Experimental Studies on Energy Appropriation in a Single Cylinder Diesel Engine

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The energy produced due to combustion of fuel in an engine is partly converted into work and the rest is lost. The knowledge of how the energy is lost will help in finding means to reduce the same to improve the performance of the engine in terms of efficiency and power output. In the experimental work, reported in this paper, a single cylinder, naturally aspirated. 4-stroke diesel engine fully instrumented for the measurement of engine output, speed, fuel consumption, air flow rate and temperature of the cylinder liner and exhaust gases and lubricating oil, was used. Estimation of energy lost due to heat transfer to engine components was arrived at.

Keywords : Energy loss; Heat transfer through liner; Energy balance

NOTATION

k	: thermal conductivity, W/m K
l	: finite length of the cylinder, m
Q	: rate of heat transfer, W
r_i, r_o	: inner and outer radii, respectively, m
T	: temperature at corresponding position, °C

Subscripts

exh	: exhaust gases
1, 2, 3, 4, 11, 21, 31, 41	: respective thermocouple positions as shown in Figure 2

INTRODUCTION

Diesel engines are being extensively used for rail and road transportation, agriculture applications and power generation. Increasing demand and depleting fossil fuels have lead to research and development on production of energy efficient engines. Minimising the energy losses in the engine definitely improves the power output and efficiency of the engine.

In order to improve that performance of the engine, many experimental and theoretical investigations have been carried out elucidating the heat transfer characteristics of compression ignition engines.

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This paper was received on April 3, 2003. Written discussion on the paper will be entertained till September 30, 2004.

Saugerud and Sandmark[1] discussed a simplified two-dimensional model, used in the analysis of thermally loaded components such as engine piston and radial turbine. They concluded that it is sufficient to use a two-dimensional model compared to three-dimensional model in the estimation of thermal loading of engine components.

Shilling and Woschni[2] measured instantaneous surface temperature variations at five positions of the combustion chamber side of the cylinder head of a modern 4-valve high-speed diesel engine using an external super charger. They compared the measured local-mean instantaneous heat transfer coefficients with the calculated values. They found the measured values to be good in agreement with the calculated values.

Woschni[3] measured experimentally steady state temperature fields in the piston for different operating conditions of a high-speed diesel engine. They also evaluated them using the relaxation method and the electrolytic tank analogue. They determined the local heat transfer coefficients at the contour of the piston. These measured values were found to be in good agreement with the predicted values.

Kamel and Watson[4] developed a numerical model to predict the heat transfer coefficient values for both motored and fired operations. They also experimentally measured the instantaneous heat flux in pre and main combustion chamber under a wide range of operating conditions, for both motored and fired engines. They found that the measured and predicted heat fluxes agree well for both motored and fired conditions.

Alkidas[5] conducted experimental studies on the unsteady state heat flux characteristics of a single cylinder spark ignition engine. He measured the transient heat flux at four positions on the cylinder head and the gas pressure. The volumetric efficiency and heat flux values were compared both at motored and fired condition.

Caton and Haywood[6] developed a model for the calculation of instantaneous heat transfer in engine exhaust port and then compared these with the experimental results.

Instantaneous cylinder pressure and exhaust gas temperatures were measured for a wide range of engine operating conditions. The pressure measurements were used to obtain the instantaneous cylinder gas state and the temperature measurements were used to validate the heat transfer models.

Alkidas and Myers[7] in their experimental studies obtained heat-flux measurements at several locations on the cylinder head and liner for a four stroke, single-cylinder SI engine. They also investigated the variations of heat transfer with air-fuel ratio and volumetric efficiency. The calculated amount of heat transferred to the walls of the combustion chamber during closed portion of the engine cycle (intake valve closing to exhaust valve opening) agreed with the corresponding values obtained from the heat-flux measurements.

Magnus[8] investigated the influence of thermal barrier on engine performance and the resultant heat transfer under a wide range of operating conditions. The data showed how different surface configurations influence the in-cylinder process, fuel consumption, heat loss and exhaust energy. Steady state temperature fields on the engine components were plotted and it was concluded that these were functions of coating thickness, location and condition.

In these works attempts were made to find the heat transfer co-efficient and temperature distribution in the engine cylinder both experimentally and theoretically. In the present work, componential heat transfer studies were made and energy balance was obtained. These componential heat transfer studies aid in improving the performance of the engines.

The present work was undertaken to critically analyse the heat loss through various components and precisely obtain the energy balance and also to obtain the temperature distribution in the cylinder liner.

EXPERIMENTAL SET-UP

The experimental set-up used in the present investigation is shown in Figure 1.

Experiments were performed on a single cylinder, 4-stroke naturally aspirated, water-cooled, constant speed, direct ignition CI engine of 80 mm bore and 110 mm stroke. The rated power and rated speed were 3.68 kW and 1500 rpm,

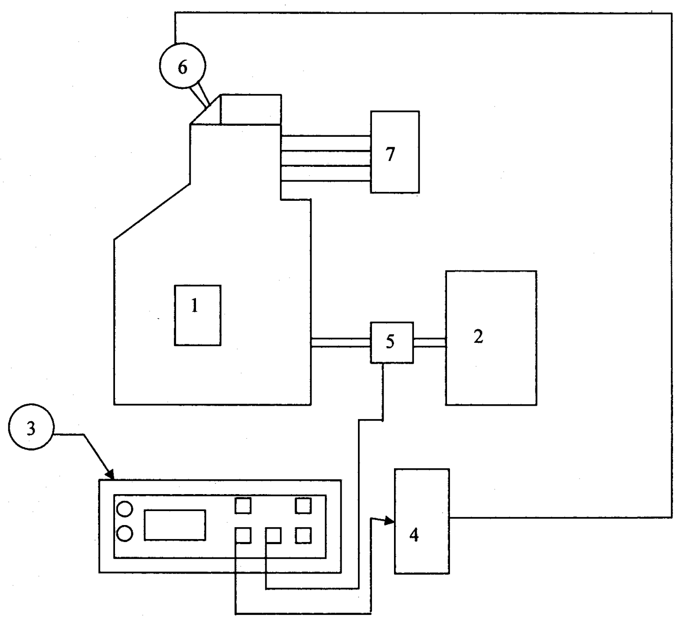

Figure 1 The experimental set-up

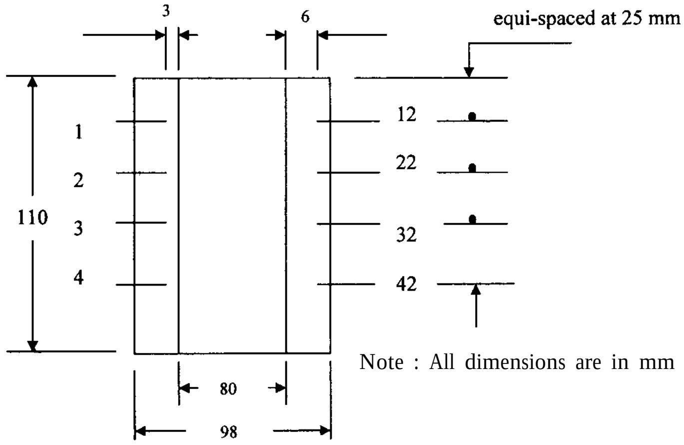

Figure 2 Position of thermocouples

respectively. The air supply to the engine was measured using a turbine flow meter in the intake pipe. A hydraulic dynamometer was used for loading the engine. The jacket water flow through the engine was also measured.

Chromel-alumel thermocouples were placed at four axial locations in the liner and at each axial location, at two radial depths, of 3 mm and 6 mm from the inner surface of the liner as shown in Figure 2.

INSTRUMENTATION

For obtaining the energy balance, the measurements made were :

Load measurement	:	Hydraulically loaded dynamometer 1 was used to load the engine.

Speed measurement	:	Engine speed was measured with the help of a tachometer.
Airflow measurement	:	The atmospheric air that was sucked into the engine cylinder was measured with the help of a turbine type flow meter.
Fuel measurements	:	The diesel flow is measured using a burette and stopwatch.
Temperature measurements	:	To calculate the heat transfer through the liner, the liner temperatures were measured at different axial location as shown in Figure 2. For making energy balance, the inlet air temperature and the exhaust gas temperature at the exit were measured. All the temperatures were collected using a data acquisition system.

EXPERIMENTAL PROCEDURE

The engine was started and allowed to run for about 25 min to reach steady state conditions. After attaining steady state conditions the fuel consumption, temperature at corresponding liner positions and the exhaust gas temperature were noted for no-load condition. After this, the engine was loaded in steps and corresponding data (fuel consumption, temperatures) for each load was noted. All the observations are tabulated as shown in Table 1. Using storage oscilloscope P-θ diagram was obtained. The area under the P-θ diagram gave indicate power produced per cycle. Finally, the friction power was calculated by substacting brake power from indicate power.

DATA REDUCTION

The temperature of the liner at various axial location, for different load conditions is shown in Table 1. The steady state heat transfer is calculated using the relation:

$$Q = 2\pi k 1 \frac{(t_o - t_i)}{\ln[r_o / r_i]}$$

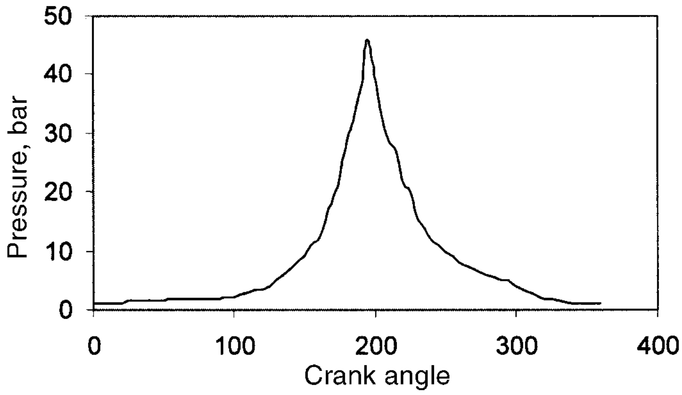

Figure 3 P-θ diagram

Having estimated the rate of heat transfer through the liner, the inner and outer surface temperatures of the liner are obtained by extrapolation.

Constant speed performance test is conducted at different loads as shown in Table 1. At each of these loads, energy balance is made. In the energy balance heat converted into work, heat transferred to engine components, heat carried away by exhaust gases and energy lost due to friction are calculated.

Based on the actual air-fuel ratio, the composition of the exhaust gases is determined assuming complete combustion giving rise to the various constituents in the exhaust gases namely, CO_2, H_2O, N_2, O_2. Knowing the exhaust gas temperature, the specific heats of individual constituents are obtained. Assuming ideal gas behaviour for the constituents, the specific heat of the exhaust gases is obtained from principles of mixtures of gases.

The variation in the cylinder pressure with crank angle is obtained using a storage oscilloscope. The energy produced in the engine is calculated from the P-θ diagram as shown in Figure 3. The difference in the energy produced and measured output of the engine gives frictional power of the engine.

Energy input due to combustion of fuel=brake power+heat transferred to liner+energy lost due to friction+heat carried away by the exhaust gases + miscellaneous (radiation, heat lost to cylinder head, piston).

Table 1 Experimental data obtained during the constant speed performance test at 1500 rpm

Load, kg	T_1	T_2	T_3	T_4	T_{12}	T_{22}	T_{32}	T_{42}	T_{exh}	Time for 10 cc of oil consumed, s	Time for 1m^3 of air consumed s
0	91.0	86.5	81.0	76.0	87.0	85.5	80.0	76.0	219.5	78	150
0.8	98.0	92.5	86.0	79.5	92.5	89.0	83.0	77.5	309.5	49	144
1.6	106.5	99.5	92.0	83.5	99.0	94.5	87.5	79.5	408.0	40	163
2.4	110.5	102.5	94.5	85.5	102.5	97.0	90.0	81.5	443.5	35	170
3.3	132.5	122.5	111.5	99.5	112.0	109.0	99.5	91.0	710.5	20	172

RESULTS AND DISCUSSIONS

Table 2 shows the energy balance at various loads. It can be observed from the table that the percentage of heat lost to the liner increases with load. The heat transfer to the liner is both due to convection and radiation from the hot gases. The heat transfer to the liner increases with load due to increase in heat transfer coefficient and also due to increase in the temperature of the products of combustion.

Figures 4(a), 4(b) and 4(c) show the temperature distribution in the liner at various axial locations. It is seen from the figure that the liner temperature at any radial position gradually decreases from the top to the bottom. Similarly, it can also be observed that the temperature gradient gradually decreases from the top to the bottom of the liner. The same trend is observed at other loads also. This is because as the expansion takes place the temperature of the products of combustion decreases.

Table 2 Energy balance at various loads

	no load	¼ load	½ load	¾ load	full load
Energy supplied, kW	4.65	7.41	9.07	10.37	18.14
Brake power, kW	0	0.88	1.76	2.64	3.63
Heat lost to liner, kW	0.65	1.58	2.15	2.38	5.88
Heat lost to exhaust gases, kW	1.62	2.60	3.21	3.47	6.41
Friction power, kW	1.38	1.38	1.38	1.38	1.38
Miscellaneous losses, kW	0.93	0.94	0.53	0.47	0.81

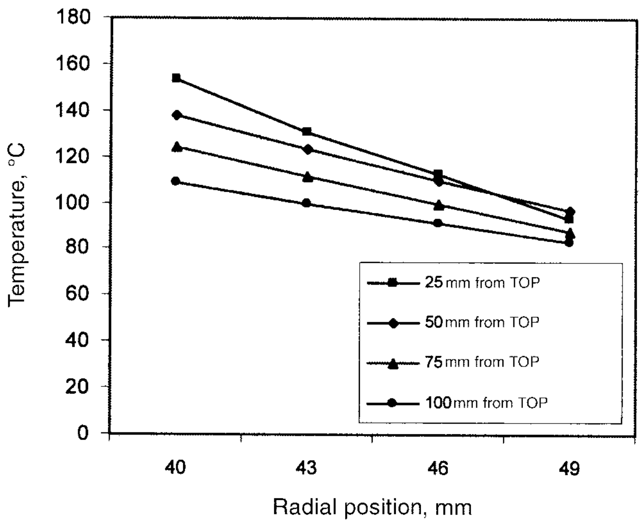

Figure 4(a) Radial temperature distribution at full load

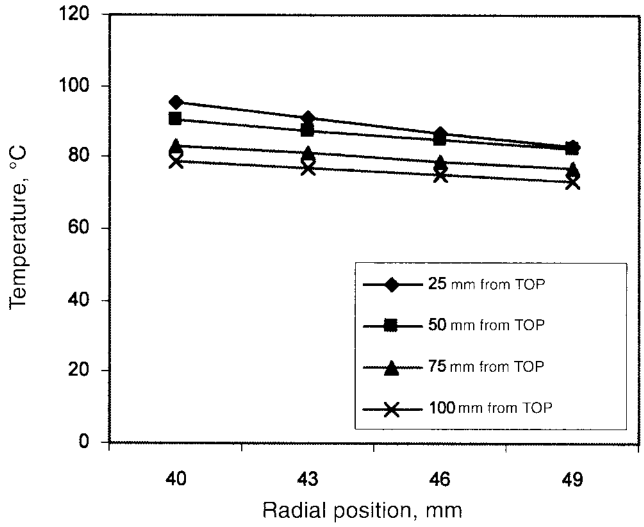

Figure 4(b) Radial temperature distribution at no load

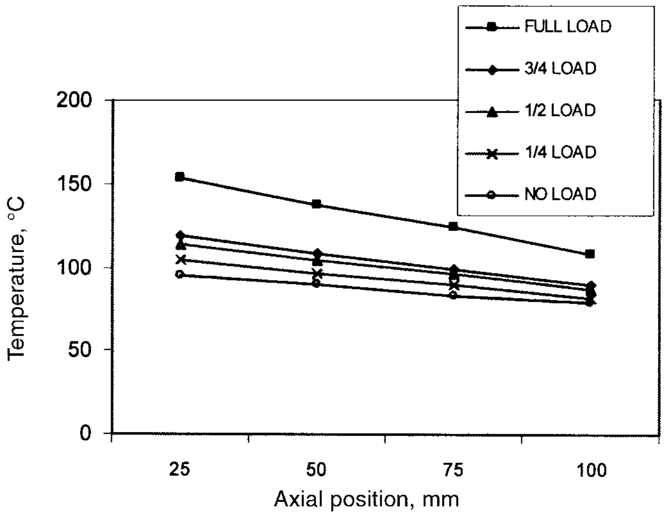

Figure 4(c) Temperature distribution along innerside of cylinder liner

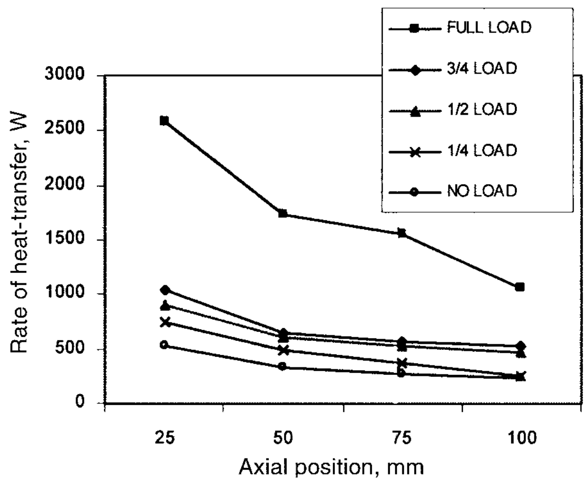

Figure 5 Variation of heat transfer along the cylinder liner

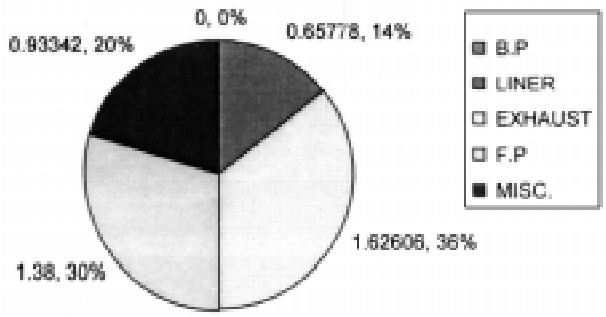

Figure 6(a) Pie diagram showing heat balance at no load

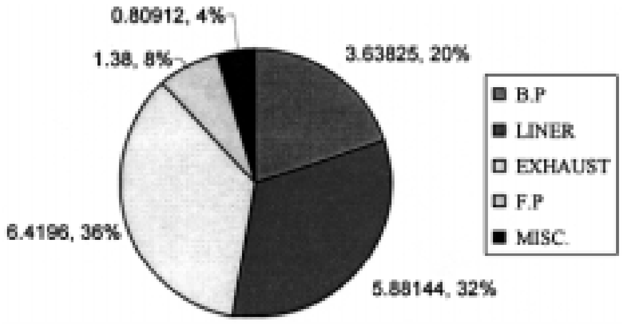

Figure 6(b) Pie diagram showing heat balance at full load

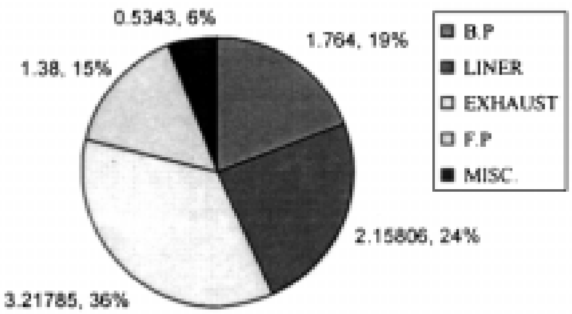

Figure 6(c) Pie diagram showing heat balance at 1/2 load

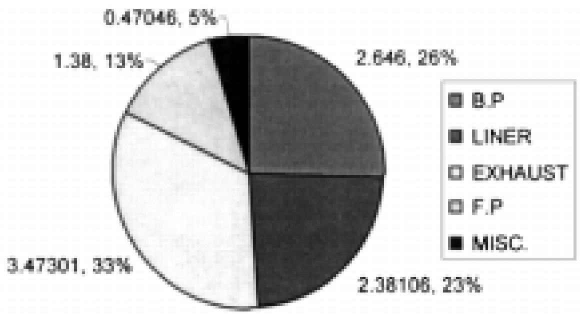

Figure 6(d) Pie diagram showing heat balance at 3/4 load

Figure 5 shows the variation in heat transfer along the cylinder liner at various axial locations for various loads. The heat transfer has decreased from top to bottom of liner as the temperature gradient decreased.

Figures 6(a)–(d) are pie diagrams drawn to show how the energy is accounted for different loads in the engine. From these pie diagrams it is observed that heat lost through liner increases as load increases.

CONCLUSIONS

Precise energy balance is obtained by considering heat lost through liner rather than considering heat lost to coolant water.

It can also be concluded that the heat transfer through liner has shown a dropping tendency from the top to the bottom of the liner.

The heat transfer through the liner has increased dramatically at full load as shown in Figure 5. The heat transfer gradient for the first two axial positions (*ie*, at 25 mm and 50 mm) is high when compared with the heat transfer gradient at other positions.

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