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Energy Equilibrium of Diesel Engine IVECO F1C

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Abstract

The purpose of this technical report is to study an internal combustion engine (I.C.E.). More specifically, the energy balance of a Diesel car engine was examined. The technical report includes the description of the laboratory exercise, the experiment's setup, information about the engine under study, the data collected from measurements at the Laboratory of Applied Thermodynamics (L.A.T.), the necessary calculations and the rationale behind them, as well as the requested graphs, the obtained results, comments, and finally, the study's conclusions. The responsible professors are Zisis Samaras, Dimitrios Kolokotronis, and Dimitrios Christoforidis.

Contents

Chapter 1

Introduction

1.1 Summary of laboratory exercise

At the beginning, the engine, its structure and the cooling system were presented. Subsequently, with the assistance of the laboratory personnel, measurements were conducted for four different operating points. Measurements were taken for air and fuel quantities, as well as ambient and exhaust temperatures. Pollutant quantities such as NO_x and $SOOT$ were also monitored, along with the oxygen percentage for various combinations of engine's rotational speed and torque.

The requirements of the exercise for each operating point are:

1. The coolant flow rate.
2. The energy balance and the percentages of energy for each term in the equilibrium.
3. The specific fuel consumption and its relationship to efficiency.
4. The air-fuel equivalence ratio.
5. The specific emissions of NO_x and $SOOT$.

1.2 Engine information

The engine under examination is a 4-stroke, 4-cylinder diesel, liquid-cooled, turbocharged direct injection (TDI) engine with air-to-air intercooling and a common rail injection system. It also complies with Euro 6 emission standards. The characteristics of the engine, as well as the engine itself, are shown in Table 1.1 and Figure 1.1, respectively.

IVECO F1C EuVI Step C	
Engine type	Diesel supercharged with direct injection
Number of cylinders	4
Displacement volume	2998 cm^3
Cylinder diameter (bore)	95.8 mm
Stroke	104 mm
Stroke length	110 mm
Compression rate	17.2
Maximum power	110 kW @ 3500 rpm
Maximum torque	300 Nm @ 1400-3500 rpm
Fuel injection system	Common rail

Table 1.1: IVECO F1C Specifications

Figure 1.1: IVECO F1C Diesel Engine

1.3 Data

The data for the exercise were obtained from tables of enthalpy for air and the theoretical combustion gas, as well as from supporting notes. This data can be found in Figures 1.2 and 1.3, respectively.

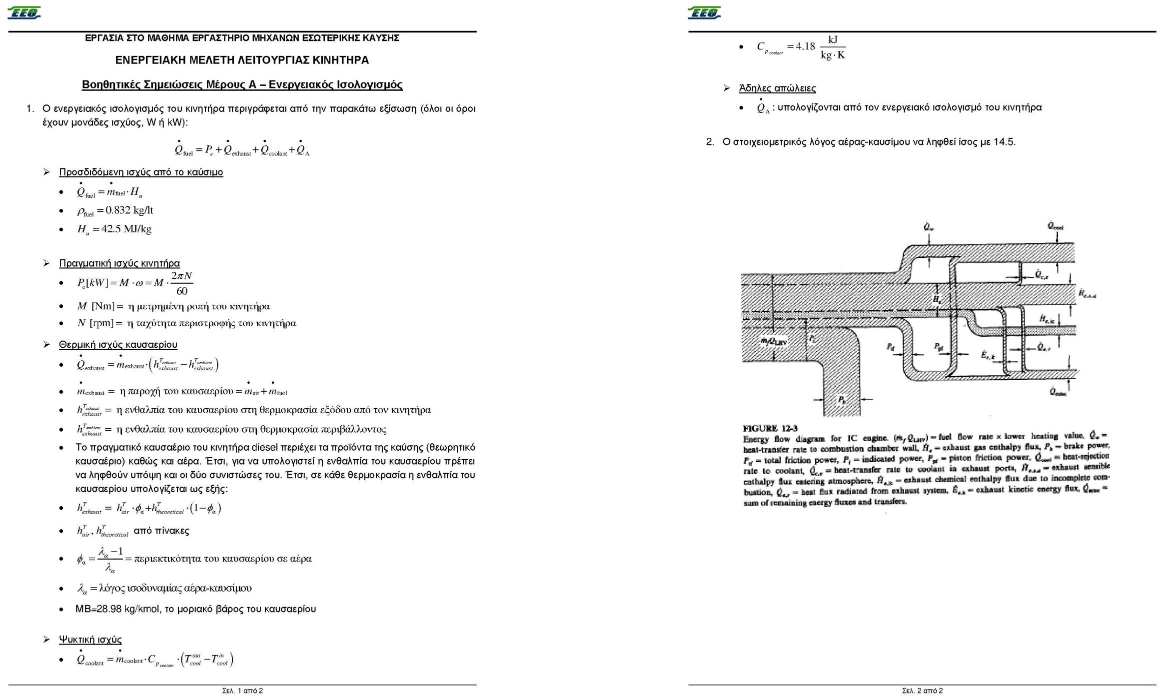
Τιμές μέσης μοριακής θερμοχωρητικότητας, ενθαλπίας και συναρ-
τήσεως Γ_{mp}.

ϑ	ΑΕΡΑΣ (φ_α=1)			ΘΕΩΡΗΤΙΚΟ ΚΑΥΣΑΕΡΙΟ (φ_α=0)		
	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}
°C	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$
0	29.09	0	0.00	30.50	0	0.00
10	29.09	290	1.04	30.53	305	1.09
20	29.10	582	2.05	30.57	611	2.16
30	29.11	873	3.03	30.61	918	3.19
40	29.12	1164	3.97	30.65	1226	4.18
50	29.13	1456	4.89	30.69	1534	5.15
60	29.14	1748	5.78	30.73	1844	6.10
70	29.14	2040	6.64	30.77	2154	7.01
80	29.15	2332	7.48	30.81	2464	7.91
90	29.16	2624	8.30	30.85	2776	8.78
100	29.17	2917	9.09	30.89	3089	9.62
110	29.18	3210	9.87	30.92	3402	10.45
120	29.19	3503	10.62	30.96	3716	11.26
130	29.20	3797	11.36	31.00	4030	12.05
140	29.22	4091	12.08	31.04	4345	12.82
150	29.23	4385	12.79	31.08	4662	13.58
160	29.25	4680	13.48	31.11	4978	14.32
170	29.27	4976	14.15	31.15	5296	15.04
180	29.28	5272	14.81	31.19	5614	15.75
190	29.30	5568	15.46	31.22	5933	16.45
200	29.32	5865	16.09	31.26	6253	17.13
210	29.34	6162	16.71	31.30	6573	17.80
220	29.36	6459	17.32	31.33	6894	18.46
230	29.38	6758	17.92	31.37	7216	19.11
240	29.40	7057	18.51	31.41	75[illegible]3	19.74
250	29.42	7356	19.09	31.45	7863	20.37
260	29.44	7656	19.66	31.49	8188	20.98
270	29.47	7957	20.21	31.53	8514	21.59
280	29.49	8258	20.76	31.57	8840	22.19
290	29.51	8560	21.30	31.61	9168	22.77
300	29.54	8863	21.84	31.65	9496	23.35
310	29.56	9166	22.36	31.69	9825	23.92
320	29.59	9469	22.88	31.73	10155	24.48
330	29.61	9774	23.39	31.77	10486	25.03
340	29.64	10079	23.89	31.82	10818	25.58
350	29.67	10385	24.38	31.86	11152	26.12
360	29.69	10691	24.87	31.90	11485	26.65
370	29.72	10999	25.35	31.94	11820	27.18
380	29.75	11307	25.83	31.99	12156	27.69
390	29.78	11615	26.30	32.03	12493	28.21
400	29.81	11925	26.76	32.07	12831	28.71

ϑ	ΑΕΡΑΣ (φ_α=1)			ΘΕΩΡΗΤΙΚΟ ΚΑΥΣΑΕΡΙΟ (φ_α=0)		
	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}
°C	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$
410	29.84	12235	27.22	32.12	13170	29.21
420	29.87	12545	27.67	32.16	13509	29.70
430	29.90	12857	28.11	32.21	13850	30.19
440	29.92	13169	28.56	32.25	14192	30.67
450	29.95	13481	28.99	32.29	14534	31.15
460	29.98	13795	29.42	32.34	14877	31.62
470	30.01	14109	29.85	32.38	15222	32.09
480	30.05	14424	30.27	32.43	15567	32.55
490	30.08	14739	30.68	32.47	15913	33.01
500	30.11	15055	31.09	32.52	16260	33.46
510	30.14	15372	31.50	32.56	16608	33.91
520	30.17	15689	31.90	32.61	16957	34.35
530	30.20	16007	32.30	32.65	17306	34.79
540	30.23	16326	32.70	32.69	17657	35.22
550	30.26	16645	33.09	32.74	18009	35.65
560	30.29	16965	33.47	32.78	18361	36.08
570	30.32	17286	33.86	32.83	18714	36.50
580	30.35	17608	34.24	32.87	19068	36.91
590	30.39	17930	34.61	32.92	19423	37.33
600	30.42	18253	34.98	32.96	19779	37.74
610	30.45	18576	35.35	33.01	20136	38.14
620	30.48	18900	35.72	33.05	20493	38.55
630	30.51	19224	36.08	33.09	20851	38.95
640	30.54	19550	36.44	33.14	21210	39.34
650	30.57	19875	36.79	33.18	21570	39.73
660	30.60	20202	37.14	33.22	21930	40.12
670	30.64	20528	37.49	33.27	22291	40.51
680	30.67	20856	37.84	33.31	22653	40.89
690	30.70	21184	38.18	33.35	23016	41.27
700	30.73	21513	38.52	33.40	23380	41.64
710	30.76	21842	38.85	33.44	23744	42.01
720	30.79	22171	39.19	33.48	24109	42.38
730	30.82	22502	39.52	33.52	24474	42.75
740	30.85	22833	39.85	33.56	24841	43.11
750	30.88	23164	40.17	33.61	25208	43.47
760	30.91	23496	40.49	33.65	25577	43.83
770	30.94	23828	40.82	33.69	25945	44.19
780	30.97	24161	41.13	33.73	26315	44.54
790	31.00	24495	41.45	33.77	26686	44.89
800	31.03	24829	41.76	33.82	27057	45.24
810	31.06	25164	42.07	33.86	27429	45.58
820	31.09	25499	42.38	33.90	27802	45.93
830	31.12	25834	42.68	33.94	28175	46.27
840	31.15	26170	42.99	33.98	28549	46.60
850	31.18	26507	43.29	34.02	28924	46.94
860	31.21	26844	43.59	34.06	29299	47.27
870	31.24	27181	43.88	34.11	29675	47.60
880	31.27	27519	44.18	34.15	30052	47.93
890	31.30	27857	44.47	34.19	30429	48.26
900	31.32	28196	44.76	34.23	30808	48.58

ϑ	ΑΕΡΑΣ (φ_α=1)			ΘΕΩΡΗΤΙΚΟ ΚΑΥΣΑΕΡΙΟ (φ_α=0)		
	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}	$\bar{c}_{pmo\vartheta}$	$h_{mo\vartheta}$	Γ_{mp}
°C	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole.K}$	$\frac{kJ}{kmole}$	$\frac{kJ}{kmole.K}$
910	31.35	28535	45.05	34.27	31186	48.90
920	31.38	28875	45.33	34.31	31565	49.22
930	31.41	29215	45.62	34.35	31945	49.54
940	31.44	29555	45.90	34.38	32326	49.85
950	31.47	29896	46.18	34.42	32706	50.16
960	31.49	30238	46.46	34.46	33088	50.47
970	31.52	30579	46.73	34.50	33470	50.78
980	31.55	30921	47.01	34.54	33852	51.09
990	31.58	31264	47.28	34.58	34235	51.39
1000	31.60	31607	47.55	34.61	34619	51.70
1010	31.63	31950	47.82	34.65	35003	52.00
1020	31.66	32294	48.08	34.69	35388	52.30
1030	31.68	32638	48.35	34.73	35773	52.59
1040	31.71	32983	48.61	34.76	36158	52.89
1050	31.74	33328	48.88	34.80	36544	53.18
1060	31.76	33673	49.14	34.84	36931	53.47
1070	31.79	34018	49.39	34.87	37318	53.76
1080	31.81	34364	49.65	34.91	37705	54.05
1090	31.84	34711	49.91	34.94	38093	54.33
1100	31.87	35058	50.16	34.98	38481	54.62
1110	31.89	35405	50.41	35.01	38870	54.90
1120	31.92	35752	50.66	35.05	39259	55.18
1130	31.94	36100	50.91	35.08	39649	55.46
1140	31.97	36448	51.16	35.12	40039	55.73
1150	31.99	36796	51.40	35.15	40430	56.01
1160	32.02	37145	51.65	35.19	40821	56.28
1170	32.04	37494	51.89	35.22	41213	56.56
1180	32.07	37844	52.13	35.25	41605	56.83
1190	32.09	38193	52.37	35.29	41998	57.10
1200	32.12	38544	52.61	35.32	42392	57.36
1210	32.14	38894	52.85	35.36	42785	57.63
1220	32.16	39245	53.08	35.39	43180	57.90
1230	32.19	39596	53.32	35.42	43574	58.16
1240	32.21	39947	53.55	35.45	43969	58.42
1250	32.23	40298	53.78	35.49	44365	58.68
1260	32.26	40650	54.01	35.52	44761	58.94
1270	32.28	41002	54.24	35.55	45157	59.20
1280	32.30	41354	54.47	35.58	45554	59.46
1290	32.33	41706	54.69	35.62	45951	59.71
1300	32.35	42059	54.92	35.65	46349	59.96
1310	32.37	42412	55.14	35.68	46746	60.22
1320	32.39	42765	55.36	35.71	47145	60.47
1330	32.42	43119	55.59	35.74	47544	60.72
1340	32.44	43472	55.81	35.77	47943	60.96
1350	32.46	43826	56.02	35.80	48342	61.21
1360	32.48	44180	56.24	35.84	48742	61.46
1370	32.50	44535	56.46	35.87	49142	61.70
1380	32.52	44889	56.67	35.90	49543	61.94
1390	32.55	45244	56.89	35.93	49944	62.19
1400	32.57	45600	57.10	35.96	50345	62.43

Figure 1.2: Enthalpy tables



ΕΡΓΑΣΙΑ ΣΤΟ ΜΑΘΗΜΑ ΕΡΓΑΣΤΗΡΙΟ ΜΗΧΑΝΩΝ ΕΣΩΤΕΡΙΚΗΣ ΚΑΥΣΗΣ

ΕΝΕΡΓΕΙΑΚΗ ΜΕΛΕΤΗ ΛΕΙΤΟΥΡΓΙΑΣ ΚΙΝΗΤΗΡΑ

Βοηθητικές Σημειώσεις Μέρους Α – Ενεργειακός Ισολογισμός

1. Ο ενεργειακός ισολογισμός του κινητήρα περιγράφεται από την παρακάτω εξίσωση (όλοι οι όροι έχουν μονάδες ισχύος, W ή kW):

$$\dot{Q}_{\text{fuel}} = P_e + \dot{Q}_{\text{exhaust}} + \dot{Q}_{\text{coolant}} + \dot{Q}_{\text{A}}$$

- Προσδιδόμενη ισχύς από το καύσιμο
  - $\dot{Q}_{\text{fuel}} = \dot{m}_{\text{fuel}} \cdot H_u$
  - $\rho_{\text{fuel}} = 0.832$ kg/lt
  - $H_u = 42.5$ MJ/kg
- Πραγματική ισχύς κινητήρα
  - $P_e[kW] = M \cdot \omega = M \cdot \frac{2\pi N}{60}$
  - M [Nm] = η μετρημένη ροπή του κινητήρα
  - N [rpm] = η ταχύτητα περιστροφής του κινητήρα
- Θερμική ισχύς καυσαερίου
  - $\dot{Q}_{\text{exhaust}} = \dot{m}_{\text{exhaust}} \cdot \left(h_{exhaust}^{T_{exhaust}} - h_{exhaust}^{T_{ambient}}\right)$
  - $\dot{m}_{\text{exhaust}} =$ η παροχή του καυσαερίου $= \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}$
  - $h_{exhaust}^{T_{exhaust}} =$ η ενθαλπία του καυσαερίου στη θερμοκρασία εξόδου από τον κινητήρα
  - $h_{exhaust}^{T_{ambient}} =$ η ενθαλπία του καυσαερίου στη θερμοκρασία περιβάλλοντος
  - Το πραγματικό καυσαέριο του κινητήρα diesel περιέχει τα προϊόντα της καύσης (θεωρητικό καυσαέριο) καθώς και αέρα. Έτσι, για να υπολογιστεί η ενθαλπία του καυσαερίου πρέπει να ληφθούν υπόψη και οι δύο συνιστώσες του. Έτσι, σε κάθε θερμοκρασία η ενθαλπία του καυσαερίου υπολογίζεται ως εξής:
  - $h_{exhaust}^{T} = h_{air}^{T} \cdot \phi_\alpha + h_{theoretical}^{T} \cdot (1 - \phi_\alpha)$
  - $h_{air}^{T}, h_{theoretical}^{T}$ από πίνακες
  - $\phi_\alpha = \frac{\lambda_\alpha - 1}{\lambda_\alpha} =$ περιεκτικότητα του καυσαερίου σε αέρα
  - $\lambda_\alpha =$ λόγος ισοδυναμίας αέρα-καυσίμου
  - MB=28.98 kg/kmol, το μοριακό βάρος του καυσαερίου
- Ψυκτική ισχύς
  - $\dot{Q}_{\text{coolant}} = \dot{m}_{\text{coolant}} \cdot C_{p_{coolant}} \cdot \left(T_{cool}^{out} - T_{cool}^{in}\right)$

Σελ. 1 από 2

- $C_{p_{coolant}} = 4.18 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$
- Άδηλες απώλειες
  - $\dot{Q}_{\text{A}}$: υπολογίζονται από τον ενεργειακό ισολογισμό του κινητήρα

2. Ο στοιχειομετρικός λόγος αέρας-καυσίμου να ληφθεί ίσος με 14.5.

FIGURE 12-3
Energy flow diagram for IC engine. $(\dot{m}_f Q_{LHV})$ = fuel flow rate × lower heating value, $\dot{Q}_w$ = heat-transfer rate to combustion chamber wall, $\dot{H}_e$ = exhaust gas enthalpy flux, P_b = brake power, P_{tf} = total friction power, P_i = indicated power, P_{pf} = piston friction power, $\dot{Q}_{cool}$ = heat-rejection rate to coolant, $\dot{Q}_{c,e}$ = heat-transfer rate to coolant in exhaust ports, $\dot{H}_{e,s,a}$ = exhaust sensible enthalpy flux entering atmosphere, $\dot{H}_{e,ic}$ = exhaust chemical enthalpy flux due to incomplete combustion, $\dot{Q}_{e,r}$ = heat flux radiated from exhaust system, $\dot{E}_{e,k}$ = exhaust kinetic energy flux, $\dot{Q}_{misc}$ = sum of remaining energy fluxes and transfers.

Σελ. 2 από 2

Figure 1.3: Notes

1.4 Measurements

The measurements were taken at four operating points: the first one at 2000 rpm and 120 Nm, the second one at 2000 rpm and 200 Nm, the third one at 2300 rpm and 80 Nm and the fourth one at 2300 rpm and 160 Nm. The measurements from **GROUP 12** which will be used for calculations, are shown in Table 1.2.

#	$Speed$	$Torque$	$Throttle$	T_{exh}	T_{air}	$T_{w,in}/T_{w,out}$	$Airflow$	INJ	NO_x	$SOOT$	O_2
-	rpm	Nm	%	$°C$	$°C$	$°C$	Kg/h	mg/str	ppm	mg/m^3	%
1	2000	120	41	387	23.4	83/84	133	26.7	70	30	6.1
2	2000	200	52.5	468	25.4	81/84	185	42.6	92	115	3.8
3	2300	80	33.7	321	25.4	83/84	142	18.5	94	26	9.3
4	2300	160	50.3	441	26.1	81/84	190	34.7	86	110	4.9

Table 1.2: GROUP 12 Measurements

Where:

1. T_{exh} - Exhaust gas temperature
2. T_{air} - Inlet air temperature
3. $T_{w,in}$ - Coolant (water) inlet temperature
4. $T_{w,out}$ - Coolant (water) outlet temperature
5. $Airflow$ - Mass flow rate of air at the intake for all four cylinders
6. INJ - Fuel mass flow rate for one cylinder in one thermodynamic cycle

Chapter 2

Calculations

2.1 Question 1

For the calculations, an Excel spreadsheet was created and theory from the course's supporting notes was used (Figure 1.3).

In the first question, the mass flow rate of the coolant, which in this case is water, needs to be calculated. The flow can be calculated using the equation:

$$\dot{Q}_{coolant} = \dot{m}_{coolant} \cdot c_{p_{coolant}} \cdot (T^{out}_{coolant} - T^{in}_{coolant}) \quad (W) \tag{2.1}$$

However, with the given assumption that the power of the coolant is 30% of the power of the fuel:

$$\dot{Q}_{coolant} = 30\% \cdot \dot{Q}_{fuel} \tag{2.2}$$

It is sufficient to find the power of the fuel. That is calculated as:

$$\dot{Q}_{fuel} = \dot{m}_{fuel} \cdot H_u \quad (W) \tag{2.3}$$

where:

- $\boldsymbol{\dot{m}_{fuel}}$ is the mass flow rate of the fuel
- $\boldsymbol{H_u}$ is the calorific value of the fuel, which is known and given as $H_u = 42.5\ MJ/kg$

To calculate the fuel flow for each operating point, the measurements of $\boldsymbol{INJ}$, which shows the mass flow rate of the fuel for 1 cylinder in 1 thermodynamic cycle, from Table 1.2 are used. Also, since the engine is 4-stroke, one thermodynamic cycle corresponds to 720°, which is equivalent to 2 rotations (1 $stroke$ = 2 $rotations$). Therefore, for the 4 cylinders, it will be:

$$\dot{m}_{fuel} = 4\ cylinders \cdot INJ(\frac{mg}{stroke}) \cdot \frac{rotations}{min} <=>$$
$$\dot{m}_{fuel} = 4\ cylinders \cdot INJ \cdot \frac{mg}{2\ rotations} \cdot \frac{rotations}{min} <=>$$
$$\dot{m}_{fuel} = 4\ cylinders \cdot INJ \cdot \frac{rpm}{2} \cdot \frac{10^{-6} kg}{60\ seconds} <=>$$
$$\dot{m}_{fuel} = \frac{INJ \cdot rpm}{30000000} \quad (kg/s) \tag{2.4}$$

Thus, by using Equation 2.4, the fuel mass flow rates for each operating point are calculated through the spreadsheet and shown in Table 2.1.

#	INJ	$\dot{m}_{fuel}$
-	mg/str	kg/s
1	26.7	0.00178
2	42.6	0.00284
3	18.5	0.001418333
4	34.7	0.002660333

Table 2.1: Fuel mass flow rates for each operating point

Once the fuel mass flow rates are known, the power of the fuel can be calculated from Equation 2.3 using the known $H_u = 42500000\ J/kg$. Through these values, the power of the coolant is also calculated using Equation 2.2. Finally, the **mass flow rate of the coolant** is calculated with the known specific heat capacity of the coolant (water) $c_{p_{coolant}} = 4180\ J/kg \cdot K$ and a rearrangement of Equation 2.1 as follows:

$$\dot{m}_{coolant} = \frac{\dot{Q}_{coolant}}{c_{p_{coolant}} \cdot (T^{out}_{coolant} - T^{in}_{coolant})} \quad (kg/s)$$

Using the values of $T^{out}_{coolant}$ $(T_{w,in})$ and $T^{in}_{coolant}$ $(T_{w,out})$ for each operating point from Table 1.2, these values are calculated through the spreadsheet and shown in Table 2.2.

#	$\dot{m}_{fuel}$	$\dot{Q}_{fuel}$	$\dot{Q}_{coolant}$	$\dot{m}_{coolant}$
-	kg/s	W	W	kg/s
1	0.00178	75650	22695	5.43
2	0.00284	120700	36210	2.89
3	0.001418333	60279.17	18083.75	4.33
4	0.002660333	113064.17	33919.25	2.70

Table 2.2: Coolant mass flow rates for each operating point

The values of the coolant mass flow rates are also shown in a diagram for constant rotational speed of the engine in Figure 2.1. It can be observed that for a **constant** number of revolutions N and as the engine torque M **increases**, the mass flow rate of the coolant $\dot{m}_{fuel}$ **decreases**.

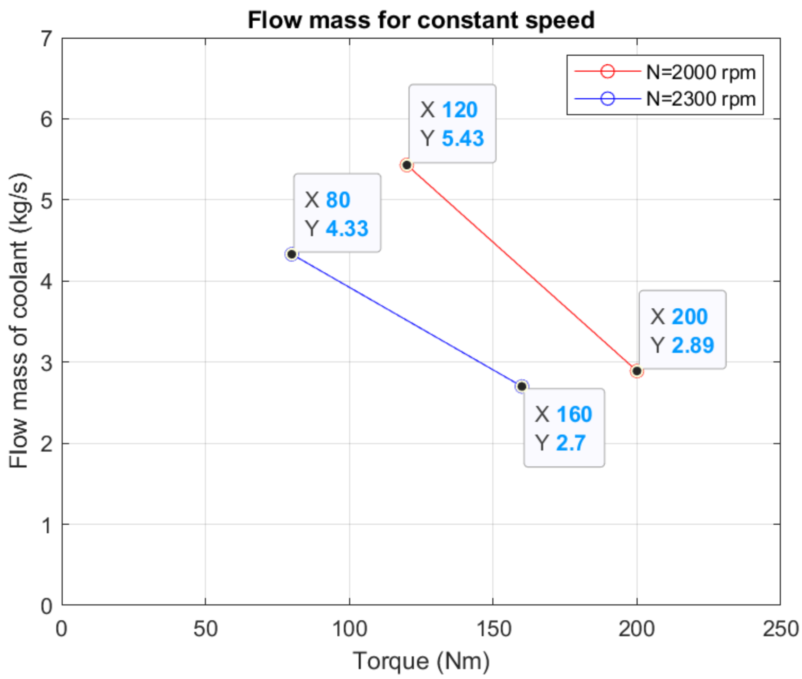


Figure 2.1: Coolant mass flow rate as a function of torque at constant RPM.

2.2 Question 2

The second question concerns the energy balance of the engine and the percentages of energy distribution for each term. The energy balance of the engine is described by the following equation:

$$\dot{Q}_{fuel} = P_e + \dot{Q}_{exhaust} + \dot{Q}_{coolant} + \dot{Q}_A \tag{2.5}$$

where:

- $\boldsymbol{P_e}$ is the engine's true power.
- $\boldsymbol{\dot{Q}_A}$ represents the unaccounted losses.
- $\boldsymbol{\dot{Q}_{exhaust}}$ stands for the thermal power of the exhaust gas.

The power of the fuel $\boldsymbol{\dot{Q}_{fuel}}$ and the coolant $\boldsymbol{\dot{Q}_{coolant}}$ are shown in Table 2.2. Therefore, only the engine power, thermal power and unaccounted losses need to be calculated. P_e is calculated as follows:

$$P_e = M \cdot \omega <=>$$
$$P_e = M \cdot \frac{2\pi \cdot N}{60} \quad (W)$$

With known values of torque M (Nm) and speed N (rpm), the values of P_e (W) are derived for each of the four operating points in Table 2.3.

#	N	M	P_e
-	rpm	Nm	W
1	2000	120	25132.74
2	2000	200	41887.90
3	2300	80	19268.43
4	2300	160	38536.87

Table 2.3: Actual engine power at operating points

To calculate $\dot{Q}_{exhaust}$, the following equation is used:

$$\dot{Q}_{exhaust} = \dot{m}_{exhaust} \cdot (h_{exhaust}^{T_{exhaust}} - h_{exhaust}^{T_{ambient}}) \quad (W) \tag{2.6}$$

where:

- $\boldsymbol{\dot{m}_{exhaust}}$ is the mass flow rate of the exhaust gas.
- $\boldsymbol{h_{exhaust}^{T_{exhaust}}}$ is the enthalpy of the exhaust gas at the exit temperature from the engine.
- $\boldsymbol{h_{exhaust}^{T_{ambient}}}$ is the enthalpy of the exhaust gas at ambient temperature.

The conservation of mass applies and therefore:

$$\dot{m}_{exhaust} = \dot{m}_{air} + \dot{m}_{fuel} \tag{2.7}$$

The mass flow rate of fuel has been calculated in Table 2.1 and the mass flow rate of air is obtained from the experimental measurements per hour ($Airflow$) in Table 1.2. Therefore, from equation 2.7, the mass flow rate of the exhaust gas is calculated after converting $Airflow$ from kg/h to kg/s, as shown in Table 2.4.

The enthalpy of the exhaust gas is calculated each time using the enthalpy of theoretical fuel and air as follows:

$$h_{exhaust}^{T} = h_{air}^{T} \cdot \phi_\alpha + h_{theoretical}^{T} \cdot (1 - \phi_\alpha) \tag{2.8}$$

Here, the enthalpies of air and theoretical fuel are given by the enthalpy tables (Figure 1.2) for various temperature values. The temperatures of interest are $T_{exhaust}$ and T_{air} from Table 1.2. Using linear interpolation between the values of $380 - 390°C$ and $20 - 30°C$, the values of $h_{air}^{T_{exhaust}}$, $h_{theoretical}^{T_{exhaust}}$, $h_{air}^{T_{ambient}}$, and $h_{theoretical}^{T_{ambient}}$ are obtained. The rest of the enthalpies are similarly calculated. These values are shown in Table 2.5.

#	$Airflow$	$\dot{m}_{fuel}$	$\dot{m}_{exhaust}$
-	kg/s	kg/s	kg/s
1	0.036944444	0.00178	0.038724444
2	0.051388889	0.00284	0.054228889
3	0.039444444	0.001418333	0.040862778
4	0.052777778	0.002660333	0.055438111

Table 2.4: Mass flow rate values for each operating point

#	$h_{air}^{T_{ambient}}$	$h_{air}^{T_{exhaust}}$	$h_{theoretical}^{T_{ambient}}$	$h_{theoretical}^{T_{exhaust}}$
-	$kJ/kmole$	$kJ/kmole$	$kJ/kmole$	$kJ/kmole$
1	680.94	11522.6	715.38	12391.9
2	747.87	14046.2	785.99	15153
3	747.87	9499.5	785.99	10188.1
4	759.51	13200.2	798.27	14226.2

Table 2.5: Enthalpies of air and theoretical fuel at temperatures of interest

Next, the air-to-fuel ratio is calculated for each operating point as follows:

$$\lambda_{\alpha} = \frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

From these values, the air content of the fuel is determined as:

$$\phi_{\alpha} = \frac{\lambda_{\alpha} - 1}{\lambda_{\alpha}}$$

With the known molecular weight of the fuel, which is $MB = 28.98\ kg/kmole$ and simple division by the values in Table 2.5, the required enthalpies $\boldsymbol{h_{exhaust}^{T_{ambient}}}$ and $\boldsymbol{h_{exhaust}^{T_{exhaust}}}$ in $\boldsymbol{kJ/kg}$ are calculated using equation 2.8. These values are summarized in Table 2.6.

#	λ_{α}	ϕ_{α}	$h_{exhaust}^{T_{ambient}}$	$h_{exhaust}^{T_{exhaust}}$
-	-	-	kJ/kg	kJ/kg
1	20.75530587	0.951819549	399.0504923	23.55415234
2	18.09467919	0.944735135	486.7966581	25.87911307
3	27.81041911	0.964042254	328.6494308	25.85371668
4	19.83878378	0.949593684	457.2780152	26.27549168

Table 2.6: Combustion gas enthalpies, air-fuel ratio and air content of the fuel

From the values of the exhaust gas enthalpies (Table 2.6), the mass flow rate of exhaust gas (Table 2.4) and equation 2.6, the **exhaust gas power** is calculated in kJ/s. Multiplying by 1000 the values are converted in $J/s = Watts$. These values are shown in Table 2.7.

#	$\dot{Q}_{exhaust}$
-	W
1	14540.89
2	24995.05
3	12373.07
4	23893.97

Table 2.7: Exhaust gas power at operating points

Now, from equation 2.5, the **latent power** $\boldsymbol{\dot{Q}_A}$ can be calculated since all the other terms are known. All the values of the terms in the energy balance, as well as the % of the provided energy for each operating point, are given in Table 2.8.

#	$\dot{Q}_{exhaust}$		$\dot{Q}_{coolant}$		P_e		$\dot{Q}_A$		$\dot{Q}_{fuel}$	
-	W	%	W	%	W	%	W	%	W	%
1	14540.89	19.22	22695	30	25132.74	33.22	13281.37	17.56	75650	100
2	24995.05	20.71	36210	30	41887.90	34.70	17607.05	14.59	120700	100
3	12373.07	20.53	18083.75	30	19268.43	31.97	10553.91	17.51	60279.17	100
4	23893.97	21.13	33919.25	30	38536.87	34.08	16714.08	14.78	113064.17	100

Table 2.8: Engine energy balance

In Figure 2.2, the energy balance of the engine for various torque values for constant speed N is shown.

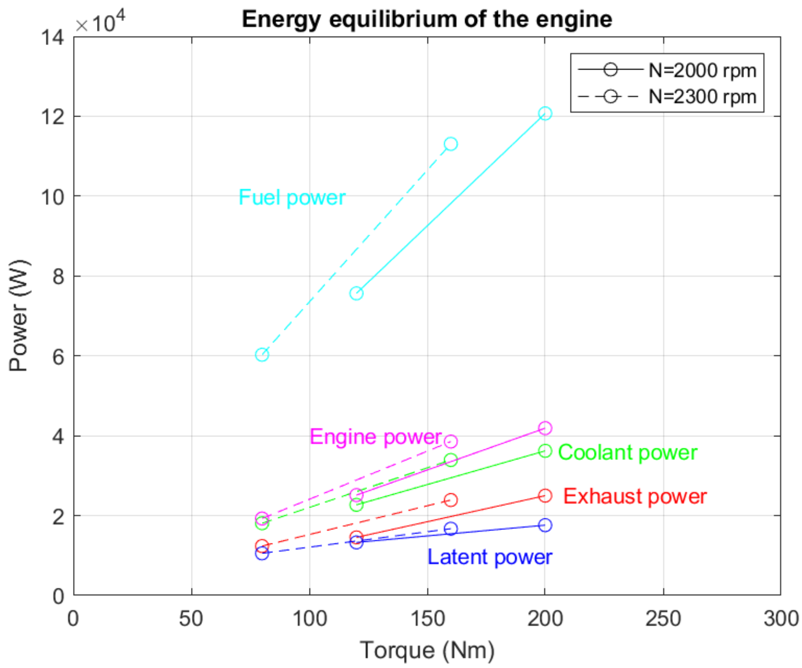


Figure 2.2: Engine energy equilibrium

It is observed that for **constant** speed N and as the torque M **increases**, the power values **also increase**.

2.3 Question 3

In question three, the specific fuel consumption (SFC) for each operating point is requested. Having gathered data from the previous questions, it is calculated as:

$$specific\ fuel\ consumption\ (SFC) = \frac{rate\ of\ fuel\ consumption}{energy\ produced\ by\ the\ engine}$$

This translates to:

$$SFC = \frac{\dot{m}_{fuel}}{P_e} \quad (gr/kWh) \tag{2.9}$$

where $\boldsymbol{\dot{m}_{fuel}}$ is in $\boldsymbol{gr/h}$ and $\boldsymbol{P_e}$ is in $\boldsymbol{kW}$. For these, it holds that:

$$\dot{m}_{fuel}\ (gr/h) = \dot{m}_{fuel}\ (kg/s) \cdot 3.6 \times 10^6 \quad \& \quad P_e\ (kW) = 1000 \cdot P_e\ (W) \tag{2.10}$$

Also, for the efficiency, it holds that:

$$\eta = \frac{work\ done\ by\ the\ engine}{heat\ absorbed\ by\ the\ engine} <=>$$

$$\eta = \frac{P_e}{\dot{Q}_{fuel}} \tag{2.11}$$

Therefore, the relationship between efficiency and specific fuel consumption can be easily derived as:

$$\eta = \frac{P_e}{\dot{Q}_{fuel}} <=>$$

$$\eta = \frac{P_e}{\dot{m}_{fuel} \cdot H_u} <=>$$

$$\eta = \frac{1}{SFC \cdot H_u} \tag{2.12}$$

The values of efficiency and specific fuel consumption are obtained from Tables 2.2 and 2.3, as well as from equations 2.9, 2.10, and 2.11 and are shown in Table 2.9:

#	$\dot{m}_{fuel}$	P_e	SFC	η	η
-	gr/h	kW	gr/kWh	-	%
1	6408	25.13	254.97	0.3322	33.22
2	10224	41.89	244.08	0.3470	34.70
3	5106	19.27	264.99	0.3197	31.97
4	9577.2	38.54	248.52	0.3408	34.08

Table 2.9: Specific fuel consumption and efficiency

In Figure 2.3, the specific fuel consumption and efficiency are shown as a function of engine torque for constant engine speed. It is observed that for **constant** engine speed N and as the torque M **increases**, the specific fuel consumption SFC **decreases**, while efficiency η slightly **increases**.

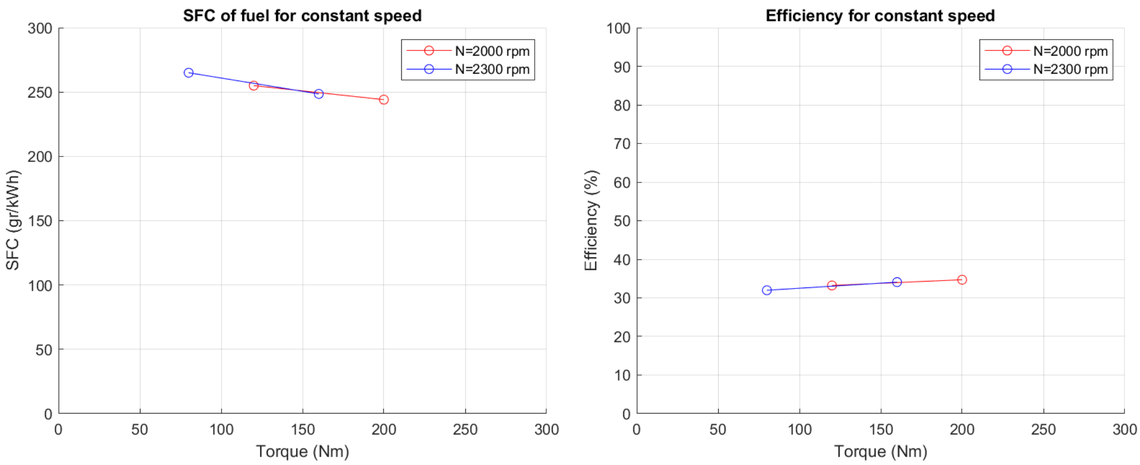


Figure 2.3: Specific fuel consumption and efficiency for constant RPM.

2.4 Question 4

In the fourth question, the air-fuel equivalence ratio (AFR) is requested for each operating point. This ratio is defined as:

$$AFR = \frac{\lambda_{real}}{\lambda_{stoich}} \tag{2.13}$$

Where λ_{stoich} is known and given as $\boldsymbol{\lambda_{stoich}}$ **= 14.5**, while $\lambda_{real} = \lambda_a$, which has been calculated and is presented in Table 2.6. Therefore, the values of AFR can be easily computed using equation 2.13. These values are provided in Table 2.10.

#	λ_a	AFR
-	-	-
1	20.7553	1.4314
2	18.0947	1.2479
3	27.8104	1.9180
4	19.8388	1.3682

Table 2.10: Air-Fuel Equivalence Ratio Values for Operating Points

In Figure 2.4, the values of the air-fuel equivalence ratio are plotted for the operating points. It is observed that for a **constant** engine speed N and as the torque M **increases**, the air-fuel equivalence ratio AFR **decreases**.

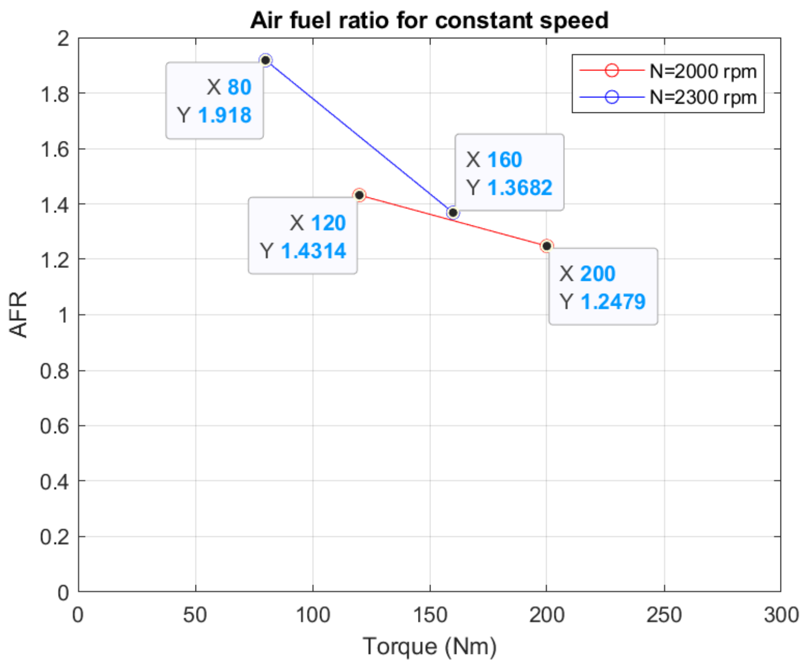


Figure 2.4: Air-fuel equivalence ratio for constant RPM.

2.5 Question 5

In the fifth and final question, the specific emissions of NO_x and $SOOT$ are requested. Similar to the specific fuel consumption (Equation 2.9), specific emissions can be defined as follows:

$$SNO_xE = \frac{\dot{m}_{NO_x}}{P_e} \quad (mg/kWh) \tag{2.14}$$

$$SSOOTE = \frac{\dot{m}_{SOOT}}{P_e} \quad (mg/kWh) \tag{2.15}$$

To calculate these specific emissions, we need to first calculate the mass flows of pollutants. Based on the data in Table 1.2, we have the concentration of NO_x in parts per million (ppm). To convert this to mg/h, the following process is applied:

$$NOx\ (ppm) = \frac{n_{solute}\ (mol)}{n_{solution}\ (mol)} \times 10^6 <=>$$

$$NOx\ (ppm) = \frac{\dot{n}_{solute}\ (mol/s)}{\dot{n}_{solution}\ (mol/s)} \times 10^6 <=>$$

$$NOx\ (ppm) = \frac{\dot{n}_{NO_x}\ (mol/s)}{\dot{n}_{exhaust}\ (mol/s)} \times 10^6 <=>$$

$$NOx\ (ppm) = \frac{\frac{\dot{m}_{NO_x}\ (kg/s)}{MB_{NO_x}\ (kg/mol)}}{\frac{\dot{m}_{exhaust}\ (kg/s)}{MB_{exhaust}\ (kg/mol)}} \times 10^6 <=>$$

$$\dot{m}_{NO_x}\ (kg/s) = \frac{\dot{m}_{exhaust}\ (kg/s) \cdot NO_x\ (ppm) \cdot MB_{NO_x}\ (kg/mol)}{MB_{exhaust}\ (kg/mol)} \times 10^{-6} \tag{2.16}$$

$$\dot{m}_{NO_x}\ (mg/h) = \dot{m}_{NO_x}\ (kg/s) \cdot 36 \times 10^8 \tag{2.17}$$

Given the values of molecular weights, $MB_{NO_x} = 0.0460055 kg/mol$ and $MB_{exhaust} = 0.02898 kg/mol$, the values of the mass flow rate of NO_x can be calculated as shown in Table 2.11.

#	NO_x	$\dot{m}_{NO_x}$	$\dot{m}_{NO_x}$
-	ppm	$\times 10^{-6}\ kg/s$	mg/h
1	70	4.303	15491.63
2	92	7.920	28512.31
3	94	6.098	21951.77
4	86	7.569	27247.13

Table 2.11: Mass flow rate of NO_x values for each operating point

Therefore, using Equation 2.14, the specific emissions can be calculated from the mass flow rate of NO_x (mg/h) and the power values from Table 2.9. The calculated specific emissions are shown in Table 2.12.

#	SNO_xE
-	mg/kWh
1	616.39
2	680.68
3	1139.26
4	707.04

Table 2.12: Specific emissions of NO_x for the operating points

To calculate the specific emissions of $SOOT$, we need to determine the mass flow rate again. It is known from Thermodynamics that a gas mixture can be considered as a new ideal gas. Thus, the exhaust gas mixture can be considered an ideal gas for which the Ideal Gas Equation is written as:

$$P \cdot V = n \cdot R \cdot T <=>$$

$$P = \frac{\rho_{exhaust}}{MB_{exhaust}} \cdot R \cdot T_{exhaust} <=>$$

$$\rho_{exhaust} = \frac{P \cdot MB_{exhaust}}{R \cdot T_{exhaust}}$$

where:

- $\boldsymbol{P}$ is the pressure at which the exhaust gas is located, and it is equal to atmospheric pressure, $10^5\ Pa$
- $\boldsymbol{R}$ is the universal gas constant for gases, and it is equal to 8.3145 $J/mol \cdot K$
- $\boldsymbol{MB_{exhaust}}$ is the molecular weight of the exhaust gas, given as 28.98 $kg/kmol$
- $\boldsymbol{T_{exhaust}}$ is the temperature at which the exhaust gas is located for each operating point and is given by measurements

Summarizing, we have:

$$\rho_{exhaust} = \frac{10^5\ (Pa) \cdot 28.98\ (kg/kmol)}{8.3145 \times 10^{-3}\ (J/kmol \cdot K) \cdot T_{exh}}$$

or finally:

$$\rho_{exhaust} = \frac{348.548}{T_{exh}}\ (kg/m^3) \tag{2.18}$$

So, from Table 1.2 and Equation 2.18, the density of the exhaust gas is calculated at each operating point. Then, by dividing these values by the mass flow rates of the exhaust gas from Table 2.4, the volumetric flow rates of the exhaust gas are obtained according to the equation:

$$\dot{V}_{exhaust}\ (m^3/s) = \frac{\dot{m}_{exhaust}\ (kg/s)}{\rho_{exhaust}\ (kg/m^3)}$$

These values are shown in Table 2.13.

#	$\rho_{exhaust}$	$\dot{m}_{exhaust}$	$\dot{V}_{exhaust}$
-	kg/m^3	kg/s	m^3/s
1	0.52810	0.03872	0.073328
2	0.47037	0.05423	0.115289
3	0.58678	0.04086	0.069639
4	0.48816	0.05544	0.113565

Table 2.13: Values of density, mass flow rate, and volume flow rate of the exhaust gas for each operating point

Since the pollutant is contained in the exhaust gas flow rate $\dot{V}_{exhaust}$, we have:

$$\dot{m}_{SOOT}\ (mg/s) = SOOT\ (mg/m^3) \cdot \dot{V}_{exhaust}\ (m^3/s) <=>$$

$$\dot{m}_{SOOT}\ (mg/h) = 3600 \cdot SOOT\ (mg/m^3) \cdot \dot{V}_{exhaust}\ (m^3/s) \tag{2.19}$$

With the given density of $SOOT$ in mg/m^3 (from the measurements in Table 1.2), from Table 2.13 and Equation 2.19, the mass flow rate values of the pollutant are obtained.

Finally, using Equation 2.15 and for the known power values, the requested specific emissions are obtained in Table 2.14.

	$\dot{m}_{SOOT}$	$SSOOTE$
-	mg/h	mg/kWh
1	7919.37	315.10
2	47729.51	1139.46
3	6518.20	338.28
4	44971.73	1166.98

Table 2.14: Values of mass flow rate and specific emission $SOOT$ for each operating point

Finally, in Figures 2.5 and 2.6, the specific emissions of pollutants NO_x and $SOOT$ are shown as functions of torque for a constant number of revolutions. It can be observed that for a **constant** number of revolutions N and as torque M **increases**, the specific emission $SOOT$, $SSOOTE$, **increases**. Also, for a **constant** number of revolutions $N = 2000\ rpm$ and as torque M **increases**, the specific emission NO_x, SNO_xE, **increases**, while for a **constant** number of revolutions $N = 2300\ rpm$, it decreases.

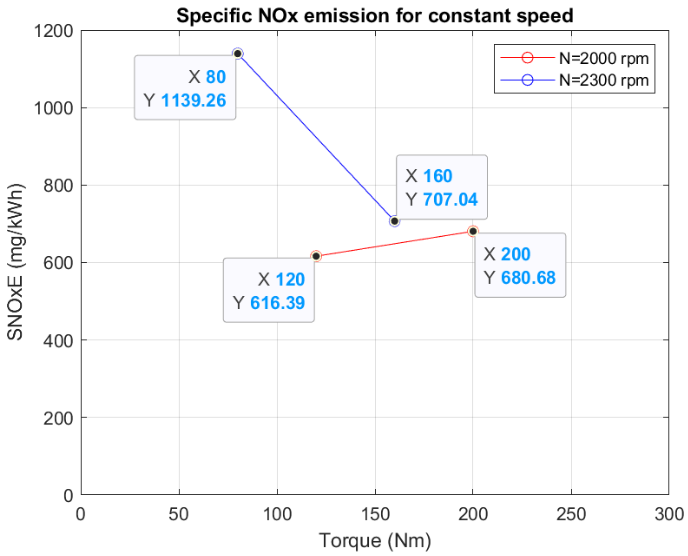


Figure 2.5: Specific emission NO_x for a constant number of revolutions

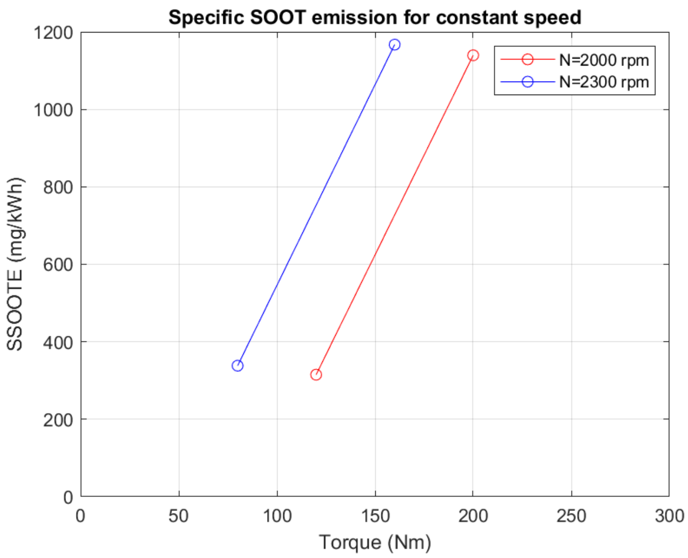


Figure 2.6: Specific emission $SOOT$ for a constant number of revolutions

Chapter 3

Conclusions

Upon completing this study, the following conclusions can be drawn:

1. Observing Figure 2.1, it appears that keeping N constant, higher torques result in lower coolant flow rates. Therefore, for higher engine torques, there is less need for cooling, and consequently, the engine strains the cooling system more for lower torque values M. Additionally, it is also observed that increasing the number of revolutions N decreases the coolant flow rate, indicating less need for cooling at lower RPMs.

2. Observing Figure 2.2, it is evident that keeping N constant, higher torques correspond to higher power values. Thus, for higher engine torques, more energy is added to the fuel ($\dot{Q}_{fuel}$ increases). Furthermore, one can see that power values decrease for lower RPMs. Finally, it is noted that most of the energy is contributed to the fuel by the actual engine power P_e, followed by coolant power $\dot{Q}_{coolant}$, exhaust gas power $\dot{Q}_{exhaust}$, and lastly, the losses $\dot{Q}_A$. It generally holds that:

$$\dot{Q}_{fuel} > P_e > \dot{Q}_{coolant} > \dot{Q}_{exhaust} > \dot{Q}_A$$

3. From Figure 2.3, it is observed that keeping N constant, higher torques result in lower specific fuel consumption values SFC and higher efficiency values η. Therefore, for higher engine torques, less fuel is consumed. The engine works harder at lower torques. Conversely, for efficiency, the engine performs better at higher torques but consumes more fuel. Finally, from equation 2.12, it is understood that the efficiency η and specific fuel consumption SFC are inversely proportional:

$$\eta \propto \frac{1}{SFC}$$

4. In Figure 2.4, it is noticed that keeping N constant, higher torques correspond to lower air-fuel ratio values AFR. This implies that with increasing torque, the air mass flow decreases compared to the fuel mass flow. Therefore, less air is burned at low engine torques. Finally, it is observed that this ratio generally decreases for lower RPMs.

5. In Figure 2.6, it is evident that keeping N constant, higher torques result in higher specific SOOT emission values $SSOOTE$. Thus, at higher engine torques, more SOOT is emitted into the atmosphere. Also, it can be seen that, in general, emissions of soot increase for lower RPMs. From Figure 2.5, no conclusions can be drawn about the variation of specific nitrogen oxide emissions NO_x for a constant number of revolutions. However, it could be speculated that beyond a certain RPM value, SNO_xE increases with increasing engine torque, while below a certain RPM, it decreases. Finally, it is observed that, in general, specific NO_x emissions increase for higher RPMs.

List of Figures

List of Tables

References

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