EGB322: Engine Performance Practical

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Adam Telfer
N12068373

Contents

List of Symbols

Symbol	Unit	Description
A_O	m^2	Orifice area
C_O	$-$	Orifice correction factor
C_{comp}	$N \cdot m$	Compression constant
C_{exp}	$N \cdot m$	Expansion constant
d_O	m	Orifice diameter
Δh	m	Height difference of manometer fluid
m	kg	Mass
$\dot{m}$	kg/s	Mass Flow Rate
N	rev/min	Engine Revolutions Per Minute
N_s	rev/s	Engine Revolutions Per Second
n_{comp}	$-$	Polytropic coefficient of compression
n_{exp}	$-$	Polytropic coefficient of expansion
P	kPa	Pressure
Q	kJ	Heat energy
$\dot{Q}$	kW	Rate of heat energy
r	$-$	Compression ratio
r_c	$-$	Cutoff Ratio
T	$°C$ or K	Temperature
V	m^3	Volume
$\dot{V}$	m^3/s	Volumetric Flow Rate
W	kJ	Work
$\dot{W}$	kW	Power
AF	$-$	Air-Fuel Ratio
$BMEP$	kPa	Brake Mean Effective Pressure
$IMEP$	kPa	Indicated Mean Effective Pressure
SFC	$g/kW \cdot hr$	Specific Fuel Consumption
η	$-$	Efficiency
ρ	kg/m^3	Density

1. Introduction

1.1. Background and Context

Internal combustion engines remain a critical technology for transportation and power generation due to their high thermal efficiency and robust performance. Thus, understanding and quantifying their performance under various operating conditions is fundamental in engineering. The efficiency and power output of a combustion engine are directly influenced by operational parameters, such as Revolutions Per Minute (RPM) and load, and by its design, such as the four-stroke cycle and compression ignition. This report details an experimental investigation into a four-cylinder, four-stroke diesel engine to characterise its performance and energy distribution.

1.2. Theory

The engine being investigated operates on the four-stroke cycle, wherein the piston completes four distinct strokes (Intake, Compression, Power, and Exhaust) for every two revolutions of the crankshaft [1, p. 457]. This sequence facilitates the use of compression ignition, where air is compressed above the fuel's auto-ignition temperature. This process allows for higher compression ratios than spark-ignition engines, resulting in greater thermal efficiency and mitigating engine knock [1, pp. 464-465].

To simplify the real combustion process for analysis, the air-standard cycle is used as an idealised model. This model assumes that air is the working fluid, all processes are internally reversible, and the exhaust/intake processes are replaced by a closed-loop heat rejection process. The cycle assumes that the air undergoes isentropic compression, constant-pressure heat addition, isentropic expansion, and constant-volume heat rejection [1, p. 454].

Comparing experimental Pressure-Volume (P-V) data to the theoretical cycle and performing an Energy Balance helps to quantify losses due to irreversibilities and non-ideal heat transfers, which are typically ignored by the air-standard assumptions. Metrics, such as brake power, thermal efficiency, and specific fuel consumption are used to quantify the engine performance.

1.3. Aim

The aim of this experiment is to investigate and analyse the performance of a four-cylinder diesel engine under varying RPM and load conditions, using performance metrics to understand relevant thermodynamic concepts. This analysis will be supplemented by an energy balance conducted at full load at differing RPMs, and by comparing the experimental pressure-volume (P-V) data from the combustion chamber to an idealised theoretical diesel cycle.

1.4. Outline

Section 2 details the experimental setup and methodology. Section 3 presents the raw and calculated results, along with relevant figures and calculations. Section 4 contains a detailed discussion on the results, comparing experimental findings to theoretical principles. Section 5 is the conclusion of the report, summarising the findings made.

2. Test Engine and Instrumentation

2.1. Equipment

- **Perkins 404D-22**, 2.2 L naturally aspirated inline 4-cylinder, 4-stroke diesel engine.
- **Dynamometer**, connected to the engines output shaft, measuring the engines output torque (in Nm) and its angular velocity/speed (in RPM). The output power (aka brake power) is then calculated. The measured speed had a resolution to a single RPM and torque to a single Nm.
- **Cooling Water Flow Meter** measures the volume flow rate of the cooling water in litres per minute (LPM). The resolution was to 0.1 LPM.
- **U-Tube Manometer** measures the height difference of the manometer fluid in mm. Used for calculating the pressure difference between the ambient air and the inside of the air tank (seen in Figure 3). The pressure difference, along with the diameter of the orifice (0.03 m) is used to calculate the volumetric flow rate of air into the engine. Manometer resolution is to the whole mm. Manometer had an initial error of 14 mm on the RHS when engine was not running.
- **3 x Thermocouples**, for recording the input and output temperature of the cooling water, calorimeter and exhaust gas in degrees Celsius ($°C$).
- **Exhaust Gas Calorimeter**, to measure the heat content of the engines exhaust gas. During the experiment it was not working and so considered an extension of the exhaust system.
- **DynoLog Software**, for recording and displaying engine metrics, with the throttle position, RPM, torque and power being recorded from it. The power was displayed with a resolution of 0.1 kW, and throttle position as a percentage from zero to wide open.

Figure 1: Engine apparatus, front side.

Figure 2: Engine apparatus, rear side.

Figure 3: U-Tube manometer for measuring difference in the ambient air pressure and inside the air tank.

Figure 4: Flow meter for measuring the volumetric flow rate of the cooling water in LPM.

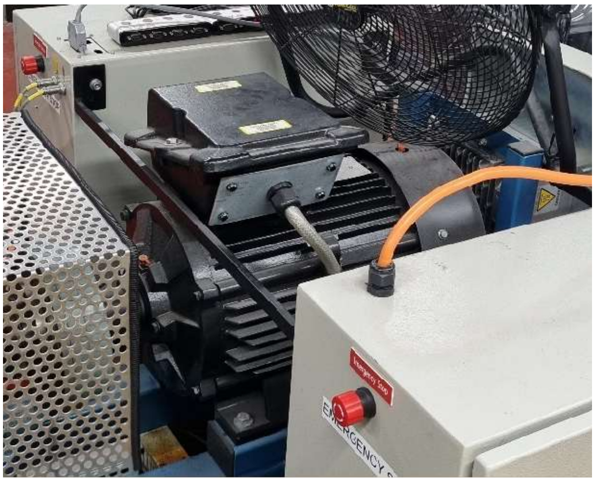

Figure 5: Dynamometer. Output shaft from engine comes from left of image.

2.2. Method

1. The engine was turned on and allowed to run under 0% load for 10 minutes to allow it to warm up.
2. The engine control was set to a nominal speed of 1400 RPM.
3. The engines throttle was set to 100% and let to run for 2 minutes to allow it to reach thermal equilibrium. The metrics in the 'Engine Lab Table' were recorded for 100% load.
4. The throttle was then adjusted to achieve approximately 50% of the maximum torque recorded at that RPM and allowed to run for 2 minutes to re-establish thermal equilibrium. The metrics were then recorded for the 50% load condition.
5. Steps 2-4 were repeated for 1800 and 2000 RPM.
6. The engine was allowed to run under no load for several minutes before being shut off.

2.3. Control Volume

The engine is modelled as an open control volume (CV) operating under steady-flow conditions, as mass and energy cross the boundary. Figure 6 is a diagram showing that the input energy comes from the energy in the diesel fuel ($\dot{Q}_{fuel}$), and the energy is output in the form of brake power ($\dot{W}_B$), cooling heat ($\dot{Q}_{cooling}$), exhaust heat ($\dot{Q}_{exh}$), and unaccounted for losses ($\dot{Q}_{other}$) such as radiation and mechanical losses. Mass enters the CV as air ($\dot{m}_{air}$) and fuel ($\dot{m}_{fuel}$) and exits as exhaust gas ($\dot{m}_{exh}$).

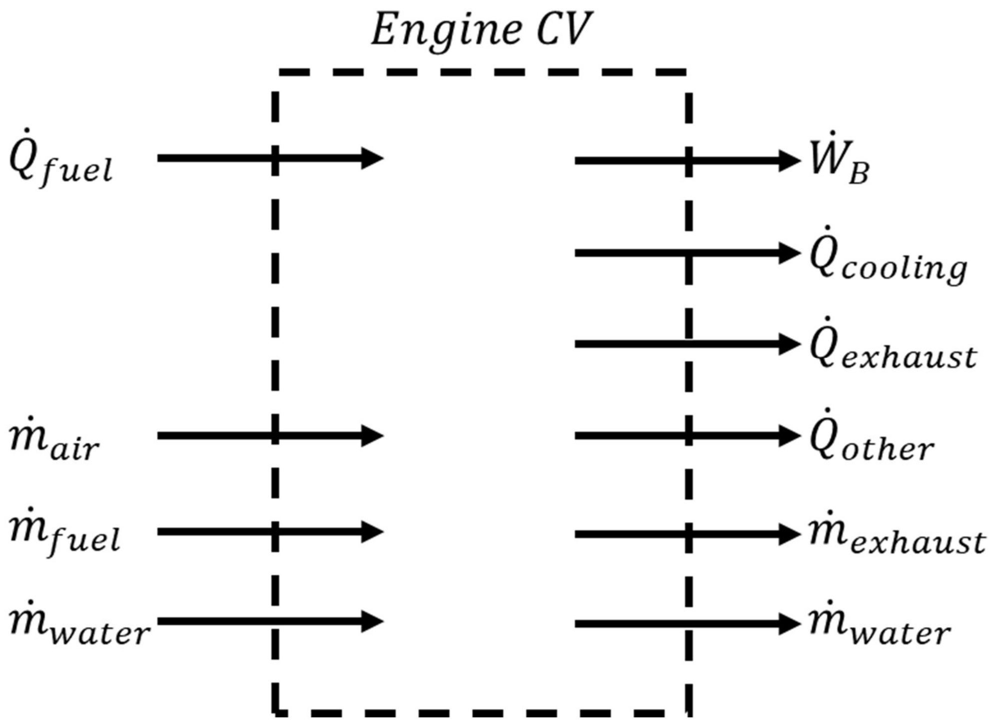

Figure 6: Control volume diagram, showing the energy and mass inputs and outputs.

3. Results

3.1. Engine Lab Table

Table 1: Data collected during experiment.

Nominal Speed (RPM)	Engine Load (%)	Measured Speed (RPM)	Throttle (%)	$\dot{W}_B$ (kW)	Torque $(N \cdot m)$	$T_{water,in}$ $(°C)$	$T_{water,out}$ $(°C)$
1400	50	1395	68	9.5	64	26.65	36.48
1400	100	1395	100	18.4	126	26.47	35.69
1800	50	1787	85	10.7	56	27.4	39.12
1800	100	1787	100	22.3	119	27.36	40.44
2000	50	1991	95	11.6	56	28.32	40.29
2000	100	1991	100	22.7	109	28.12	41.65
Nominal Speed (RPM)	**Engine Load (%)**	**$T_{exh,in}$ $(°C)$**	**$T_{exh,out}$ $(°C)$**	**$T_{cal,in}$ $(°C)$**	**$T_{cal,out}$ $(°C)$**	**Cool Rate (LPM)**	**Fuel Rate (LPM)**	**Δh (mm)**
1400	50	268.71	250.82	205	150	32.8	0.054	0.120
1400	100	494.8	428.38	308	227	32.8	0.098	0.114
1800	50	282.39	247.46	225	173	33.3	0.059	0.204
1800	100	533.66	478.96	372	284	33.3	0.118	0.192
2000	50	300.69	284.29	236	186	33.9	0.062	0.250
2000	100	517.57	472.4	364	289	33.9	0.124	0.234
Room temperature during experiment: 24.7 $°C$								

3.2. Engine Performance

Table 2: Results from engine performance analysis.

Nominal Speed (RPM)	Engine Load (%)	Measured Speed (RPM)	$\dot{W}_B$ (kW)	Torque $(N \cdot m)$	BMEP (kPa)	SFC $\left(\frac{g}{kWhr}\right)$	η_V (%)	η_T (%)
1400	50	1395	9.5	64	363	293	73.35	27.90
1400	100	1395	18.4	126	714	275	71.49	29.77
1800	50	1787	10.7	56	317	285	74.66	28.76
1800	100	1787	22.3	119	675	273	72.43	29.97
2000	50	1991	11.6	56	317	276	74.18	29.67
2000	100	1991	22.7	109	618	282	71.76	29.03
Nominal Speed (RPM)	**Engine Load (%)**	**AF**	**$\dot{V}_{fuel}$ (m^3)**	**$\dot{m}_{fuel}$ (kg/s)**	**$\dot{V}_{air}$ (m^3)**	**$\dot{m}_{air}$ (kg/s)**		
1400	50	28.91	0.0000009	0.000774	0.018901256	0.022379087		
1400	100	15.53	1.63333E-06	0.001404667	0.018422665	0.021812436		
1800	50	34.50	9.83333E-07	0.000845667	0.024644222	0.029178759		
1800	100	16.74	1.96667E-06	0.001691333	0.023908408	0.028307555		
2000	50	36.35	1.03333E-06	0.000888667	0.027281613	0.03230143		
2000	100	17.58	2.06667E-06	0.001777333	0.026394167	0.031250694		

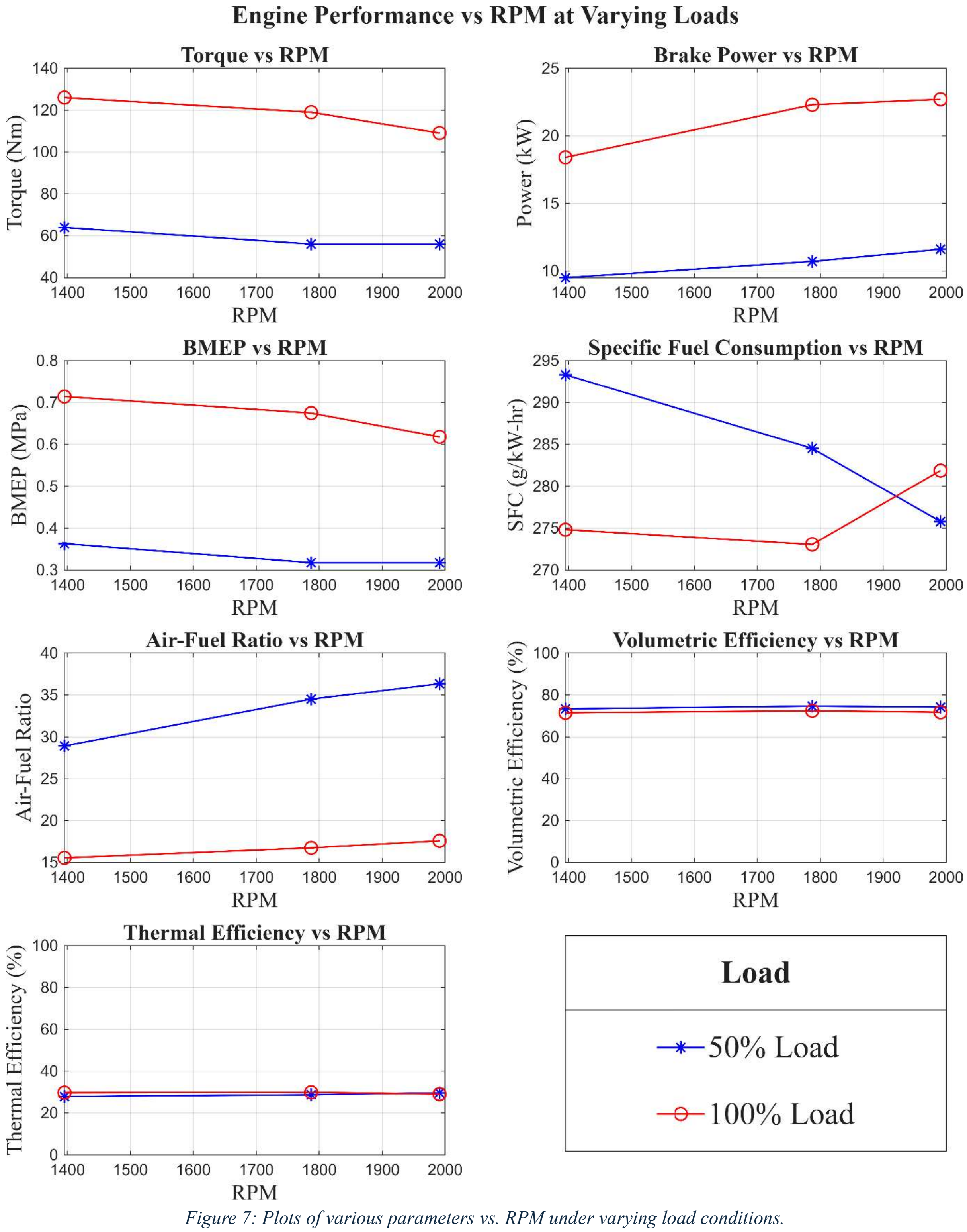

Figure 7: Plots of various parameters vs. RPM under varying load conditions.

Engine Performance at 1400 RPM and 50% Load

$N = 1395\ RPM$
$\dot{W}_B = 9.5\ kW$
$T = 64\ N \cdot m$
$Fuel\ Flow = 0.054\ LPM$
$\rho_{fuel} = 860\ \ kg/m^3$ [2]
$\rho_{air} = 1.184\ \ kg/m^3$ [3, pp. 6-18]
$\rho_{man} = 1000\ \ kg/m^3$ [4]
$Q_{HV,diesel} = 44000\ \ kJ/kg$ [2]

$\Delta h = 120\ mm$
$B = 84\ mm$ [2]
$L = 100\ mm$ [2]
$n_{cyl} = 4$ [2]
$n_r = 2$ [1, p. 462]
$C_0 = 0.6$ [4]
d_0 = 0.03 m [4]

Engine Displacement

$$V_D = \frac{\pi}{4}B^2Ln_{cyl} = \frac{\pi}{4} \cdot 0.084 \cdot 0.1 \cdot 4 = 0.002217\ m^3$$

Brake Mean Effective Pressure

$$BMEP = \frac{2\pi n_r T}{V_D} = \frac{2\pi \cdot 2 \cdot 64}{0.002217} = 363\ kPa$$

Volumetric flow rate of fuel and Specific Fuel Consumption

$$\dot{V}_{fuel} = \frac{0.054}{60000} = 9.0 \cdot 10^{-7}\frac{m^3}{s}$$

$$Fuel\ FLow \cdot \frac{L}{min} \cdot \frac{m^3}{1000\ L} \cdot \frac{min}{60\ s} = \frac{m^3}{s}$$

$$\dot{m}_{fuel} = \dot{V}_{fuel} \cdot \rho_{fuel} = 9.0 \cdot 10^{-7} \cdot 860 = 7.74 \cdot 10^{-4}\frac{kg}{s}$$

$$SFC = \frac{\dot{m}_{fuel}}{\dot{W}_B} \cdot 3600000$$

$$\frac{m^3}{s} \cdot \frac{kg}{m^3} \cdot \frac{3600\ s}{hr} \cdot \frac{1000\ g}{kg}$$

$$SFC = \frac{7.74 \cdot 10^{-4}}{9.5} \cdot 3600000 = 293.3\ \ \frac{g}{kW \cdot hr}$$

Volumetric and Mass Flow Rate of Air

$$A_o = \frac{\pi}{4} \cdot d_o^2 = 7.06 \cdot 10^{-4}\ m^2$$

$$\Delta P = (\rho_{man} - \rho_{air}) \cdot g \cdot \Delta h = (1000 - 1.184) \cdot 9.81 \cdot 0.120 = 1176\ Pa$$

$$\dot{V}_{air} = C_0 \cdot A_0 \cdot \sqrt{2\frac{\Delta P}{\rho_{air}}} = 0.6 \cdot 7.06 \cdot 10^{-4} \cdot \sqrt{2\frac{1176}{1.184}} = 1.89 \cdot 10^{-2}\frac{m^3}{s}$$

$$\dot{m}_{air} = \dot{V}_{air} \cdot \rho_{air} = 1.89 \cdot 10^{-2} \cdot 1.184 = 2.24 \cdot 10^{-2}\frac{kg}{s}$$

Air-Fuel Ratio

$$AF = \frac{\dot{m}_{air}}{\dot{m}_{fuel}} = \frac{2.24 \cdot 10^{-2}}{7.74 \cdot 10^{-4}} = 28.9$$

Volumetric Efficiency

$$\eta_V = \frac{n_r \cdot \dot{m}_{air}}{\rho_{air} \cdot V_D \cdot N_s} = \frac{2 \cdot 2.24 \cdot 10^{-2}}{1.184 \cdot 0.002217 \cdot \frac{1395}{60}} \cdot 100 = 73.4\%$$

Thermal Efficiency

$$\eta_T = \frac{\dot{W}_B}{\dot{m}_{fuel} \cdot Q_{HV,diesel}} = \frac{9.5}{7.74 \cdot 10^{-4} \cdot 44000} \cdot 100 = 27.9\%$$

3.3. Heat Balance

Table 3: Heat balance results at nominal RPM and 100% load.

Nominal Speed (RPM)	$\dot{Q}_{fuel}$ (kW)	$\dot{Q}_{cooling}$ (kW)	$\dot{Q}_{exhaust}$ (kW)	$\dot{W}_B$ (kW)	$\dot{Q}_{other}$ (kW)	$\dot{m}_{exhaust}$ (kg/s)	$\dot{m}_{water}$ (kg/s)
1400	61.81	21.17	6.35	18.4	15.88	0.023217102	0.546666667
2000	78.20	32.11	7.76	22.7	15.64	0.033028027	0.565

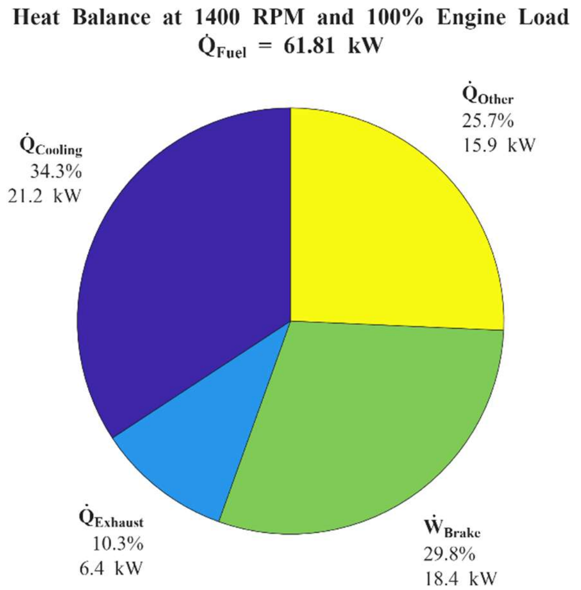

Figure 8: Pie chart of heat balance at 1400 RPM and 100% load.

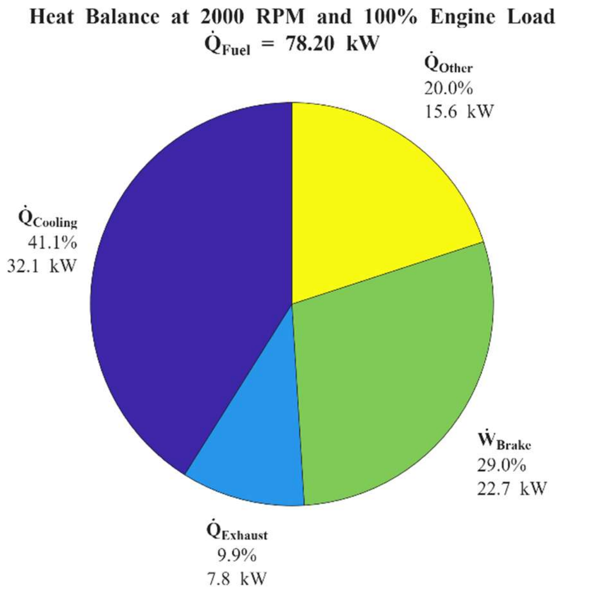

Figure 9: Pie chart of heat balance for 2000 RPM and 100% load.

Heat Balance at 1400 RPM and 100% Load

$\dot{m}_{fuel} = 1.404 \cdot 10^{-3}\ kg/s$

$\dot{m}_{air} = 2.181 \cdot 10^{-2}\ kg/s$

$Water\ Flow = 32.8\ LPM$

$\dot{W}_B = 18.4\ kW$

$T_{water,in} = 26.47\ °C$

$T_{water,out} = 35.69\ °C$

$T_{exhaust,in} = 494.78\ °C$

$T_{exhaust,out} = 428.38\ °C$

$T_{cal,out} = 227\ °C$

$Q_{HV,diesel} = 44000\ kJ/kg$ [2]

$c_{p,water} = 4.2\ kJ/kg \cdot K$

Heat Input from Fuel

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot Q_{HV,diesel} = 1.404 \cdot 10^{-3} \cdot 44000 = 61.78\ kW$$

Heat Output to Cooling System

$$\dot{V}_{water} = \frac{Flow\ Water}{60000} = \frac{32.8}{60000} = 5.47 \cdot 10^{-4} \frac{m^3}{s}$$

$$\dot{m}_{water} = \dot{V}_{water} \cdot \rho_{water} = 5.47 \cdot 10^{-4} \cdot 1000 = 0.547 \frac{kg}{s}$$

$$\dot{Q}_{cooling} = \dot{m}_{water} \cdot c_{p,water} \cdot \left(T_{water,out} - T_{water,in}\right)$$

$$\dot{Q}_{cooling} = 0.547 \cdot 4.2 \cdot (35.69 - 26.47) = 21.18\ kW$$

Heat Output to Exhaust

$$\dot{m}_{exhaust} = \dot{m}_{fuel} + \dot{m}_{air} = 1.404 \cdot 10^{-3} + 2.181 \cdot 10^{-2} = 2.321 \cdot 10^{-2}\ kg/s$$

$$c_{p,exhaust} \approx c_{p,air@T_{avg,exh}} = 1.022\ kJ/kg \cdot K$$

$c_{p,air}$ taken from [1, pp. 842, Table A-2(b)]

$$T_{avg,exh} = \frac{494.78 + 428.38}{2} = 462\ °C$$

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,exhaust} \cdot \left(T_{exhaust,in} - T_{cal,out}\right)$$

$$\dot{Q}_{exhaust} = 2.321 \cdot 10^{-2} \cdot 1.022 \cdot (494.78 - 227) = 6.35\ kW$$

Unaccounted for Heat Losses

$$\dot{Q}_{fuel} = \dot{W}_B + \dot{Q}_{fuel} + \dot{Q}_{exhaust} + \dot{Q}_{other}$$

$$\dot{Q}_{other} = \dot{Q}_{fuel} - \dot{W}_B - \dot{Q}_{fuel} - \dot{Q}_{exhaust} = 15.85\ kW$$

Heat Balance at 2000 RPM and 100% Load

$\dot{m}_{fuel} = 1.177 \cdot 10^{-3} \ kg/s$

$\dot{m}_{air} = 3.125 \cdot 10^{-2} \ kg/s$

$Water\ Flow = 33.9\ LPM$

$\dot{W}_B = 22.7\ kW$

$T_{water,in} = 28.12\ °C$

$T_{water,out} = 41.65\ °C$

$T_{exhaust,in} = 517.57\ °C$

$T_{exhaust,out} = 472.40\ °C$

$T_{cal,out} = 289\ °C$

$Q_{HV,diesel} = 44000 \ kJ/kg$ [2]

$c_{p,water} = 4.2 \ kJ/kg \cdot K$

Heat Input from Fuel

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot Q_{HV,diesel} = 1.177 \cdot 10^{-3} \cdot 44000 = 78.19\ kW$$

Heat Output to Cooling System

$$\dot{V}_{water} = \frac{Flow\ Water}{60000} = \frac{33.9}{60000} = 5.65 \cdot 10^{-4} \frac{m^3}{s}$$

$$\dot{m}_{water} = \dot{V}_{water} \cdot \rho_{water} = 5.65 \cdot 10^{-4} \cdot 1000 = 0.565 \ \frac{kg}{s}$$

$$\dot{Q}_{cooling} = \dot{m}_{water} \cdot c_{p,water} \cdot \left(T_{water,out} - T_{water,in}\right)$$

$$\dot{Q}_{cooling} = 0.565 \cdot 4.2 \cdot (41.65 - 28.12) = 32.12\ kW$$

Heat Output to Exhaust

$$\dot{m}_{exhaust} = \dot{m}_{fuel} + \dot{m}_{air} = 1.177 \cdot 10^{-3} + 3.125 \cdot 10^{-2} = 3.303 \cdot 10^{-2} \ kg/s$$

$$c_{p,exhaust} \approx c_{p,air@T_{avg,exh}} = 1.028 \ kJ/kg \cdot K$$

$c_{p,air}$ taken from [1, pp. 842, Table A-2(b)]

$$T_{avg,exh} = \frac{517.57 + 472.40}{2} = 495\ °C$$

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,exhaust} \cdot \left(T_{exhaust,in} - T_{cal,out}\right)$$

$$\dot{Q}_{exhaust} = 3.303 \cdot 10^{-2} \cdot 1.028 \cdot (517.57 - 289) = 7.76\ kW$$

Unaccounted for Heat Losses

$$\dot{Q}_{fuel} = \dot{W}_B + \dot{Q}_{fuel} + \dot{Q}_{exhaust} + \dot{Q}_{other}$$

$$\dot{Q}_{other} = \dot{Q}_{fuel} - \dot{W}_B - \dot{Q}_{fuel} - \dot{Q}_{exhaust} = 15.61\ kW$$

3.4. P-V Analysis and States

Table 4: Results for P-V analysis at nominal RPM and 100% load.

Nominal Speed (RPM)	Measured Speed (RPM)	W_i (kJ)	$\dot{W}_i$ (kW)	η_{mech} (%)	IMEP (kPa)	Heat Input (%)	Q_{in} (kJ)	Q_{fuel} (kJ)
1400	1395	0.424	19.72	93.3	765	42.9	0.570	1.33
2000	1991	0.403	26.74	84.9	727	26.6	0.313	1.18

Nominal Speed (RPM)	Cutoff Ratio	m_{fuel} (kg)	m_{air} (kg)	n_{comp}	C_{comp}	n_{exp}	C_{exp}
1400	2.33	3.02079E-05	0.000469085	1.292	0.00557	1.213	0.03537
2000	1.74	2.67805E-05	0.000470879	1.383	0.00396	1.268	0.02631

Nominal Speed (RPM)	P_1 (kPa)	V_1 (m^3)	T_1 (K)	P_2 (kPa)	V_2 $(m$^3)	T_2 (K)
1400	111	0.00047	297.9	4072	2.9E-05	908.5
2000	145.6	0.0005	297.9	6619	3.2E-05	898.2

Nominal Speed (RPM)	P_3 (kPa)	V_3 (m^3)	T_3 (K)	P_4 (kPa)	V_4 (m^3)	T_4 (K)
1400	4072	6.7E-05	2118.2	386	0.00047	694.4
2000	6619	5.5E-05	1560.0	403	0.0005	517.3

P-V Analysis at 1400 RPM and 100% Load

$N = 1395\ RPM$
$W_i = 0.424\ kJ$
$V_1 = V_{max} = 4.70 \cdot 10^{-4}\ m^3$
$V_2 = V_{min} = 2.89 \cdot 10^{-5}\ m^3$

$\dot{W}_B = 18.4\ kW$
$\dot{m}_{fuel} = 1.404 \cdot 10^{-3}\ kg/s$

$k_{air} = 1.4$ [1, pp. 847, Table A-2]
$Q_{HV,diesel} = 44000\ kJ/kg \cdot K$ [2]
$V_D = 2.217 \cdot 10^{-3}\ m^3$
$c_{p,air} = 1.005\ kJ/kg \cdot K$
[1, pp. 847, Table A-2]
$T_1 = 24.7°C$
$\dot{m}_{air} = 2.181 \cdot 10^{-2}\ kg/s$

Indicated Power

$$\dot{W}_i = \frac{W_i \cdot N_s \cdot n_{cyl}}{n_r} = \frac{0.424 \cdot \frac{1395}{60} \cdot 4}{2} = 19.7\ kW$$

Mechanical Efficiency

$$\eta_{mech} = \frac{\dot{W}_B}{\dot{W}_i} \cdot 100 = \frac{18.4}{19.7} \cdot 100 = 93.4\%$$

Indicated Mean Effective Pressure

$$IMEP = \frac{W_i \cdot n_{cyl}}{V_D} = \frac{0.424 \cdot 4}{2.217 \cdot 10^{-3}} = 765\ kPa$$

Polytropic Coefficients of Compression and Expansion

The polytropic coefficients of compression and expansion can be found by plotting the P-V data in log-log scale, picking two points on the linear section for expansion and compression, and finding the slope of the line between the points. This process can be seen in {figure reference}. The two right hand points were chosen to be the maximum volume that was in the linear region for expansion and compression, so $V_{max} = V_1 = V_4$. The left-hand points were chosen to be somewhere in linear section. The constant C allows calculation of the pressure given the volume, and vice versa.

$$n = \frac{\log(P_2) - \log(P_1)}{\log(V_2) - \log(V_1)} \qquad PV^n = C \qquad \frac{N}{m^2} \cdot m^3 = N \cdot m$$

Compression Stroke

$P_1 = 111\ kPa$ $\qquad V_1 = 4.7 \cdot 10^{-4}\ m^3$

$P_2 = 2678\ kPa$ $\qquad V_2 = 4.0 \cdot 10^{-5}\ m^3$

$$n_{\text{comp}} = \frac{\log(2678) - \log(111)}{\log(4.0 \cdot 10^{-5}) - \log(4.7 \cdot 10^{-4})} = 1.292$$

$$C_{\text{comp}} = 111 \cdot (4.7 \cdot 10^{-4})^{1.292} = 0.0055677$$

Expansion Stroke

$P_1 = 2524\ kPa$ $\qquad V_1 = 1.0 \cdot 10^{-4}\ m^3$

$P_2 = 386\ kPa$ $\qquad V_2 = 4.7 \cdot 10^{-4}\ m^3$

$$n_{\text{comp}} = \frac{\log(386) - \log(2524)}{\log(4.7 \cdot 10^{-4}) - \log(1.0 \cdot 10^{-4})} = 1.213$$

$$C_{\text{comp}} = 2524 \cdot (1.0 \cdot 10^{-4})^{1.213} = 0.035369$$

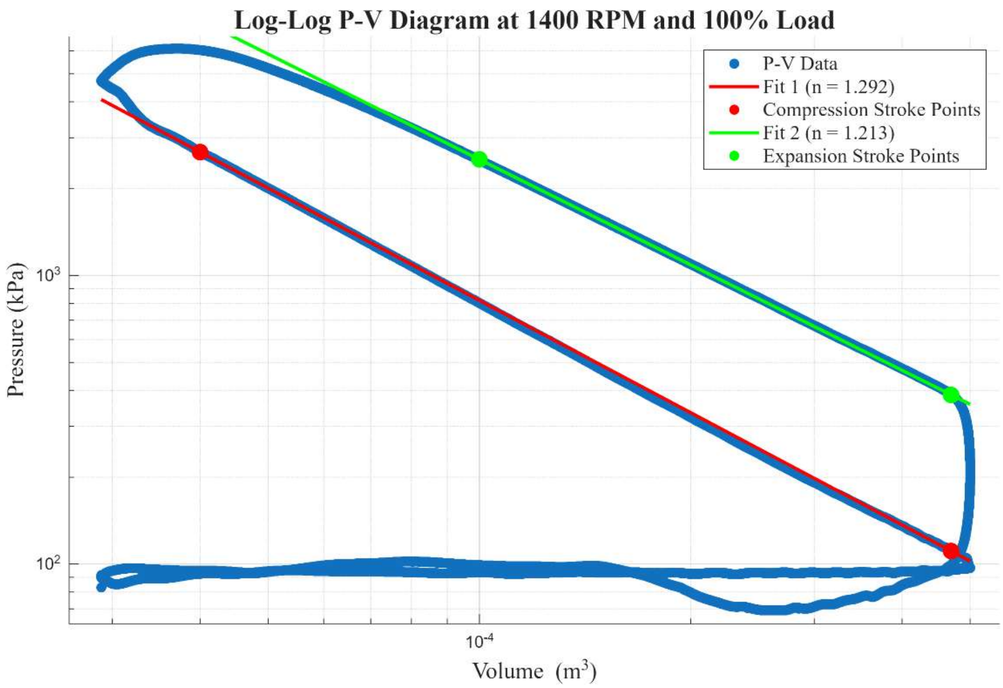

Figure 10: Plot used for finding the polytropic coefficients of compression and expansion at 1400 RPM.

States of Ideal Diesel Cycle

$V_1 = 4.70 \times 10^{-4}\ m^3$ $V_2 = 2.89 \times 10^{-5}\ m^3$

$n_{\text{comp}} = 1.292$ $C_{\text{comp}} = 0.0055678$

$n_{\text{exp}} = 1.213$ $C_{\text{exp}} = 0.035369$

Calculate the pressure and volume at the four different states in the ideal diesel cycle.

$$PV^n = C \qquad P = \frac{C}{V^n} \qquad V = \left(\frac{C}{P}\right)^{\frac{1}{n}}$$

$$P_1 = \frac{C_{\text{comp}}}{V_1^{n_{\text{comp}}}} = \frac{0.0055678}{(4.7 \cdot 10^{-4})^{1.292}} = 111.0\ kPa$$

$$P_2 = \frac{C_{\text{comp}}}{V_2^{n_{\text{comp}}}} = \frac{0.0055678}{(2.89 \cdot 10^{-5})^{1.292}} = 4075\ kPa$$

$$P_3 = P_2 = 4075\ kPa$$

$$V_3 = \left(\frac{C_{\text{exp}}}{P_3}\right)^{\frac{1}{n_{\text{exp}}}} = \left(\frac{0.035369}{4075}\right)^{\frac{1}{1.213}} = 6.718 \cdot 10^{-5}\ m^3$$

$$V_4 = V_1 = 4.70 \cdot 10^{-4}\ m^3$$

$$P_4 = \frac{C_{\text{exp}}}{V_4^{n_{\text{exp}}}} = \frac{0.035369}{(4.7 \cdot 10^{-4})^{1.213}} = 384.9\ kPa$$

Calculate the temperature at the different states.

$T_1 = 24.7°C = 297.85\ K$ $k_{air} = 1.4$

Compression Ratio: $r = \dfrac{V_1}{V_2} = \dfrac{4.7 \cdot 10^{-4}}{2.89 \cdot 10^{-5}} = 16.26$

$$T_2 = T_1 \cdot r^{k-1} = 297.85 \cdot 16.26^{0.4} = 908\ K$$

$$T_3 = T_2 \cdot \frac{V_3}{V_2} = 908 \cdot \frac{6.718 \cdot 10^{-5}}{2.89 \cdot 10^{-5}} = 2110\ K$$

$$T_4 = T_3 \cdot \left(\frac{1}{r}\right)^{k-1} = 2110 \cdot \left(\frac{1}{16.26}\right)^{0.4} = 692\ K$$

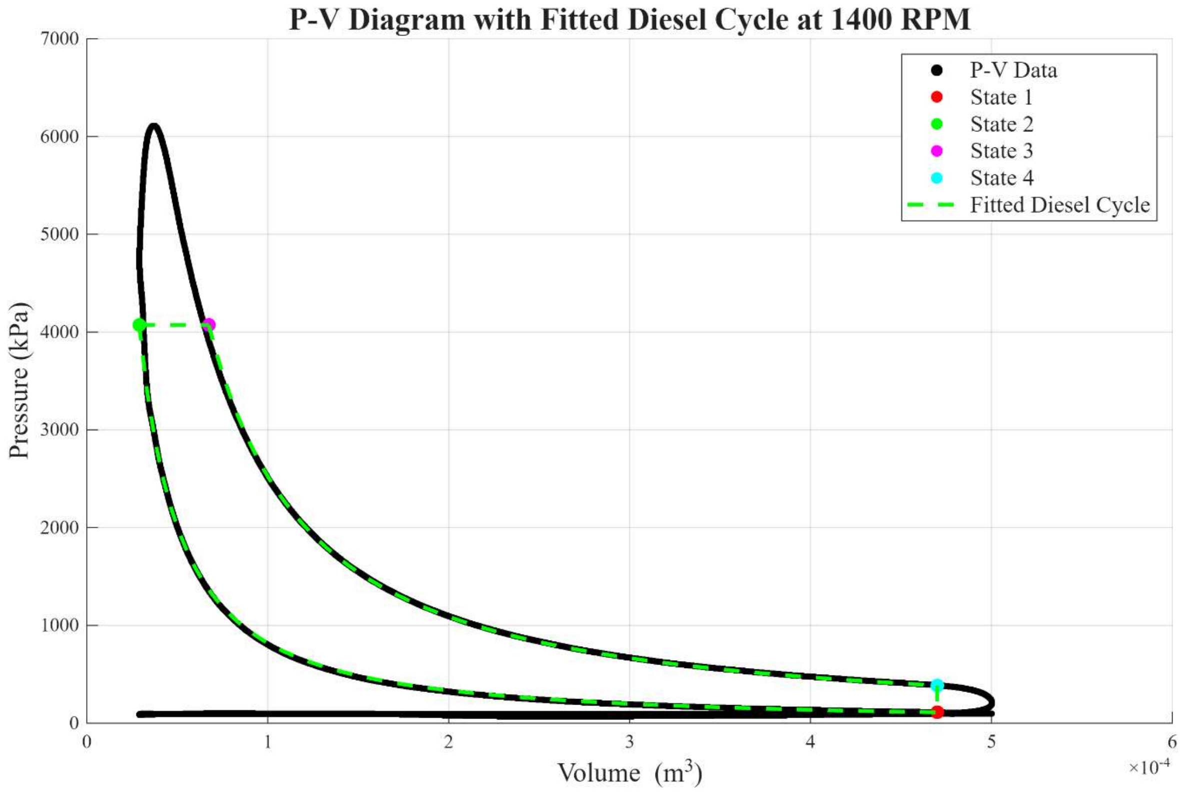

Figure 11: Plot of recorded P-V data and fitted ideal diesel cycle at 1400 RPM.

Cutoff Ratio

$$r_c = \frac{V_3}{V_2} = \frac{6.718 \cdot 10^{-5}}{2.89 \cdot 10^{-5}} = 2.33$$

Heat Input

$$m_{air} = \frac{\dot{m}_{air} \cdot n_r}{N_s \cdot n_{cyl}} = \frac{2.181 \cdot 10^{-2} \cdot 2}{\frac{1395}{60} \cdot 4} = 4.69 \cdot 10^{-4}\ kg$$

$$m_{fuel} = \frac{\dot{m}_{fuel} \cdot n_r}{N_s \cdot n_{cyl}} = \frac{1.404 \cdot 10^{-3} \cdot 2}{\frac{1395}{60} \cdot 4} = 3.02 \cdot 10^{-5}\ kg$$

$$Q_{fuel} = m_{fuel} \cdot Q_{HV,diesel} = 3.02 \cdot 10^{-5} \cdot 44000 = 1.329\ kJ$$

$$Q_{in} = m_{air} \cdot C_{p,air} \cdot (T_3 - T_2) = 4.69 \cdot 10^{-4} \cdot 1.005 \cdot (2110 - 908) = 0.567\ kJ$$

$$\frac{Q_{in}}{Q_{fuel}} \cdot 100 = \frac{0.567}{1.329} \cdot 100 = 42.6\%$$

P-V Analysis at 2000 RPM and 100% Load

$N = 1991\ RPM$
$W_i = 0.403\ kJ$
$V_1 = V_{max} = 5.00 \cdot 10^{-4}\ m^3$
$V_2 = V_{min} = 3.166 \cdot 10^{-5}\ m^3$

$\dot{W}_B = 22.7\ kW$
$\dot{m}_{fuel} = 1.778 \cdot 10^{-3}\ \ kg/s$

$k_{air} = 1.4$ [1, pp. 847, Table A-2]
$Q_{HV,diesel} = 44000\ \ kJ/kg$ [2]
$V_D = 2.217 \cdot 10^{-3}\ m^3$
$c_{p,air} = 1.005\ kJ/kg \cdot K$
[1, pp. 847, Table A-2]
$T_1 = 24.7°C$
$\dot{m}_{air} = 3.125 \cdot 10^{-2}\ \ kg/s$

Indicated Power

$$\dot{W}_i = \frac{W_i \cdot N_s \cdot n_{cyl}}{n_r} = \frac{0.403 \cdot \frac{1991}{60} \cdot 4}{2} = 26.75\ kW$$

Mechanical Efficiency

$$\eta_{mech} = \frac{\dot{W}_B}{\dot{W}_i} \cdot 100 = \frac{22.7}{26.75} \cdot 100 = 84.6\%$$

Indicated Mean Effective Pressure

$$IMEP = \frac{W_i \cdot n_{cyl}}{V_D} = \frac{0.403 \cdot 4}{2.217 \cdot 10^{-3}} = 727\ kPa$$

Polytropic Coefficients of Compression and Expansion

$$n = \frac{\log(P_2) - \log(P_1)}{\log(V_2) - \log(V_1)} \qquad PV^n = C$$

Compression Stroke

$P_1 = 145.6\ kPa$
$P_2 = 2209\ kPa$

$V_1 = 5.0 \cdot 10^{-4}\ m^3$
$V_2 = 7.0 \cdot 10^{-5}\ m^3$

$$n_{\text{comp}} = \frac{\log(2209) - \log(145.6)}{\log(4.0 \cdot 10^{-5}) - \log(5.0 \cdot 10^{-4})} = 1.383$$

$$C_{\text{comp}} = 145.6 \cdot (5.0 \cdot 10^{-5})^{1.383} = 0.0039567$$

Expansion Stroke

$P_1 = 3101\ kPa$
$P_2 = 403\ kPa$

$V_1 = 1.0 \cdot 10^{-4}\ m^3$
$V_2 = 5.0 \cdot 10^{-4}\ m^3$

$$n_{\text{comp}} = \frac{\log(403) - \log(3101)}{\log(5.0 \cdot 10^{-4}) - \log(1.0 \cdot 10^{-4})} = 1.268$$

$$C_{\text{comp}} = 3101 \cdot (1.0 \cdot 10^{-4})^{1.268} = 0.026306$$

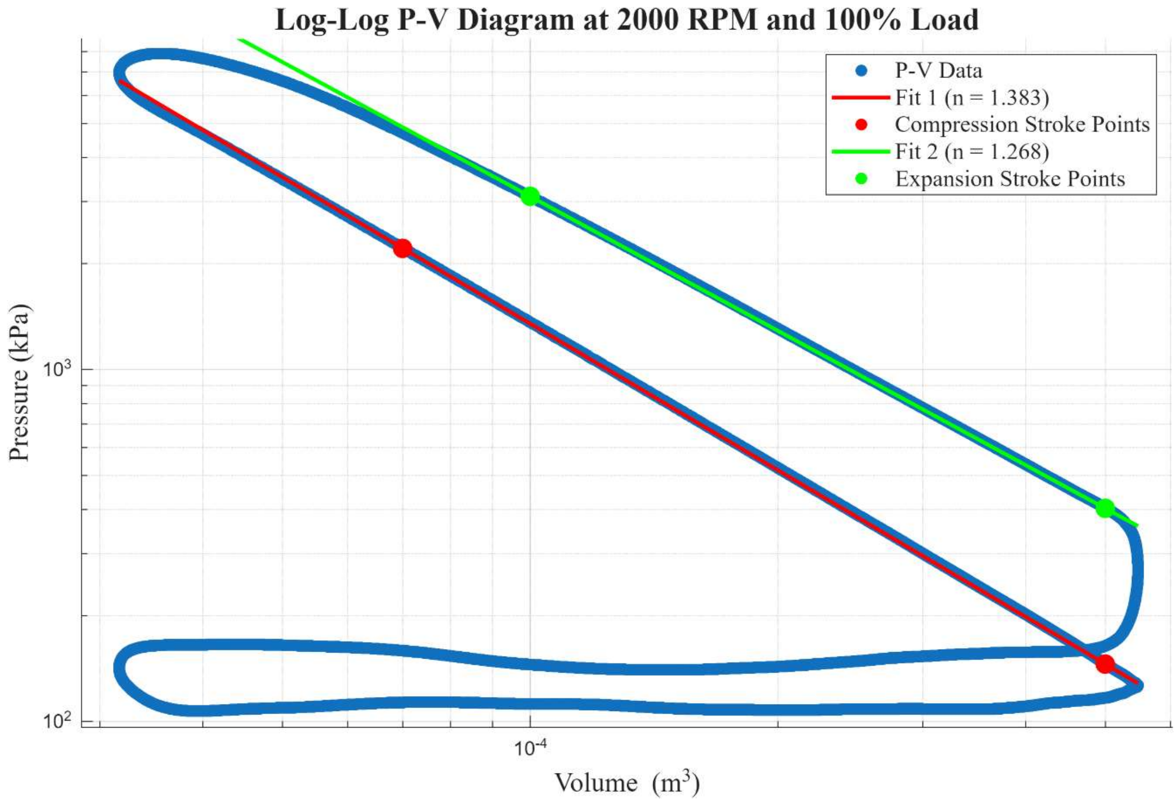

Figure 12: Plot used for finding the polytropic coefficients of compression and expansion at 2000 RPM.

States of Ideal Diesel Cycle

$V_1 = 5.0 \times 10^{-4}\ m^3$ $\quad$ $V_2 = 3.166 \times 10^{-5}\ m^3$

$n_{\text{comp}} = 1.383$ $\quad$ $C_{\text{comp}} = 0.0039567$

$n_{\text{exp}} = 1.268$ $\quad$ $C_{\text{exp}} = 0.026306$

Calculate the pressure and volume at the four different states in the ideal diesel cycle.

$$PV^n = C \qquad P = \frac{C}{V^n} \qquad V = \left(\frac{C}{P}\right)^{\frac{1}{n}}$$

$$P_1 = \frac{C_{comp}}{V_1^{n_{comp}}} = \frac{0.0.0039567}{(5.0 \cdot 10^{-4})^{1.383}} = 142.2\ kPa$$

$$P_2 = \frac{C_{comp}}{V_2^{n_{comp}}} = \frac{0.0039567}{(3.166 \cdot 10^{-5})^{1.383}} = 6608\ kPa$$

$$P_3 = P_2 = 6608\ kPa$$

$$V_3 = \left(\frac{C_{exp}}{P_3}\right)^{\frac{1}{n_{exp}}} = \left(\frac{0.026306}{6608}\right)^{\frac{1}{1.268}} = 5.512 \cdot 10^{-5}\ m^3$$

$$V_4 = V_1 = 5.0 \cdot 10^{-4}\ m^3$$

$$P_4 = \frac{C_{\text{exp}}}{V_4^{n_{\text{exp}}}} = \frac{0.026306}{(5.0 \cdot 10^{-4})^{1.268}} = 399.7\ kPa$$

Calculate the temperature at the different states.

$T_1 = 24.7°C = 297.85\ K$ $\qquad\qquad k_{air} = 1.4$

Compression Ratio: $\quad r = \dfrac{V_1}{V_2} = \dfrac{5.0 \cdot 10^{-4}}{3.166 \cdot 10^{-5}} = 15.79$

$$T_2 = T_1 \cdot r^{k-1} = 297.85 \cdot 15.79^{0.4} = 898\ K$$

$$T_3 = T_2 \cdot \frac{V_3}{V_2} = 898 \cdot \frac{5.512 \cdot 10^{-5}}{3.166 \cdot 10^{-5}} = 1564\ K$$

$$T_4 = T_3 \cdot \left(\frac{1}{r}\right)^{k-1} = 1564 \cdot \left(\frac{1}{15.79}\right)^{0.4} = 519\ K$$

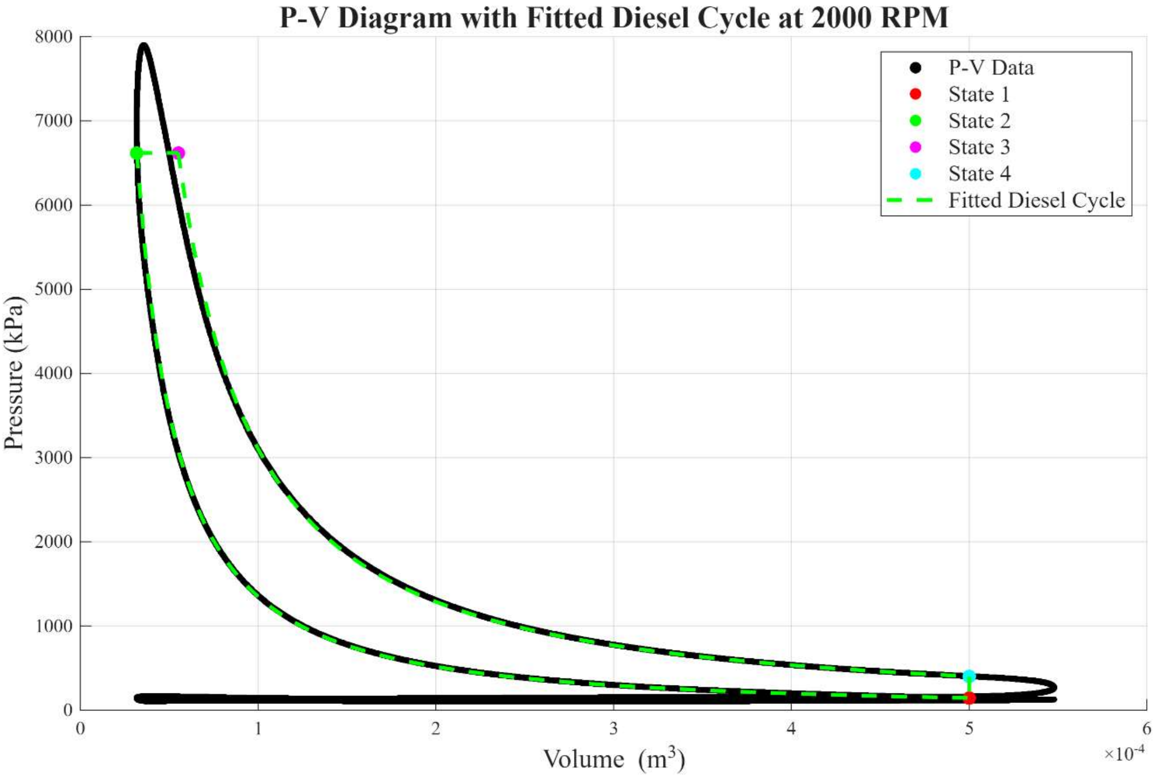

Figure 13: Plot of recorded P-V data and fitted ideal diesel cycle at 2000 RPM.

Cutoff Ratio

$$r_c = \frac{V_3}{V_2} = \frac{5.512 \cdot 10^{-5}}{3.166 \cdot 10^{-5}} = 1.74$$

Heat Input

$$m_{air} = \frac{\dot{m}_{air} \cdot n_r}{N_s \cdot n_{cyl}} = \frac{3.125 \cdot 10^{-2} \cdot 2}{\frac{1991}{60} \cdot 4} = 4.71 \cdot 10^{-4}\ kg$$

$$m_{fuel} = \frac{\dot{m}_{fuel} \cdot n_r}{N_s \cdot n_{cyl}} = \frac{1.778 \cdot 10^{-3} \cdot 2}{\frac{1991}{60} \cdot 4} = 2.68 \cdot 10^{-5}\ kg$$

$$Q_{fuel} = m_{fuel} \cdot Q_{HV,diesel} = 2.68 \cdot 10^{-5} \cdot 44000 = 1.179\ kJ$$

$$Q_{in} = m_{air} \cdot C_{p,air} \cdot (T_3 - T_2) = 4.71 \cdot 10^{-4} \cdot 1.005 \cdot (1564 - 898) = 0.315\ kJ$$

$$\frac{Q_{in}}{Q_{fuel}} \cdot 100 = \frac{0.315}{1.179} \cdot 100 = 26.7\%$$

Diesel vs. Petrol P-V Cycles

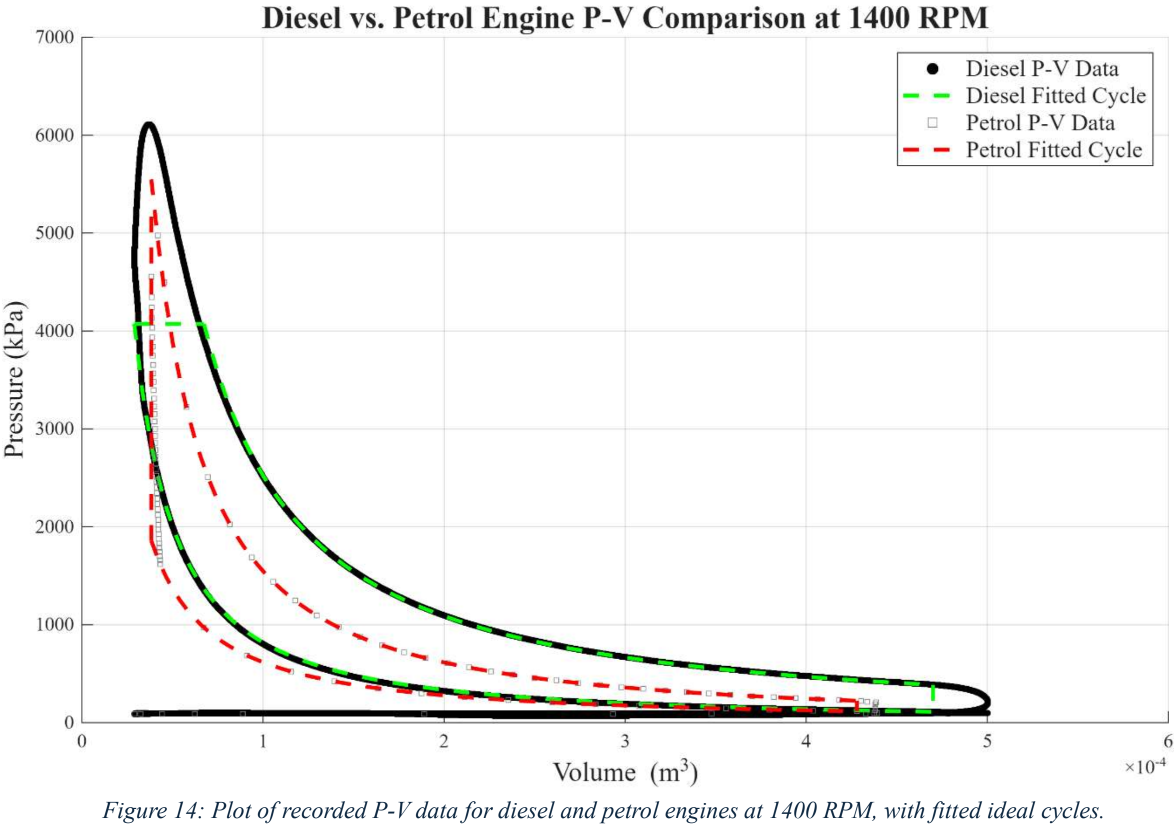

Figure 14: Plot of recorded P-V data for diesel and petrol engines at 1400 RPM, with fitted ideal cycles.

4. Discussion

4.1. Engine Performance

Engine Performance metrics plotted at different RPM and load conditions are found in Figure 7, with these metrics being torque, brake mean effective pressure, air-fuel ratio, volumetric efficiency and thermal efficiency.

Torque and Brake Power

The torque plot represents the torque ay the crank shaft and remains stable across different RPMs at 50% load. At 100% load the torque decreases as RPM increases. This does not align with the manufacturer's performance data which shows the engines peak performance to be at 1800 RPM [2, p. 7].

Brake power is the power output at the crankshaft, measured by the torque and speed at the crankshaft. Under both load conditions the power increases with RPM, in line with the manufacturer’s performance data [2, p. 7]. Due to the limited number of data points the relationship between power and RPM cannot be determined to be linear or non-linear.

BMEP

BMEP represents a single fictitious pressure that if acting on the piston during the entire power stroke would produce the same net work as the actual cycle. This means that a larger BMEP corresponds to more work produced per power cycle [1, p. 456]. The BMEP at 100% load was approximately twice the BMEP at 50% load for the same RPM. This makes sense as having twice the power output would double the BMEP.

SFC and Thermal Efficiency

SFC gives a measure of the fuel required to produce a set amount of work [5, p. 34]. A lower SFC corresponds to a more efficient engine, as less fuel is needed to produce the same work. Under 50% load the SFC decreased with increasing RPM, whereas under 100% load the SFC increased, meaning the engine is less efficient at high load and RPM conditions.

Thermal efficiency is a measure of how much of the fuels heat energy is converted into useful work at the crankshaft [1, p. 256]. A higher thermal efficiency means less fuel is required to produce a set amount of work and is a similar type of metric to SFC. The thermal efficiencies were between 27% and 30%, with the peak at 1800 RPM at 100% load and 2000 RPM at 50% load.

AF and Volumetric Efficiency

The AF is the ratio between the mass of air to the mass of fuel into the engine [1, p. 714]. For diesel fuel the stoichiometric AF is 14.5:1 [6]. At 100% load the engine has an AF of 15-17, meaning that there was more air than is theoretically needed. This excess of air means that complete combustion is more likely to occur, meaning all the fuel reacted with oxygen [1, p. 716]. At 50% load the AF was 30-35, meaning there is a large excess of air going into the engine.

Volumetric efficiency is a measure of how effectively the engine draws in air [5, p. 35]. It is quantified as the ratio of the engine's displacement to the actual volume of air drawn in per cycle. Volumetric efficiency was between 73%-75% at 50% load, and 71%-73% at 100% load, meaning at lower load conditions more air was taken into the engine.

4.2. Dynamometer vs. In-Cylinder Measurements

The difference between the results from the in-cylinder measurements (i.e. the indicated power and indicated mean effective pressure (IMEP)) compared to the dynamometer results (i.e. brake power and BMEP) can be accounted for by mechanical losses when transmitting power from the piston to the crank shaft. These losses will be from friction between moving components, such as the piston rings against the cylinder wall, piston against the crank shaft, and the bearings. This is why the IMEP and indicated power are larger, as not all the work done by the expansion of the combustion gas against the piston goes to the crankshaft. These losses can then be quantified through mechanical efficiency, a measure of how much indicated power is converted into brake power [5, p. 35], which was 94% for 1400 RPM and 85% for 2000 RPM.

4.3. Energy Balance

The energy balance is used to identify and quantify the sources of energy into the engine, and where that energy goes. Through the engine performance analysis, the thermal efficiency was found to be around 30%, meaning that 30% of the energy was converted to useful energy at the crankshaft, while 70% of the energy did not.

Accounted Energy Losses

The largest loss of this energy went to the engines cooling water, to keep the engine at a safe operating temperature. At 1400 RPM, 34% of losses were to the cooling system, and at 41% at 2000 RPM. This increase in proportion may be attributed to the higher engine speed requiring more cooling power, while other energy losses may not increase with RPM to the same degree.

Heat remaining in the exhaust gas accounted for about 10% of the energy for both RPM conditions, though not all the energy from the exhaust gas would be accounted for as the exhaust exiting was still relatively hot.

Unaccounted Energy Losses

Around 20-25% of the energy went to other sources that are not directly accounted for through the dynamometer, cooling system and exhaust systems.

The exhaust gas still carried significant heat energy as seen by the high exhaust output temperatures. The exhaust gas may also contain chemical energy due to incomplete combustion of the fuel.

Heat energy will have been lost through conduction and radiation by the engine body and components to the surrounding environment. Mechanical losses will be from friction between moving components, and the work required to pump gasses into and out of the engine. Some of these frictional losses will be accounted for in the cooling system, and some will be conducted away by the engine.

There is also energy required for the engines auxiliary systems necessary for the function of the engine. These will be the pumps with water, oil and fuel, and the alternator for the electrical system.

4.4. Real Engine Cycle and Air Standard Cycle

The air-standard cycle is an idealised model composed of four reversible processes, isentropic compression (state 1-2), constant pressure heat addition (state 2-3), isentropic compression (state 3-4), and constant volume heat rejection (state 4-1). These ideal processes can be compared to real P-V data to see how they differ.

The air standard cycle assumes that the compression and expansion processes (state 1-2, state 3-4) are isentropic, i.e. they are adiabatic and fully reversible. The real processes are instead polytropic, as heat will be lost through conduction by the cylinder walls, and so cannot be reversed.

In the air standard cycle, the heat addition process (state 2-3) occurs at a constant pressure, taken to be the cylinder pressure at the minimum volume (when the piston is TDC). The P-V data in Figure 11 and Figure 13 shows the pressure continues to rise after TDC. This is because combustion is not an instantaneous process, as fuel is injected over time. The real maximum cylinder pressure is 50% higher than the ideal heat addition.

The air standard cycle assumes that the working fluid is continuously circulated, being returned to its original state through a heat rejection process (state 4-1) under constant volume. The real process involves removing the combustion products after BDC is reached and replacing it with air, making it an open system [1, pp. 459-459]. This process can be seen by the loop in the lower regions of the P-V diagrams (more easily seen in Figure 10 and Figure 12). This exhaust and intake cycle requires work input and is absent from the air standard cycle.

4.5. Diesel vs. Petrol Engine Cycles

The differences between the Diesel and Petrol (Otto) cycle can be seen in Figure 14, which plots the P-V data of the diesel engine against a petrol engine at 1400 RPM. The fundamental difference is their method of ignition, and how this affects the compression and heat addition processes [1, p. 464].

The Petrol cycle compresses an air-fuel mixture, so its compression ratio is limited by not raising the mixture above the auto-ignition temperature, which will cause engine knock which hurts performance and can damage the engine [1, p. 460].

Since the diesel cycle only compresses air, there is no risk of auto-ignition and so can operate at higher compression ratios making them more efficient [1, p. 465], seen in Figure 14 by the Diesel engine having a larger difference between the minimum and maximum volume. The compression ratio for the Petrol engine is 15.2, and 16.3 for the Diesel engine.

The heat addition processes for the Petrol cycle is idealised a constant volume process, which the P-V data shows to be accurate by the vertical spike in pressure when TDC is reached [1, p.

458]. The real Diesel cycle has fuel injected over time at TDC, causing a more gradual spike in pressure, which is idealised as a constant pressure process.

The Diesel cycles higher compression ratio and greater heat addition results in a visibly larger area contained by the P-V curve compared to the Petrol cycle. As the area enclosed by the process curve of a cycle represents the net work by the cycle [1, p. 451], the Diesel cycle can be determined to produce more work per power cycle.

4.6. Experimental Errors and Accuracy

This discussion addresses the potential sources of experimental error, their impact on results, suitability of the instrumentation used and any suggestions for improvement to the experiment.

Methodology and Setup Errors

There is limited test data, as the experiment was only six data points were collected for each metric (three RPMs and two load conditions), which did not cover the full range of the engines rated speed of up to 3000 RPM [2, p. 7]. The experiment was only run once, so the repeatability of the results is unknown. This means trends between RPM, load conditions and engine performance can either not be determined or stated with confidence. This can be improved by running the experiment multiple times and taking measurements across more RPM and load conditions.

The fuel flow rate was already provided on the 'Engine Lab Table', instead of being measured during the experiment. The actual fuel flow rate may be slightly different to the provided values, changing any of the results that relate to the energy input, such as the heat balance, SFC and thermal efficiency. An improvement would be to measure the fuel flow rate during the experiment, so that there is confidence in the fuel consumption data under the current experiment conditions.

Instruments

The manometer had a zero error of 14 mm, meaning calculations for the volume and mass flow rates of air will be inaccurate, affecting any results that use it. This can be fixed by zeroing the manometer correctly. The resolution was 1 mm, giving a relative error of <1%, and had a range suitable to measure the pressure differences of the intake air.

The water and exhaust thermometers had a 6-figure readout, with some results having a resolution to 0.0001 $°C$, though the accuracy to this level is suspect. Even if rounding to 0.1 $°C$ the relative error for temperature differentials would be <1%. The thermometers had suitable range, able to record the high exhaust temperatures, but also the relatively small difference in the input and output water temperatures ($\Delta T \approx 10°C$).

The calorimeter thermometers had a resolution to the whole $°C$. For the temperature differentials calculated using the calorimeter temperatures (for $\dot{Q}_{exhaust}$), this will have a small impact on the results. The small impact the low resolution of the calorimeter thermometers means they were suitable for the experiment.

The cooling water flow meter measured the volume flow rate to 0.1 LPM, which for the recorded values gives a relative error of about 0.3%. This uncertainty will have a small effect on the heat balance equations. The range was suitable for the experiment.

The recorded values from the dynamometer were the RPM, torque and power and throttle position. The range for recorded values was suitable for the experiment, though the engine was not taken to its maximum rated RPM. The resolution of the RPM was to the whole RPM, meaning the relative error is negligible. The torque reading had a relative error around 2%, but it was not directly used in calculations so was not an issue. The power had a resolution to 0.1 kW, giving a relative error of approximately 1%, which will have a minimal impact on the heat balance and efficiency calculations. The dynamometer was suitable for the experiment.

5. Conclusion

Aim of the report was to investigate and analyse engine performance, conduct an energy balance and P-V analysis on a Perkins 404D-22 Diesel engine. This was done by comparing experimental results to the idealised air standard models at different RPM and load conditions.

The engine performance analysis showed that the engine operated with a peak thermal efficiency of 30% at 1800 RPM and 100% load. The maximum power output was observed at the maximum tested speed of 2000 RPM. Thermal efficiency and specific fuel consumption became worse at higher speeds, and mechanical efficiency fell from 94% at 1400 RPM to 85% at 2000 RPM.

The energy balance revealed that 70% of the heat energy input from the fuel was lost and not output as useful work at the crankshaft. The single largest accounted loss was to the cooling system (35%-40%), which is necessary to keep the engine at safe operating conditions. Around 10% of the heat energy was accounted for in the engines exhaust, though this would not be all of it. The remaining 20%-25% was unaccounted for energy, attributed to unaccounted heat in the exhaust, mechanical losses, conduction and radiation to the environment, and powering the engines ancillary systems.

The P-V analysis provided insight into the thermodynamic cycle. Through comparison of the real P-V data and air standard cycle, it was confirmed that real and ideal models do differ. The comparison between the Diesel and Petrol engines showed that the Diesel engine can achieve greater compression ratios and so can be more efficient than a Petrol engine. The Diesel engine was also seen to produce more work per power cycle than the petrol engine.

Despite limitations from the single experimental run and limited data points, the experimental results successfully captured the fundamental trends and validated the key operating principles of a Diesel engine.

6. References

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