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Thermal balance of a single cylinder diesel engine operating on alternative fuels

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Abstract

The thermal balance of a constant speed stationary compression ignition engine operating on diesel, ethanol–diesel blends and fumigated ethanol was established at different loading conditions of the engine. The thermal balance was in respect of useful work, heat lost to cooling water, heat lost through exhaust, heat carried away by the lubricating oil and other losses (unaccounted-for losses). The results indicate that the thermal balance of the engine operating on 5 and 10% ethanol–diesel blends and fumigated ethanol was not significantly different at the 5% level of significance when compared to diesel. However, in the case of 15 and 20% ethanol–diesel blends, the thermal balance was significantly different compared to diesel. © 2000 Published by Elsevier Science Ltd. All rights reserved.

Keywords: Thermal balance; Alternative fuels; Ethanol–diesel blends

1. Introduction

Alcohols have continued to receive worldwide attention as alternative fuels in spite of surpluses in crude oil. Methanol has been targeted as the fuel of the future on the basis of its low cost of production, while support for the use of ethanol has increased in recent years, in the wake of anti-pollution regulations, owing to its anti-knock properties and higher miscibility

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with gasoline as compared to methanol. In the long term, as the world's crude oil supplies cease to meet global consumption, it is likely that engines running on pure alcohol will become more viable.

In the short term, particularly in those countries vulnerable to a shortage in crude oil supplies, contingency plans in the form of alternative liquid fuels to meet the needs of their transport and agricultural sectors are necessary. The extension of diesel fuel supplies is therefore, of particular concern.

The use of ethanol in compression-ignition engines has, therefore, received considerable attention with particular emphasis on adapting the fuel to meet the requirements of the engine. The preliminary steps of measuring engine performance and conducting limited durability tests have been performed by a number of researchers [1–4]. Hansen et al. [4] investigated the combustion of ethanol and blends of ethanol with diesel fuel with the aid of a heat release model. They observed that the effects of adding ethanol to diesel fuel were increased ignition delay, increased rates of premixed combustion, increased thermal efficiency and reduced exhaust smoke. Czerwinski [5] used a rapeseed oil, ethanol and diesel fuel blend and compared the heat release curves with diesel fuel. He observed that the addition of ethanol caused longer ignition lag at all operating conditions. At higher and full loads, the combustion speeds were high with strong premixed phases.

Ali et al. [6] operated a Cummins N 14-410 engine on 12 fuels produced by blending methyl tallowate, methyl soyate and fuel ethanol with diesel fuel. The addition of ethanol to the fuel blends did not affect ignition delay. The charge temperature was reported to decrease with a decrease in the diesel content of the fuel blends.

It is apparent that very little information is available on the thermal balance of medium size compression ignition engines operating on alternative fuels. The objective of the study reported in this paper was to establish the thermal balance of a constant speed medium size compression ignition engine operating on ethanol–diesel blends and fumigated ethanol as fuels.

2. Materials and methods

2.1. Engine and instrumentation

A stationary, constant speed, single cylinder, 10 Bhp diesel engine was used for the study. Specifications of the engine are presented in Table 1.

The engine was coupled to an Al-Tech. make, BK type hydraulic dynamometer. The engine temperature at various points, the inlet and outlet water temperatures as well as lubricating oil temperatures were measured using a temperature measuring device which consisted of a board on which five digital temperature indicators were fitted. Each indicator had a four way switch and thermocouples were connected to the switches. The device for temperature measurements is shown schematically in Fig. 1. The types of thermocouples used and the points of use are presented in Table 2.

Table 1
Tested engine specifications

Make	Kirloskar
Model	TV 110
Horsepower (rated)	10 hp (7.4 KW)
Rated speed	1500 rpm
No. of cylinder	1
Bore × stroke (mm)	110 × 116
Displacement volume	1102 cm^3
Compression ratio	15.6:1
Cooling system	Water cooled
Lubrication system	Force feed

2.2. Testing procedure

The engine was operated with ethanol–diesel blends having 5, 10, 15 and 20% ethanol on volume basis as well as on fumigated ethanol–diesel fuel. For the fumigation operation, the ethanol from a separate tank was supplied to the engine through a variable jet carburetor. Tests on diesel fuel alone were also conducted as a basis for comparison. The engine was run on the no load condition and its speed was adjusted to 1500 + 20 rpm by adjusting the screw provided with the fuel injector pump rack. The engine was run to attain uniform speed, then it

Table 2
Types of thermocouples and their point of use

S.No	Designation	Type	Point of use
1	T_{11}	Cu–Cons[a]	Inlet water
2	T_{12}	Cu–Cons	Outlet water
3	T_{21}	Cr–Al[b]	Cylinder block (Cranking side)
4	T_{22}	Cr–Al	Cylinder block (inlet water side)
5	T_{23}	Cr–Al	Cylinder block (flywheel side)
6	T_{24}	Cr–Al	Cylinder block (exhaust side)
7	T_{31}	Cr–Al	Cylinder head (exhaust port)
8	T_{32}	Cr–Al	Cylinder head
9	T_{33}	Cr–Al	Cylinder head (inlet port)
10	T_{34}	Cr–Al	Cylinder head
11	T_{41}	Cr–Al	Crankcase
12	T_{42}	Cr–Al	Crankcase (oil sump)
13	T_{43}	Cr–Al	Crankcase (fly wheel side)
14	T_{44}	Cr–Al	Crankcase
15	T_{51}	Cr–Al	Exhaust gases
16	T_{52}	Cu–Cons	Lubricating oil
17	T_{53}	Cu–Cons	Ethanol inlet
18	T_{54}	Cu–Cons	Ethanol outlet

[a] Cu–Cons: Copper–Constantan (type T).
[b] Cr–Al: Cromel–Alumel (type K).

was gradually loaded. The experiments were conducted at five load levels, viz., no load, 25, 50 and 75% of full load and full load. For each load condition, the engine was run for at least three minutes, and the temperatures for the various points were recorded. The experiments were replicated three times.

The heat losses through the various points were calculated as follows:The total heat (Q)

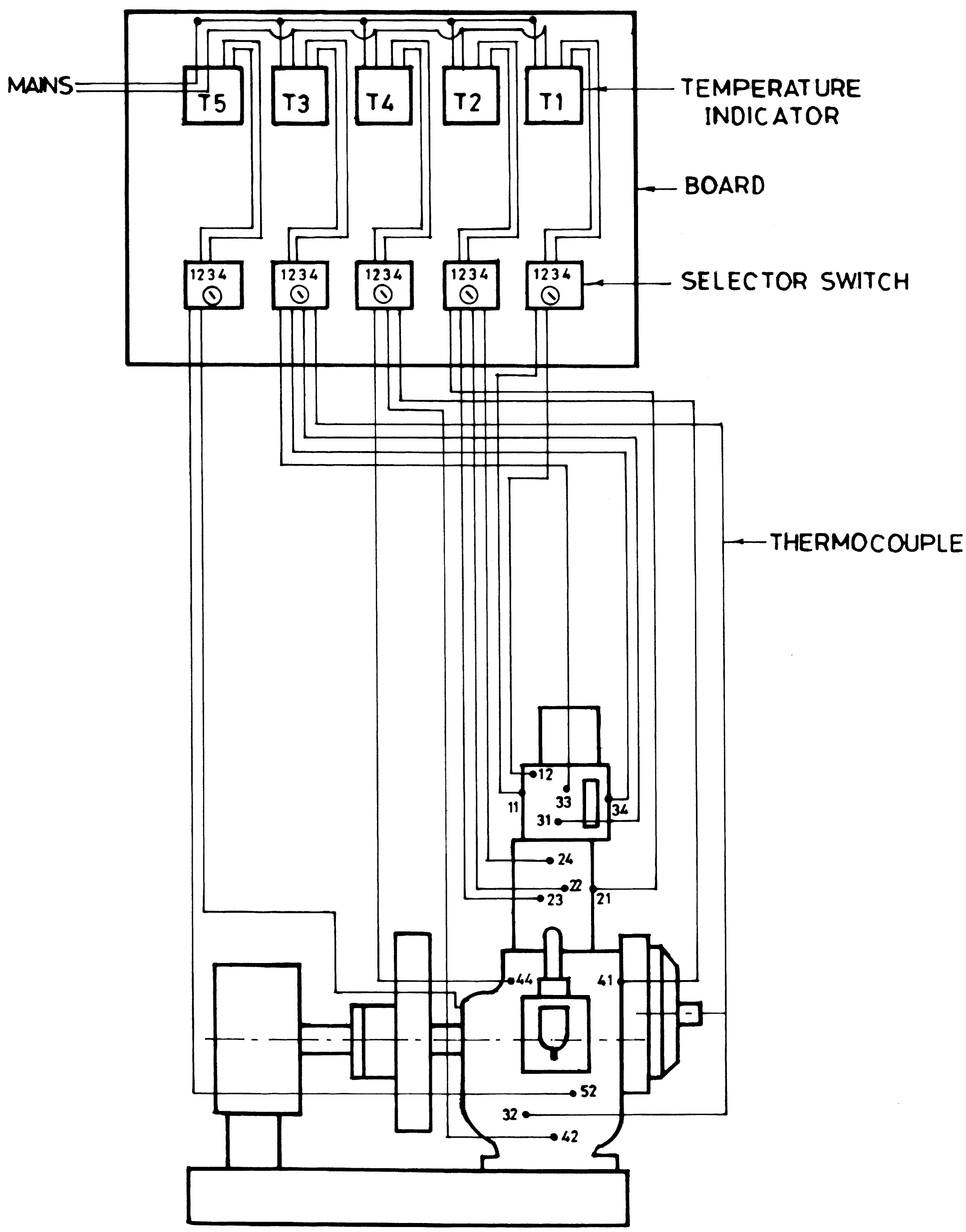

Fig. 1. Thermocouple connections for temperature measurements at various points of the engine.

supplied by the fuel is given as

$$Q = \frac{CV \times \dot{M}_f}{3600} \tag{1}$$

where $\dot{M}_f$ is the fuel consumption (kg/h) and CV is the calorific value of fuel (kJ/kg).

The percentage of heat supplied per second which is converted to useful work (q_1) is

$$q_1 = \frac{\text{Bhp} \times h_e}{Q} \times 100 \tag{2}$$

where Bhp is the brake horse power, h_e is the heat equivalence of Bhp (0.7344) and Q is the total heat supplied by fuel (kJ/s)

The percentage of heat taken from the engine by the cooling water (q_2) was determined by measuring the flow rate of water ($\dot{M}_w$) entering the engine as well as the temperature difference of inlet and outlet water.

$$q_2 = \frac{\dot{M}_w}{3.6} \times C_w \times (T_2 - T_1) = \frac{4.19}{3.6}\dot{M}_w(T_2 - T_1) \tag{3}$$

where C_w is the specific heat of water (kJ/kg °C), T_1 is the inlet water temperature (°C) and T_2 is the outlet water temperature (°C).

The percentage of heat lost through the exhaust gases (q_3) was calculated considering the heat necessary to increase the temperature of the total mass (air + ethanol + diesel fuel) $\dot{M}_g$ (kg/h) from outside conditions T_a (°C) to the temperature of the exhaust T_g (°C). This heat loss is also known as 'sensible heat', and to calculate it, it is necessary to estimate the mean specific heat (C_g) of the gases which, in this case, was assumed to be the value for air with a mean temperature of the exhaust.

$$q_3 = \frac{\dot{M}_g}{3600} \times C_g \times (T_g - T_a) + \frac{\dot{M}_g}{3512}(T_g - T_a) \tag{4}$$

The percentage of heat taken away by the lubricating oil (q_4) is calculated as

$$q_4 = \dot{M} C_{oil}\, \Delta T \tag{5}$$

where $\dot{M}$ is the mass flow rate of oil (kg/s) which is equal to (volume of oil × density of oil)/60, C_{oil} is the specific heat of oil (kJ/kg °C) and ΔT is the temperature rise in oil (°C).

The unaccounted percentage of heat losses (q_5) is given as

$$q_5 = 100 - (q_1 + q_2 + q_3 + q_4) \tag{6}$$

3. Results and discussion

The thermal balance of the engine operating on diesel, ethanol–diesel blends, and fumigated ethanol was established at different loading conditions of the engine. the thermal balance was in respect of useful work, heat lost to cooling water, heat lost through the exhaust, heat carried away by the lubricating oil and other losses (i.e., radiation, vapour in the exhaust, unaccounted for losses). The relationships between engine thermal balance and percentage load for the various fuels used are presented in Figs. 2 and 3. It can be seen from the figures that as the load on the engine increased, the percentage of useful work increased, while the other losses decreased. At the initial stage, the increase was more pronounced than at the latter stages of the loading conditions. This trend is due to the fact that the engine attains optimum operation at the latter stages of loading conditions, and as such, the differences in useful work is minimal. The engine thermal balance at maximum load is presented in Table 3. The table shows that the quantum of useful work for diesel was 28.68% whereas it was 28.73, 31.06, 31.95 and 32.89% for 5, 10, 15 and 20% ethanol–diesel blends, respectively. As the percentage of ethanol in the ethanol–diesel blends increased, there was an increase in the quantum of useful work done by the engine as compared to diesel fuel operation. This is because of the cooling effect of ethanol as well as more efficient combustion as compared to diesel. Since both the exhaust gas temperature as well as the lubricating oil temperatures, were lower in the case of ethanol–diesel blend operations, there was less heat loss through these channels, and as such, more useful work was available at the engine crankshaft. Other losses (i.e. cooling water, exhaust gas, lubricating oil and others) are also presented in Table 3.

In the case of fumigated ethanol, 28.52% of the heat input was utilized as useful work in cold fumigation whereas 28.14% was utilized in the case of preheated fumigation; thereby, resulting in higher heat input for the former than the latter. This finding is consistent with those earlier reported [6,7].

The analysis of variance for the thermal balance indicates that the thermal balance of the engine operating on diesel, 5 and 10% ethanol–diesel blends and fumigated ethanol is not significantly different at the 5% level of significance. However, the thermal balance of the engine operating on 15 and 20% ethanol–diesel blends was significantly different compared to diesel at the 5% level of significance.

Table 3
Thermal balance of engine at maximum load[a]

	Diesel	5% blend	10% blend	15% blend	20% blend	Carburetion (unheated)	Carburetion (heated)
Useful work	26162.8	28195.2	28083.6	26586.0	25894.8	28562.4	27939.6
q_1	28.68	28.73	31.06	31.95	32.89	28.52	28.14
Cooling water	17395.2	17359.2	15951.6	14385.6	14299.3	16444.8	18298.8
q_2	17.72	17.69	17.64	17.29	17.13	16.42	18.43
Exhaust	18248.4	16250.4	14558.4	13219.2	12963.6	23655.6	17168.4
q_3	18.59	16.56	16.10	15.89	15.53	23.62	17.29
Lubricating oil	18453.6	14713.2	11905.2	10148.4	10108.8	19479.6	19468.8
q_4	18.80	14.99	13.16	12.20	12.11	19.45	19.61
Other losses	15552.0	21618.0	19933.2	18856.8	20217.6	12009.6	16412.4
q_5	16.21	22.03	22.04	22.67	22.34	11.99	16.53

[a] Higher values are in kJ/h while lower values are percentages.

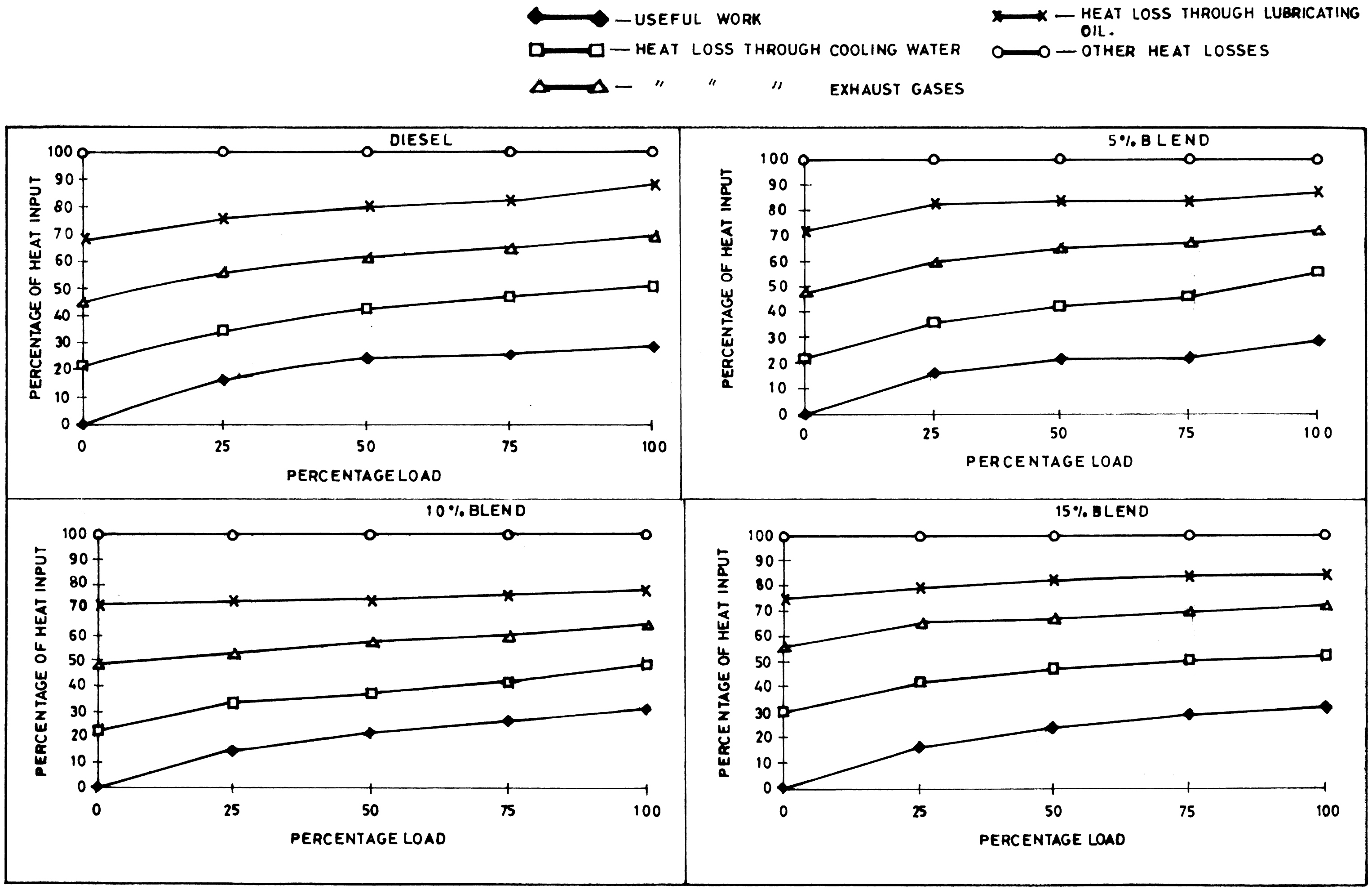

Fig. 2. Thermal balance of engine operating on diesel, 5, 10 and 15% ethanol–diesel blend.

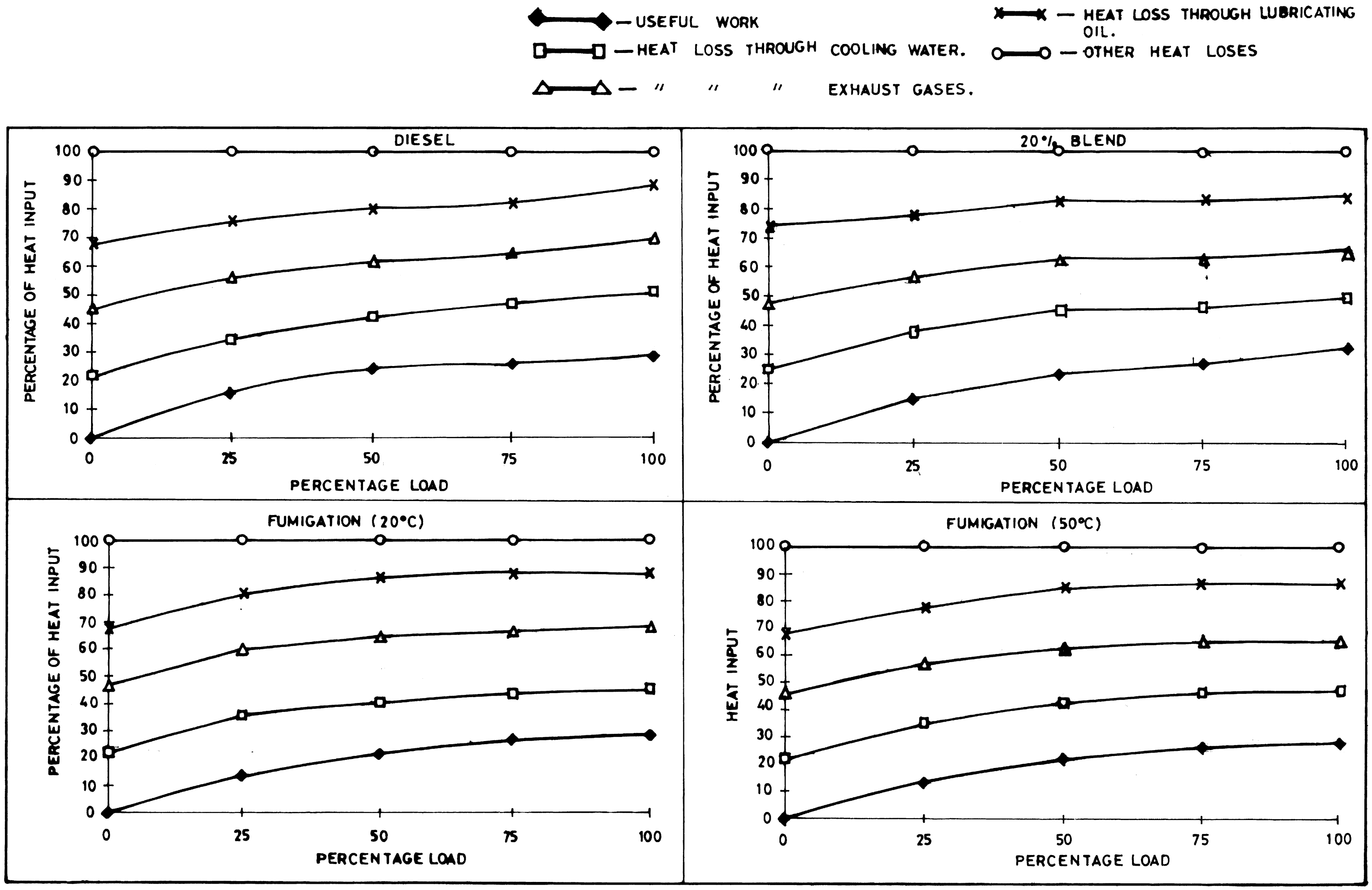

Fig. 3. Thermal balance of engine operating on diesel, 20% blend and fumigation.

4. Conclusion

The thermal balance of the engine operating on 5 and 10% ethanol–diesel blends and fumigated ethanol was not significantly different at the 5% level of significance when compared with diesel. However, the thermal balance of the engine operating on 15 and 20% ethanol–diesel blends was significantly different compared to diesel at the 5% level of significance.

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