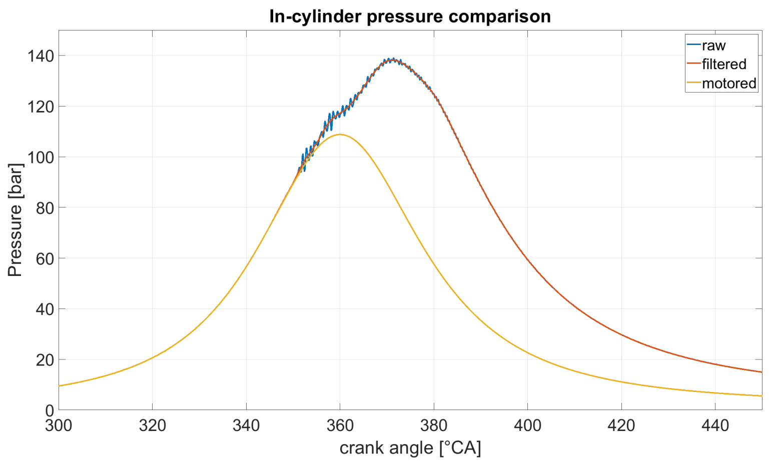

Figure 15: In-cylinder pressure comparison

4 Conclusions

This study successfully analyzed the combustion characteristics of a turbocharged compression ignition engine using experimental methods. The steady-state test results provided valuable insights into engine performance, including power, torque, volumetric efficiency, and fuel conversion efficiency. The combustion diagnostic further detailed the in-cylinder pressure variations, Start of Injection, and heat release processes.

The HRR analysis confirmed that the majority of heat release occurs shortly after fuel injection, following the expected combustion behavior. Additionally, the mass fraction burned analysis enabled the identification of key combustion phases, such as the Start of Combustion (SOC) and End of Combustion (EOC).

5 Appendix

```matlab
clear
close all
clc

load('ifile_2000FL.mat')
filename="datasheet.xlsx";

SS_data = array2table(xlsread(filename,'Steady-state tests','A:J'));
SS_data.Properties.VariableNames = ["n_engine","T_dyno","qm_fuel","qm_air","qm_EGR","p_baro","T_snorkle","Rel_Hum_Air","T_i_MF","p_i_MF"];

working_point = array2table(xlsread(filename,'HRR point','B:B')');
working_point.Properties.VariableNames = ["Engine_speed_rpm","Torque_Nm","Air_mass_flow_rate_kg_h","Fuel_mass_flow_rate_kg_h","qm_egr_kg_h","cp_EGR_J_kgK","QLHV_MJ_kg"];

a = array2table(xlsread(filename,'Steady-state tests','N1:N7')');
a.Properties.VariableNames = ["a0","a1","a2","a3","a4"];

Gas = xlsread(filename,'Steady-state tests','N10:N18')';
Gas = array2table(Gas(~isnan(Gas)));
Gas.Properties.VariableNames = ["Qlhv","Rf","Ra","cp_EGR","gamma_EGR"];

Vd = (ifile.engine.stroke*pi()*ifile.engine.bore^2/4)/1e6; % [dm^3]==[l] , displacement volume
Vc = Vd/(ifile.engine.compression_ratio-1);                % [dm^3]==[l] , residual volume
V_tot = 4*Vd;

LAM_rod = ifile.engine.stroke/(ifile.engine.conrod_length*2);  % [-] , conrod ratio

p0 = 1.013; %[bar]
T0 = 25;    %[\textdegreeC] : from data 298 [k]

%% CORRECTED POWER AND TORQUE

% correction factor mu_ci evaluation
Psat_H20_amb = a.a0 + a.a1.*SS_data.T_snorkle + a.a2.*SS_data.T_snorkle.^2 + a.a3.*SS_data.T_snorkle.^3 +a.a4.*SS_data.T_snorkle.^4; %[kPa]
Psat_H20_0 = a.a0 + a.a1*T0+ a.a2*T0^2 + a.a3*T0^3 +a.a4*T0^4; %[kPa]

Pamb_dry = SS_data.p_baro./10 - SS_data.Rel_Hum_Air.*Psat_H20_amb/100;
p0_dry = 99; %[kPa]

% from mf_dot to qf_dot
mf_dot = SS_data.qm_fuel.*1e6/60;                 %[mg/min] --> n is in [rpm]
qf_dot = mf_dot./(SS_data.n_engine.*4*Vd/2);      %[mg/(l*cycle)]

qc = qf_dot./((SS_data.p_i_MF+SS_data.p_baro)./SS_data.p_baro);

% fm and fa_tc evaluation
fm = linspace(0,0,14)';
for ii=1:14
    if qc(ii)<37.2
        fm(ii)=0.2;
    elseif qc(ii)>65
        fm(ii) = 1.2;
    else
        fm(ii)=0.036*qc(ii)-1.14;
    end
end

fa_tc = (p0_dry./Pamb_dry).^0.7.*(SS_data.T_snorkle./T0).^1.2;
```

```
% mu_ci evaluation
mu_ci = fa_tc.^fm;


% corrected power and torque
Pwr = SS_data.T_dyno.*SS_data.n_engine.*(pi()/(1e3*30)); %[kW]
Pwr_0 = mu_ci.*Pwr;                                         %[kW]

Tr_0 = mu_ci.*SS_data.T_dyno;   %[Nm]

figure
plot(SS_data.n_engine,Tr_0, 'LineWidth',3)
hold on
plot(SS_data.n_engine,SS_data.T_dyno, 'LineWidth',3)
plot(SS_data.n_engine,Pwr_0, 'LineWidth',3)
plot(SS_data.n_engine,Pwr, 'LineWidth',3)
lgd = legend('T corr', 'T dyno', 'P corr', 'P dyno');
xlabel('engine speed [rpm]')
ylabel('Power [kW] and Torque [Nm]')
title('Mechanical Characteristic')
grid on;

figure
plot(SS_data.n_engine,mu_ci, 'LineWidth',3)
title('Correction factor')
xlabel('engine speed [rpm]')
ylabel('Correction factor [-]')
grid on;
xlim([500,4000])
ylim([0.9,1.1])

%% VOLUMETRIC EFFICIENCY, BSFC AND FUEL CONVERSION EFFICIENCY

% mixture elastic constant
R_EGR = Gas.cp_EGR*(Gas.gamma_EGR-1)/Gas.gamma_EGR;
% mixture elastic constant
R_mix = (SS_data.qm_air.*Gas.Ra+SS_data.qm_EGR.*R_EGR)./(SS_data.qm_air+
    SS_data.qm_EGR);

% [kg/h]/60 --> [kg/min]*rpm --> [kg*cycle]*0.5 (2 cycles)
m_air = SS_data.qm_air./(60.*SS_data.n_engine*0.5); %[kg]
m_EGR = SS_data.qm_EGR./(60.*SS_data.n_engine*0.5); %[kg]

% intake gases density
rho_int = ((SS_data.p_baro+SS_data.p_i_MF).*100)./(R_mix.*(SS_data.T_i_MF
    +273.15)); %[kg/m^3]

% volumetric eff
lambda = (m_air+m_EGR)./(rho_int.*V_tot/1e3); %[-]
plot(SS_data.n_engine,lambda, 'LineWidth',3)
title('Volumetric Efficiency')
xlabel('engine speed [rpm]')
ylabel('\lambda_v [-]')
grid on;
xlim([500,4000])
ylim([0,1])

% bsfc
bsfc = SS_data.qm_fuel.*1e3./Pwr_0; %[g/(kWh)]
figure
plot(SS_data.n_engine,bsfc, 'LineWidth',3)
title('Brake Specific Fuel Consumption')
xlabel('engine speed [rpm]')
ylabel('bsfc [g/kWh]')
grid on;
```

```
xlim([500,4000])


% fuel conversion eff
eta_f = Pwr_0./(1e3*Gas.Qlhv.*SS_data.qm_fuel./(3600)); %[-]
figure
plot(SS_data.n_engine,eta_f, 'LineWidth',3)
title('Fuel Conversion Efficiency')
xlabel('engine speed [rpm]')
ylabel('\eta_f [-]')
grid on;
xlim([500,4000])
ylim([0,0.5])


%% COMBUSTION DIAGNOSTIC

theta = 0:0.1:719.9;                          % [\textdegreeCA], crank angle
     degrees

P_man = double(ifile.PMAN1.data);
P_man_mean = mean(P_man');

P_exh = double(ifile.PEXH1.data);
P_exh_mean = mean(P_man');

% the shift is performed considering the mean value at intake over all the
    100 cycle perfromed

P_c1_r = double(ifile.PCYL1.data);                       % [bar], relative
    pressure  , value of 7200x100
delta_shift_1 = P_man_mean(1801)-mean(P_c1_r(1801,:)); % [bar], matching
    difference, value of 1x1
P_c1_a = P_c1_r+abs(delta_shift_1);                      % [bar], absolute
    pressure  , value of 7200x100
P_c1 = mean((P_c1_a'));                                  % [bar], mean over
    100 cycle, value of 1x7200
max(P_c1)

figure
plot(theta,P_c1_a, 'LineWidth',2)
xlim([180,540])
title('In-cylinder pressure over 100 cycles')
xlabel('crank angle [\textdegreeCA]')
ylabel('Pressure [bar]')
xlim([300,450])
xticks(300:20:450)
yticks(0:20:160)
grid on;

figure
plot(theta,P_c1, 'LineWidth',3)
xlim([180,540])
ylim([0,150])
title('In-cylinder average pressure')
xlabel('crank angle [\textdegreeCA]')
ylabel('Pressure [bar]')
xlim([300,450])
xticks(300:20:450)
yticks(0:20:160)
grid on;

%% PRESSURE FILTERING

% Butterworth filter
eng_speed = double(ifile.SPEED.data); % [rpm] , engine speed value of 100x1
```

```
eng_speed = mean(eng_speed);            % [rpm] , engine speed value of 1x1
CAstep = 720/7200;                      % [-]   , sample step

fs = eng_speed/60*360/CAstep;           % [Hz]  , sampling frequency
fc = 4000;                              % [Hz]  , cutoff frequency

Wn = fc/(fs/2);                         % [-]   , ratio between cutoff and Nyquist frequency
n = 2;                                  % order of the filter

[b,a] = butter(n,Wn);
p_cyl_butter_filtfilt=filtfilt(b,a,P_c1);

figure
plot(theta,p_cyl_butter_filtfilt, 'LineWidth',3)
hold on;
plot(theta,P_c1, 'LineWidth',3)
title('Filtered in-cylinder pressure signal')
xlabel('crank angle [\textdegreeCA]')
ylabel('Pressure [bar]')
lgd = legend('Filtered signal', 'Raw signal');
xlim([350,380])
ylim([98,140])
grid on;

P_1 = p_cyl_butter_filtfilt;

%% HRR ANALYSIS

% Brutal Volume definition
R_mix_hrr = (working_point.Air_mass_flow_rate_kg_h*Gas.Ra+working_point.qm_egr_kg_h*R_EGR)/(working_point.Air_mass_flow_rate_kg_h+working_point.qm_egr_kg_h); %[J/kgK]

m_fuel_kg = working_point.Fuel_mass_flow_rate_kg_h./(60.*eng_speed*0.5*4);          %[kg]
m_air_hrr = working_point.Air_mass_flow_rate_kg_h./(60.*eng_speed*0.5*4);           %[kg]
m_EGR_hrr = working_point.qm_egr_kg_h/(60*eng_speed*0.5*4);                         %[kg]
m_TOT_hrr = m_air_hrr+m_EGR_hrr;
LHV = 42.5e6;

T = P_1.*1e5.*V./(R_mix_hrr*m_TOT_hrr); %[K]

gamma = 1.338-6*10^-5.*T+10^-8.*(T).^2;

% HRR
dQn = zeros([1,7200]);       % [J]
SOI = 3452;                 % start of injection
for i = SOI:4800
    dV = V(i)-V(i-1);
    dp = (P_1(i)-P_1(i-1))*1e5;
    dQn(i-1) = (gamma(i-1)*P_1(i-1)*1e5*dV + V(i-1)*dp)/(gamma(i-1)-1);
end

% Injection
inj=ifile.INJ1.data;
inj_avg= mean(inj');

% mass fraction burn
% from 3550 to avoid pilot evaluation phase
xb = zeros([1,7200]);
xb(3550:4500) = cumtrapz(theta(3550:4500),dQn(3550:4500));  % [%]
xb = xb./max(xb(:));                                        % [%]

```

```
% heat release
Qn = xb * m_fuel_kg * LHV;
Qf = m_fuel_kg * LHV;

figure
plot(theta,inj_avg, 'LineWidth',3)
title('Injection current signal')
ylabel('Current [A]')
xlabel('crank angle [\textdegreeCA]')
xlim([330,400])
ylim([-2,16])
grid on;

figure
plot(theta,dQn/CAstep, 'LineWidth',3)
title('HRR_n')
xlabel('crank angle [\textdegreeCA]')
ylabel('Net heat relase rate [J/\textdegreeCA]')
xlim([330,440])
ylim([-50,150])
grid on

figure
plot(theta,xb, 'LineWidth',3)
title('Mass Fraction Burned')
xlim([330,440])
ylim([-0.2,1.2])
xlabel('crank angle [\textdegreeCA]')
ylabel('x_b')
grid on

figure
hold on
yyaxis left
plot(theta,dQn/CAstep, 'LineWidth',3)
xlabel('crank angle [\textdegreeCA]')
ylabel('Net heat relase rate [J/\textdegreeCA]')
ylim([-50,150])
yyaxis right
plot(theta,inj_avg, 'LineWidth',3)
ylabel('Current [A]')
ylim([-2,60])
xlabel('crank angle [\textdegreeCA]')
xlim([330,440])
hold off
grid on

figure
plot(theta,Qn/1.13, 'LineWidth',3)
title('Heat release analysis')
xlim([330,440])
ylim([-200,3800])
xlabel('crank angle [\textdegreeCA]')
ylabel('Released heat [J]')
hold on;
plot([330,440], [Qf,Qf], 'LineWidth',3)
lgd = legend('Q_n', 'm_f Q_L_H_V');
grid on;

%% PRESSURE COMPARISON

P_motored=P_1;
for i=(SOI+1):5500
    P_motored(i)=P_motored(i-1)*(V(i-1)/V(i))^gamma(i);
end

```

```
figure
plot(theta, P_c1, 'LineWidth',3);
hold on;
plot(theta,P_1, 'LineWidth',3);
hold on;
plot(theta,P_motored, 'LineWidth',3);
title('In-cylinder pressure comparison')
legend('raw','filtered','motored')
xlabel('crank angle [\textdegreeCA]')
ylabel('Pressure [bar]')
xlim([300,450])
ylim([0,150])
xticks(300:20:450)
yticks(0:20:160)
grid on;
```