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Effect of Engine Hardware Parameters on Energy Loss Characteristics of DI Diesel Engine Using Heat Balance Sheet

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Abstract

Improper rejection of heat from exhaust can result in poor thermal efficiency of an engine. Therefore, it is necessary to evaluate the heat losses through heat balance sheet to optimize the efficiency of an engine. This is an experimental approach made to develop a heat balance sheets at varying engine hardware parameters to evaluate the heat transfer characteristics from the exhaust gas. The tests were conducted on a BSII commercial, four cylinders, four strokes, in-line overhead valve,water-cooled direct injection diesel engine.The test conditions were set at OEM setting and the heat balance sheet was prepared. Thereafter the test severity was made to match the real-timesituationsituations by varying hardware parameters like injection pressure (OEM: 240bar) to 220 & 200bar, auxiliary air pressure (0.50-0.55bar), injection timing (29°CA BTDC to 24°CA BTDC) and air throttling (0-50%). The analysis was done at varying hardware parameters individually, and in combinations. The results obtained after comparing the heat balance sheets confirms that as the hardware parameters were reduced (from OEM), it results in an increase in fuel consumption with more heat losses from exhaust gases, thus affecting the thermal efficiency of an engine. The repeatability tests were done with an increase in various parameters (from OEM) and the conclusion is confirmed after every test.

Keywords: Diesel Engine, Energy Analysis, Heat Balance Sheet, Thermal Efficiency.

Introduction

Diesel engines are gaining remarkable position in the field of passenger cars due to its improved thermal efficiencyby way of optimizing its hardware and operating parameters. Improper rejection of heat from exhaust can result in poor thermal efficiency of an engine. By the implementation of first and second laws of thermodynamics, energy analysis of engine through heat balance sheet can be an efficient tool to remark on the thermal efficiency of an engine. The energy analysis allows us develop a systematic approach that can be used to identify sites of real losses of valuable energy in thermal device. Tests performed [1,2] shows that about 40% of fuel chemical energy in the form of heat is wasted due to cooling and heat loss through exhaust gases. Mirko bovo et al. [3] stated the major area of losses and associated heat losses to determine the thermal efficiency of IC engine.

Published literature [4-6] illustrates that effect of parameters like load, turbochargers, dual fuels, pre-post injection strategies directly or indirectly hinder the energy losses in the form of heat through exhaust gases.J Fu et al [7] illustrates the energy analysis of using the bio-gas dual fuel on diesel engine by varying engine load. It was concluded that the presence of CO_2 in the bio-gas reduces the burning velocity and causes the incomplete combustion thereby increasing the exhaust gas temperature in dual fuel mode.

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Also, biogas has longer pilot ignition delay and high self-ignition temperature thus causing the delay in dual fuel combustion process. These factorsdecrease the thermal efficiency of an engine.

M Ozkan et al [8] in his study illustrated the influence of pre-injection timing strategies on thermal efficiency of an engine. It was found that pre-injection decreases the heat release from cylinder block and therefore less amount is necessitated resulting in less cooling loss and more thermal efficiency. Published literature [9] also illustrates the defects which occur in the combustion chamber due to varying operating parameters and their associated heat transfer release rates. It was found that with the change in parameters like injection timing and pressure not only effects the thermal efficiency but also results in thermal cracking or increased heat transfer rates from piston rings and skit. This thereby effects the cylinder wall temperature resulting in more cooling losses in the form of heat thereby affecting thermal efficiency.

As discussed, various engine hardware parameters directly or indirectly effect the thermal efficiency of an engine known through energy analysis. In this work, the modifications in the engine hardware parameters were done to find their effect on thermal efficiency of an engine and performing the energy analysis through heat balance sheet. The test and operating parameters were set to tie the real-time situation, by varying the injection pressure, air throttling, injection timingand auxiliary air supply to FIP. The present study focuses on to optimize the engine hardware parameters capable of delivering high thermal efficiency of an engine. For this heat transfer characteristics for individual changes and their combinations have been tested on a multi cylinder test rig and thermal efficiency of the engine is found.

Experimental Setup

Engine and Fittings

The spcifications of the engine used are enumerated in Table 1 and the schematic experimental setup is shown in Figure 1.

Table 1.Specifications of the engine used in the study.

Engine Type	Diesel, 4strokes, 4 cylinders, direct injection,BSII
Suction type	Turbocharged with intercooler
Bore * Stroke	104×113mm
Piston displacement	3.839L
Max power	88.25kW @ 2500 rpm
Max torque	40 kg-m @ 1500-1600 rpm.
Compression ratio	17.5:1
Injection Pressure	240 bars
Oil sump/system capacity	8 L

AVL Puma 1.5.3 testbed automation system controls the dynamometerand emission measuring equipment which is interfaced with the engine test bed. The test bench is prepared with coolant and oil temperature regulator systems that are supportive of keeping the coolant temperature. The fuel used is the commercial diesel fuel which meets the 50-ppm fuel Sulphur limit of Bharat stage IV Indian emission regulation norm.

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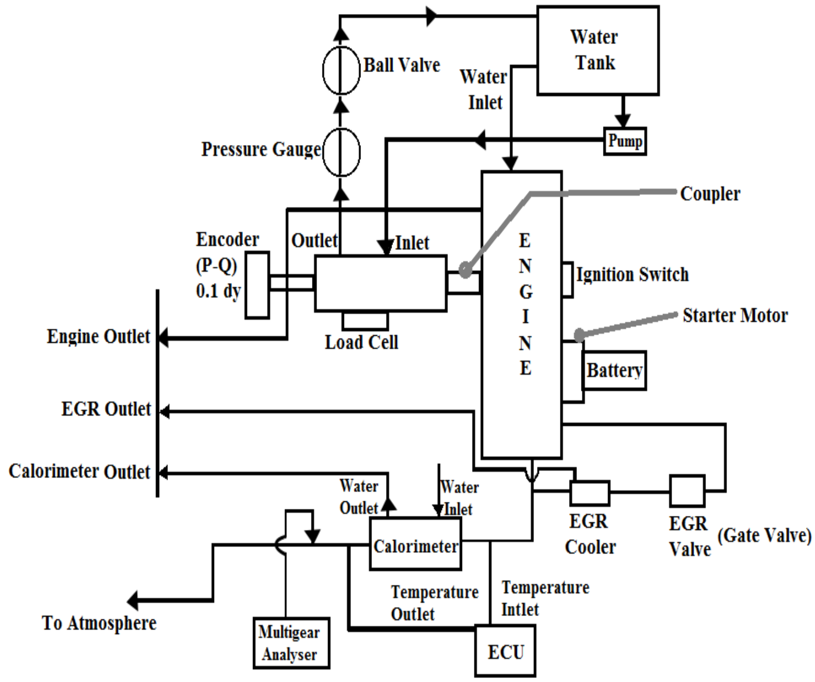

Fig 1. Schematic diagram of full engine setup

Engine Hardware Modifications

a. Change in Intake Air System:

Reducing intake air amount results in fuel rich conditions in the combustion chamber and hence effects the combustion performance. Therefore, a throttle valve (ball valve) was introduced in between the intercooler and the intake manifold as shown in Fig.2, to achieve favorable air restriction. The divisions were marked throughout the travel length of the ball valve (0%, 20%, 50%) based on its position. The values of the air intake flow were w.r.t position of the ball valve (20%, 50% etc.) and not the w.r.t quantity of air flow.

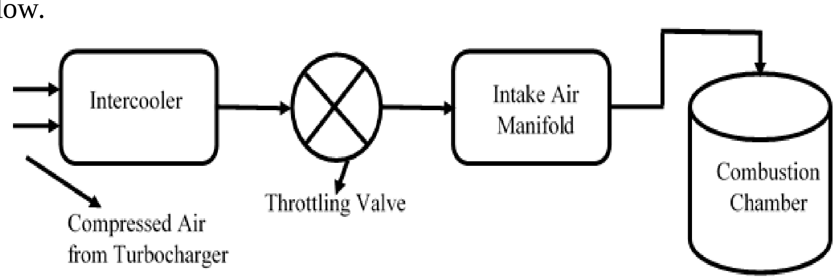

Fig. 2. The modified intake air system used in the study.

b. Auxiliary air supply system:

Fuel Injection Pump (FIP) is linked to the intake manifold of the engine. The valve regulates the amount of fuel supplied to the engine rendering to intake manifold pressure. Accordingly, at low intake pressure, less amount of fuel is injected into the combustion chamber for combustion. In order to eliminate this fluctuating behaviour of the FIP, auxiliary air supply system was engaged. Compressed air was supplied to the diaphragm valve of the FIP as shown in Fig.3, and an appropriate amount of auxiliary air pressure was maintained, which wide opens the diaphragm valve irrespective of any amount of intake pressure in the manifold. The auxiliary air was supplied at 0.50-0.55bar to the FIP diaphragm valve.

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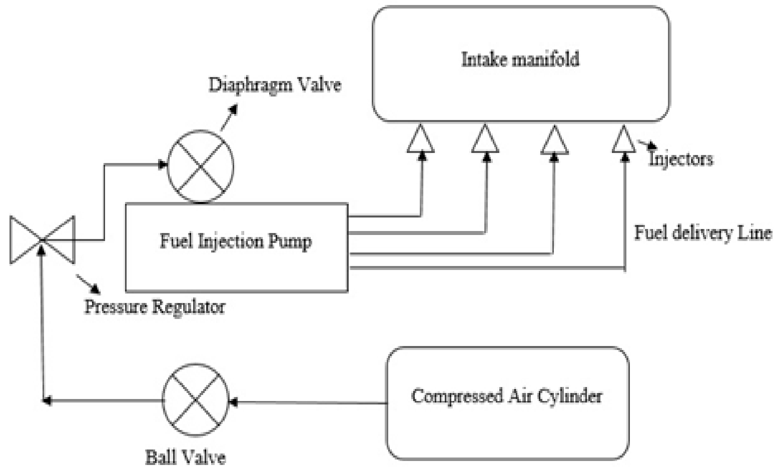

Fig.3. Supplementary air sourceto diaphragm valveFIP.

c. Change of Injector Pressures

Published studies showed that change of injector pressure has a substantial effect on heat transfer characteristics and thermal efficiency of the engine. Injection pressure from 240 bar was alteredto 200 bars and was established for combustion performance. Injection pressure was altered by introducing the shim inside the injector.

d. Change in Injection Timing

Injection timing was changed from 29° CA BTDC to 24° CA BTDC with the help of the provision provided in the FIP.

e. Post injection

After the main injection, a shorter injection was done to the combustion chamber. By this it affected the combustion and emission. The post injection was done 5° BTDC from the initial setting of 1.38 mm plunger lift.

f. Change in air intake temperature:

Air was pre-heated before it mixed with the fuel. As the pre-heated air mixed with fuel the ignition temperature of the air-fuel mixture was increased. It showed the effect in the final result of heat losses. The air is heated up to the temperature of 40-45℃.

Test Conditions

Number of combinations of modified hardware parameters as shown in Table 2were selected to conduct six tests at changed engine speeds reaching from 1100rpm to 1300rpm at fixedload (350Nm).The first test was conducted without any modification of engine hardware and the subsequent tests with a progressive increase in test severity to create a real-time environment. The other operating parameters were maintained as shown in table. 3. Energy analysis of the engine at these conditions were done and the heat balance sheet was prepared after each test. Parameters like Total heat input (Q_T), brake power (bp), indicated power (ip), friction loss (f_L), heat loss through cooling medium (water) (Q_W), heat loss through the exhaust gas (Q_{ex}) and Unaccounted heat loss (q_{unac}) were found for preparing a heat balance sheet. Similar tests were conducted at 1200 and 1300 rpm for commenting on the repeatability of the tests. Mass flow rate of fuel is noted from the AVL mass flow meter integrated with engine setup.

Test 1
During the 1st test run, no modification was incorporated and the engine was made to run at wide open throttle (WOT)and 1100rpm and at 350Nm of load.The reading of exhaust gas temperature, Intake air temperature, coolant water inlet and outlet temperature were noted along with the fuel consumption.

Test 2
Test was conducted after varying the injection pressure and injection timing outlined in table 2. at 1100rpm and at 350Nm of load. Measurement of exhaust gas temperature, coolant water inlet and outlet temperature were done along with the fuel consumption. The intake air temperature was maintained at 25-35**°C.**

Test 3
Test was conducted after varying the settings of injection pressure, injection timing and compressed air as outlined in table 2. The readings of different measuring parameters were noted at 1100 rpm and 350Nm load. No changes were incorporated for air throttling, post injection and air intake temperature.

Test 4
Test was conducted after varying the hardware parameters settings of injection pressure, injection timing, compressed air and air throttling as outlined in table 2. The readings of different measuring parameters were noted at 1100 rpm and 350Nm load.

Test 5
Test was conducted after varying the hardware parameters settings of injection pressure, injection timing, compressed air, air throttling and port injection as outlined in table 2. The readings of different measuring parameters were noted at 1100 rpm and 350Nm load.

Test 6
Test was conducted after varying the hardware parameters settings of injection pressure, injection timing, compressed air, air throttling, post injection and air intake temperature as outlined in table 2. The readings of different measuring parameters were noted at 1100 rpm and 350Nm load.

Test Condition Number	Injection Timing, (°)	Inlet air restriction (%)	Intake Air Temperature (°C)	Post Injection (°)	Injection Pressure (bar)	Auxiliary Air Pressure(bar)	Load (Nm)	Fuel Consumption (Kg/hr.)
1.	OEM settings	WOT	25-35	1.38 mm lift	240	Not Engaged	350	10.50
2.	**Retarded by 5°**	0	25-35	1.38 mm lift	**200**	Not Engaged	350	10.80
3.	**Retarded by 5°**	0	25-35	Nil	**200**	**0.55**	350	10.95
4.	**Retarded by 5°**	**50**	25-35	Nil	**200**	**0.55**	350	11.05
5.	**Retarded by 5°**	**50**	25-35	**5° BTDC**	**200**	**0.55**	350	11.30
6.	**Retarded by 5°**	**50**	**40-45**	**5° BTDC**	**200**	**0.55**	350	11.50

Table. 2. Test Conditions adopted for this study.

Table. 3. Operating parameters maintained during the test done for this study.

S.No.	Operating Parameters	Operating Values
1	Speed, rpm	1100-1300
2	Torque, Nm	350
3	Gallery Oil pressure, bars	3
4.	Water inlet temperature, °C	25

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Results and Discussion

The data obtained were applied and the calculation were done to produce a heat balance sheet for all the test performed at different engine rpm (1100; 1200 and 1300 rpm). Assumptions like constant water inlet temperature at 25-35°C, calorific value of diesel at 44800 kJ/kg K, mass flow rate of water through the water jacket at 0.490625 kg/sec, specific heat of water at constant pressure at 4.187 kJ/kg K, Specific heat of exhaust gas at constant pressure at 1.063 kJ/kg K were taken during this study. The data obtained for all the tests at 1100, 1200 and 1300 rpm and at 350 Nm are shown in table 4, 5 and 6 respectively.

Table. 4. Heat balance sheet produced from the data obtained after the tests at 1100 rpm and 350 Nm torque.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
2.	11.53	143.48	40.68	67.36	26.68	08.2991	12.3238	55.6971
3.	11.69	145.42	41.10	67.78	26.68	09.4905	13.2491	55.5204
4.	11.75	146.17	41.43	68.20	26.77	10.3122	14.0906	54.6072
5.	11.80	146.80	41.89	68.90	27.01	11.1750	15.1174	53.3476
6.	11.90	148.03	42.09	69.50	27.41	12.0542	16.1032	52.7126

Table. 5. Heat balance sheet produced from the data obtained after the tests at 1200 rpm and 350 Nm torque.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
2.	11.55	143.7	44.26	73.86	29.6	12.6130	15.6844	42.1426
3.	11.71	145.67	44.53	74.10	29.57	13.7634	16.5142	42.1324
4.	11.83	147.16	45.08	74.63	29.55	14.7494	17.6256	41.5250
5.	11.91	148.16	45.28	74.86	29.58	15.9819	18.8363	40.0818
6.	11.95	148.65	45.96	75.26	29.30	16.8036	19.9043	38.6821

Table. 6. Heat balance sheet produced from the data obtained after the tests at 1300 rpm and 350 Nm torque.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
2.	11.57	143.93	48.22	80.17	31.95	17.1939	19.0204	28.3457
3.	11.74	146.30	48..89	80.98	32.09	18.2005	19.8221	28.9074
4.	11.85	147.41	49.50	81.55	32.05	19.2482	20.7877	28.0041
5.	11.95	148.65	50.10	82.04	31.94	20.3369	21.5673	27.3758
6.	12.00	149.28	50.68	82.54	31.86	21.6516	22.8004	25.4280

It could be noted from the heat balanced sheets, the consumption of fuel is increasing from first combination to fifth combination at all speeds. Therefore, the total heat input is also increasing with increasing engine test severity. The losses through the cooling water, exhaust gas and the unaccounted loss are increasing as the combinations of modification are added one by one from (Injection Pressure + Injection Timing) to (Injection Pressure + Injection Timing + Compressed air + Air throttling + Post injection + Air intake temperature). It is therefore possible now to compare the test results without any modifications and with test conditions produced for test severity as shown in table 7.

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Table. 7. A comparison of data obtained for test condition 2 at different rpm with that of test condition 1.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
2. @ 1100 rpm	11.53	143.48	40.68	67.36	26.68	08.2991	12.3238	55.6971
2. @ 1200 rpm	11.55	143.7	44.26	73.86	29.6	12.6130	15.6844	42.1426
2. @ 1300 rpm	11.57	143.93	48.22	80.17	31.95	17.1939	19.0204	28.3457

From the table.7 it is seen that due to change in injection pressure to 200bar from 240bar and the change in injection timing 29°CA BTDC to 24°CA BTDC the fuel consumption rate increased about 1.03 kg/hr, thereby resulting in rise in average temperature of the combustion process. Hence the total heat input also increased about 12.86kW. The possible reason might be reason can be due to decrease in fineness of atomization of fuel, giving larger fuel droplet size and ignition delay thereby increasing fuel consumption. Similar rise in fuel consumption and heat release was observed at 1200 rpm and 1300 rpm tests.
Table 8. depicts the comparison between the heat values and the fuel consumption for test condition number 3 at all speeds. It is observed that due to change in injection timing and pressure along with auxiliary air pressure there is a rise in fuel consumption and total heat released. This is possible because auxiliary air pressure opens the diaphragm valves of FIP, hence delivering constant flow rate of fuel inside the engine irrespective of change of intake pressure in intake manifold caused by inlet air restriction.

Table. 8. A comparison of data obtained for test condition 2 at different rpm with that of test condition 1.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
3.@1100 rpm	11.69	145.42	41.10	67.78	26.68	09.4905	13.2491	55.5204
3. @1200 rpm	11.71	145.67	44.53	74.10	29.57	13.7634	16.5142	42.1324
3. @1300 rpm	11.74	146.30	48..89	80.98	32.09	18.2005	19.8221	28.9074

Table. 9. depicts the comparison between the heat values and the fuel consumption for test condition number 4 at all speeds. It is observed that with the addition of inlet air restriction to the above settings there is a rise in fuel consumption and total heat released. This might be due to increased air throttling from 0% (during test 1) to 50% closure of valve in test 4.

Table. 9. A comparison of data obtained for test condition 2 at different rpm with that of test condition 1.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
4. @1100 rpm	11.75	146.17	41.43	68.20	26.77	10.3122	14.0906	54.6072
4. @1200 rpm	11.83	147.16	45.08	74.63	29.55	14.7494	17.6256	41.5250
4. @1300 rpm	11.85	147.41	49.50	81.55	32.05	19.2482	20.7877	28.0041

Table 10. depicts the comparison between the heat values and the fuel consumption for test condition number 5 at all speeds. It is observed that with the addition of post injection extra quantity of fuel was injected inside the combustion chamber when compared with that of test 1. This resulted in proper

combustion and rise in combustion temperature. Similar nature was observed for test 6 which is shown in table 11.

Table. 10. A comparison of data obtained for test condition 2 at different rpm with that of test condition 1.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
5. @1100 rpm	11.80	146.80	41.89	68.90	27.01	11.1750	15.1174	53.3476
5. @1200 rpm	11.91	148.16	45.28	74.86	29.58	15.9819	18.8363	40.0818
5. @1300 rpm	11.95	148.65	50.10	82.04	31.94	20.3369	21.5673	27.3758

Table. 11. A comparison of data obtained for test condition 2 at different rpm with that of test condition 1.

Test Condition Number	Fuel consumption (kg/hr)	Q_T (kW)	bp (kW)	ip (kW)	f_L (kW)	Q_W (kW)	Q_{ex} (kW)	Q_{unac} (kW)
1.	10.50	130.62	40.29	67.16	26.87	06.7789	07.6007	49.0804
6. @1100 rpm	11.90	148.03	42.09	69.50	27.41	12.0542	16.1032	52.7126
6. @1200 rpm	11.95	148.65	45.96	75.26	29.30	16.8036	19.9043	38.6821
6. @1300 rpm	12.00	149.28	50.68	82.54	31.86	21.6516	22.8004	25.4280

Conclusion

An experimental work was carried out in order to understand the effect of various hardware parameters on energy loss pattern of an engine. Based on the extensive studies following conclusions can be supported.

1. The test was able to outline the effect of various engine hardware operating parameters on the heat balance sheet of the engine.
2. Engine workingconstraints such as intake air restriction, injection pressure, injection timing and supplementary air pressure have significant influence on total heat input and fuel consumption characteristics of an engine.
3. It is found that most protuberantconstraint which is distressing the total heat input and fuel consumption is injection timing followed by intake air restriction.
4. Comparable test results were obtained for the tests conducted at 1200 rpm and 1300 rpm with that of 1100 rpm.

   It is therefore concluded that, with the change in various hardware parameters there is an increase in thermal efficiency and fuel consumption. The possibilities of exergy analysis on factors considered in this work could be a future scope of this work for finding the optimum levels of engine hardware parameter change.

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