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INFLUENCE OF SPEED AND LOAD VARIATION ON THE ENERGY BALANCE OF AN INTERNAL COMBUSTION ENGINE WITH GASOLINE

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Abstract. *The way to detect the released thermal energy for the operation of the engines loss for the purpose of optimizing performance, is through the experimental test and applying the thermal balance to the engine. The objective of this work is to perform a thermal balance in an Otto cycle engine involving the energy released due to the gasoline combustion process, power extracted from the engine shaft, rejected energy rates for the cooling system, for the exhaust system, for the lubrication system and for the environment in the form of heat and incomplete combustion, depending on the variation of engine speed and load. The experiment consisted of making measurements on an Otto cycle engine on a bench dynamometer varying engine speed and load, which made it possible to collect data to make the engine's thermal balance. The results showed that variations in engine speed and load influence the distribution of rates and efficiencies in the engine's thermal balance. At full load and in the condition of better thermal performance of the engine, thermal efficiency, the efficiency of the cooling, exhaust, lubrication, incomplete combustion and environmental systems were 30%, 12%, 29%, 5%, 0, 6% and 22%, respectively. Reducing the load causes a reduction in thermal efficiency and an increase in other efficiencies. Increasing the speed to the point of better thermal performance of the engine causes an increase in thermal efficiency, after which the increase in speed decreases the thermal efficiency and increases the other efficiencies. This research is important because it correlates the useful energy in an Otto cycle engine with the actual losses and this allows, based on the detected energy losses, an opportunity for a more detailed study to optimize and improve the performance of the engine.*

Keywords: *Otto cycle gasoline engines, thermal balance, thermal efficiency, bench dynamometer measurement, engine performance parameters.*

1. INTRODUCTION

The Otto cycle vehicle engine is a thermal machine that operates from the combustion of gasoline and ethanol inside the combustion chamber. A considerable part of this released thermal energy is used for the operation of the engine, the remaining energy lost to the cooling system, exhaust system, lubrication system, incomplete combustion and the environment (convection and radiation). The objective of this work is to perform a thermal balance in an Otto cycle engine involving the energy released due to the gasoline combustion process, power extracted from the engine shaft, rejected energy rates for the cooling system, for the exhaust system, for the lubrication system and for the environment in the form of heat and incomplete combustion depending on the variation of engine speed and load.

Heywood (2003) describes that the major energy balance terms are the brake power, the cooling heat loss, exhaust heat loss, miscellaneous heat loss that can be subdivided into lubricating oil loss, convection and radiation heat losses. A substantial part of the exhaust loss (12%) is radiated too the surroundings and the remainder end up in the cooling system. A considerable amount of the friction power (around 50%) is dissipated between the piston and piston rings and cylinder walls and is lost as heat to the cooling system. The rest of the friction power is dissipated in the bearings, valve mechanism or drives, auxiliary devices and is lost as heat to the lubricating oil or surroundings. Colin and Allan (2016) describe that, through the application of the energy balance in spark ignition engines, the overall heat loss is the sum of the heat transfer to the water, oil, and ambient air minus the friction work. The shaft work term reveals that the engine has a brake thermal efficiency of about 30%. About 45% of the energy is rejected in the exhaust, 10 to 15% is dissipated by friction, and 10 to15% is dissipated with heat loss. The results of an energy balance on a small, spark ignition automobile engine. This engine has an internal oil pump, and the heat rejected to the oil is carried away partly by the coolant and partly by the heat lost to ambient air. As the load increases and the intake manifold pressure increases from 0.4 bar to 0.8 bar, the energy converted to shaft work increases from about 20 to 30%, the coolant load decreases from about 40 to 30%, the exhaust energy varies from about 30 to 35%, and the heat lost to ambient air decreases from about 10 to 5%. The energy dissipated by friction decreases from about 14 to 7% for the same loads. The total heat loss from the gas to the coolant and ambient air during the cycle is about 28 to 36%. Jerald (2016) describes the energy balance in spark ignition engines of the fuel energy as follows: the brake work 26.02%, the friction 5.50%, the indicated work 31.52%, the heat loss 23.70%, the net flow out 44.11% and the fuel not used 0.68%. Abedin et al. (2013) carried out a theoretical review on energy balance in internal combustion engines operating with alternative fuels alcohols, biodiesel, hydrogen and liquid petroleum gas fuel. The basic balance of energy applied in this work was the first law of thermodynamics in a permanent regime and encompassed the energy supplied by the fuel, the output power delivered by the engine, the amount of heat rejected for the by the cooling water system, lubricating oil system, exhaust system and energy discarded for the environment. The review show that the heat loss reduces by a significant amount when the percentage of alcohols in the blend is greater than fifteen. The heat losses except the exhaust loss are higher while using biodiesels due to the presence of excessive oxygen molecules. The brake power decreases with the increase of cetane number. The cooling water loss is comparatively higher than the exhaust heat loss for biodiesels. Hydrogen supplementation with gasoline fuel significantly reduces the cooling water loss. Hydrogen addition in compressed natural gas reduces the actual combustion loss, increases the incomplete combustion loss, increases the wall heat loss and leaves no effect on the gas exchange loss. The lean operation of hydrogen increases the incomplete combustion loss and the complete combustion loss compared to gasoline fuel. Liquid petroleum gas burns faster than gasoline. Water addition in liquid petroleum gas has a favorable effect on reducing heat losses compared to pure liquid petroleum gas operation. The cooling water loss is very much lower in Low heat rejection engines for all alternative fuels compared to any other engines due to the low thermal conductivity of the insulations. The brake power increased in the range 1 to 3% at any loading conditions in low heat rejection engines. A large portion of the cooling loss is lost in the cylinder block side.

Yuksel and Ceviz (2003) investigated the effects of adding hydrogen to gasoline air mixture on thermal balance and performance in a four stroke, four-cylinder and spark ignition engine. Thermal balance tests were conducted for engine thermal efficiency, heat loss through the exhaust gases, heat loss to the cooling water, lubricating oil loss and radiation heat loss, while performance tests were analyzed to the brake power, specific fuel consumption and air ratio. The results showed that hydrogen addition to gasoline decreases the heat loss to cooling water, lubricating oil loss and radiation heat loss and the heat loss through the exhaust gas is nearly the same with pure gasoline experiments. Using hydrogen supplementation leads to a significant reduction in heat loss to cooling water and unaccounted losses by about 36 and 30% of the mean average values, respectively. Heat loss through the exhaust gases is nearly the same with that of gasoline because of the increase in the temperature of the exhaust gases. Hydrogen addition to gasoline results in a little decrease in brake power, and air ratio increases by about 13.81%. Additionally, specific fuel consumption decreases, while the engine thermal efficiency and the air ratio increase. Kuntesh et al. (2017) investigated the effects of to blend ethanol with petrol fuel in the performance of spark ignition engine and improve performance of engine. The analysis of the Ethanol on various parameters like input parameter like load, output parameter like specific fuel consumption, brake power, and brake thermal efficiency and the experimental energy analysis on spark ignition engine by ethanol-petrol blend was pure petrol, petrol with 25% ethanol, petrol with 40% ethanol and pure ethanol. The single cylinder, four-stroke petrol engine connected with an electrical load bank was using in this test. The test carried out with variation in engine load from low to high load conditions. The results showed the general performance of petrol with 40% ethanol found to be better compared to the pure petrol. Ethanol addition reduces the heating value of the petrol ethanol blends, therefore, more fuel is needed to obtain same power when blended fuels are used instead of petrol. The mechanical efficiency varying by 10 to 16%, 9 to 5% and 5 to 3% for petrol with 25% ethanol, petrol with 40% ethanol and pure ethanol, respectively compare to pure petrol at varying load condition. When the latent heat of the ethanol is low, as in the case of ethanol, the effect of cooling is not sufficient to overcome the effect of vapor. Which results in reducing thermal efficiency. The heat balance sheet indicates that some amount of heat wasted by fuel. This heat utilized to increase the brake power and some amount of heat is lost in exhaust gas. For lower load to higher load out of total power generated brake power of the engine is same for different fuel blend for petrol with 25% ethanol, petrol with 40% ethanol and pure ethanol compare to pure petrol. For exhaust gas energy however, it is reducing for lower load to higher load. It reduces 15% to 25% for different blend like for petrol with 25% ethanol, petrol with 40% ethanol and pure ethanol compare to pure petrol respectively. For Colling water energy, however it increases for lower load to higher load by 10% to 17% for different blend like for petrol with 25% ethanol, petrol with 40% ethanol and pure ethanol compare to pure petrol. Mehrnoosh et al. (2012) describes, using a thermodynamic model simulation for conventional four stroke engine, that good results could be presented when use compressed natural gas (CNG) as a fuel for internal combustion engine. The first thermodynamic law was the starting point for analyses the study, and the results have shown harmony when compared with experimental data. First, temperature and pressure were determined in the compression stroke, and the balance of internal energy was measured using the first thermodynamic law. As far as the results was concerned, when engine operates with CNG fuel, the indicated specific fuel consumption has been reduced 16% over this speed range. In addition, for the engine speed range has been observed a decrease of 33%, 60% and 53% of emissions of CO_2, CO and UHC concentration, respectively, while NO concentration increased by 50%. (Santana et al., 2019) used a dynamometer to correlate the vibration level of an internal combustion engine due to degradation of the lubricating oil. In another work (Santana et al., 2020) used a dynamometer to investigate the effect of fuel and lubricant oil on the vibration level of an internal combustion engine. Cerri et al. (2013) used a dynamometer to investigate on high octane number gasoline formulations for internal combustion engines impact the performance of spark ignition engine for passenger cars.

2. METHODOLOGY

A passive hydraulic dynamometer was used in order to measure the following parameters of an internal combustion engine: engine rotation, engine torque, water flow, oil flow, inlet water temperature, outlet water temperature, inlet oil temperature, outlet oil temperature, inlet air temperature, outlet combustion gas mixture temperature, fuel consumption, air consumption, fuel air ratio and lambda factor. The engine rotation was measured directly through the engine rotation sensor. The engine torque (t_b) was measured directly on the dynamometer, which measures the force released by the engine during the experiment. The water inlet temperature (t_{wi}) was measured at the water pump inlet and the water outlet temperature (t_{wo}) was measured close to the water temperature sensor in cylinder head. The water flow (m_{water}) was measured by a flow meter installed between the engine water outlet and the dynamometer bank cooling system. The inlet (t_{oi}) and outlet (t_{oo}) oil temperatures were measured by an oil cooling system installed in the engine oil pump. The oil flow (m_{oil}) was estimated according to the flow rotation curve provided by the manufacturer of the oil pump. The air temperature (t_{ai}) was measured at the intake manifold and the combustion gas temperature (t_{go}) was measured at the exhaust manifold. Fuel consumption (m_{fuel}) was measured by the gravimetric method, which directly measures the mass of fuel consumed during the experiment. Air consumption (m_{air}) was measured by the speed density method, which uses the air temperature, air pressure and engine rotation information to estimate the air flow in the engine during experiment. Air fuel ratio (A/F) was calculated as the ratio between the mass of air and the mass of fuel consumed by the engine during the test. The lambda factor (λ) was calculated as ration between air fuel ratio (A/F) and stoichiometric air fuel ratio $(A/F)_s$. Stoichiometric air fuel ratio depends on the type of fuel used in the engine and its measurement is made by the oxygen sensor installed in the exhaust manifold. The data for all measured and calculated parameters were acquired in a data acquisition system.

Figure 1 shows the experimental setup scheme during test in present study.

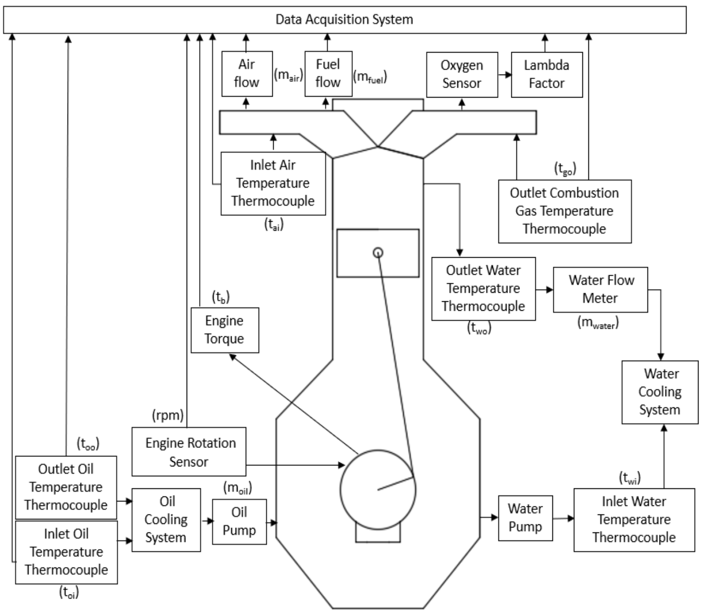

Figure 1 - Experimental setup scheme in present study.

The experiment was conducted with gasoline engine and full load according the (ISO 3046, 2002), which is the standard specifies the test conditions and methods for declaration of power and fuel consumption in internal combustion engines. Data were acquired under the following conditions: water average temperature at around 95°C and oil at 135°C. The test was performed at engine rotational speeds between 1500 to 6000 rpm with increments of 500 rpm. At each test speed, data was acquired for 2 minutes at a rate of 100Hz. After stabilization water and oil temperature conditions the data engine rotation, engine torque, water flow, oil flow, inlet water temperature, outlet water temperature, inlet oil temperature, outlet oil temperature, inlet air temperature, outlet combustion gas temperature, fuel consumption, air consumption, fuel air ratio and lambda factor has been acquired and stored acquisition system AVL PUMA software.

The first law of thermodynamics applied to the control volume involving the internal combustion engine can be represented by:

$$Q_{fuel} = Pb_{shaft} + Q_{water} + Q_{exhaust} + Q_{oil} + Q_{environment} + Q_{incconbustion} \tag{1}$$

where Q_{fuel} is the power supplied to the engine due to fuel burning, Pb_{shaft} is the output power delivered by the engine measured in dynamometer, Q_{water} is the amount of heat discharged to the water cooling system, $Q_{exhaust}$ is the amount of heat discharged to the exhaust system, Q_{oil} is the amount of heat discharged to the oil system, $Q_{environment}$ is the amount of heat rejected to the environment by convection and radiation and $Q_{inccombustion}$ is the rate of energy lost due to incomplete combustion. The Figure 2 shows the energy balance an internal combustion engine applied in this study and the control volume represented by the dashed line and energy rates crossing the boundaries of the volume of control.

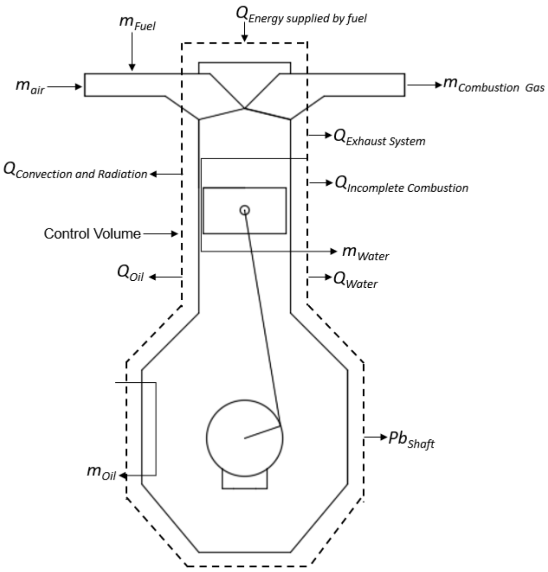

Figure 2 - Energy balance an internal combustion engine and energy rates crossing the boundaries of control volume.

The power supplied to the engine due to fuel burning Q_{fuel} is:

$$Q_{fuel} = \frac{\dot{m}_{fuel} \cdot Q_{LHV}}{3600} \tag{2}$$

where $\dot{m}_{fuel}$ is the mass flow rates of the fuel in (kg/h) and Q_{LHV} is the lower heating power of the gasoline fuel in (kJ/kg). The output power delivered by the engine measured in dynamometer Pb_{shaft} is:

$$Pb_{shaft} = \frac{2 \,.\, 9{,}8 \,.\, \pi \,.\, rpm \,.\, T_{shaft}}{60000} \tag{3}$$

where rpm is the engine speed in (rev/min) and T_{shaft} is the torque engine measured in dynamometer in (kg.m). The amount of heat discharged to the water-cooling system Q_{water} is:

$$Q_{water} = \frac{\rho_{water} \,.\, \dot{v}_{water} \,.\, cp_{water} \,.\, \Delta T_{water}}{3600} \tag{4}$$

where ρ_{water} is the specific mass of water in (kg/m^3), v_{water} is the water flow in (m^3/h), cp_{water} is the specific heat of water in (kJ/kg.K) and ΔT_{water} is the differs between water inlet and outlet temperature in the engine in (C°). The amount of heat discharged to the exhaust system $Q_{exhaust}$ is:

$$Q_{exhaust} = \frac{(\dot{m}_{fuel} + \dot{m}_{air}).\, cp_{air} \,.\, (T_{gases} - T_{air})}{3600} \tag{5}$$

where $\dot{m}_{fuel}$ is the mass flow rates of the fuel in (kg/h), $\dot{m}_{air}$ is the mass flow rates of the air in (kg/h), cp_{air} is the specific heat of air in (kJ/kg.K), T_{air} is the air inlet temperature measured in the intake manifold in (C°) and T_{gases} is the burned gases outlet temperature measured in the exhaust manifold in (C°). The amount of heat discharged to the oil cooling system Q_{oil} is:

$$Q_{oil} = \frac{\rho_{oil} \,.\, \dot{v}_{oil} \,.\, cp_{oil} \,.\, \Delta T_{oil}}{60000} \tag{6}$$

where ρ_{oil} is the specific mass of oil in (kg/m^3), v_{oil} is the oil flow in (L/min), cp_{oil} is the specific heat of oil in (kJ/kg.K) and ΔT_{oil} is the differs between oil inlet and outlet temperature in the engine in (C°). The rate of energy lost due to incomplete combustion $Q_{incccombustion}$ is:

$$Q_{inccombustion} = Q_{fuel}.(1 - \lambda) \tag{7}$$

where Q_{fuel} is the power supplied to the engine due to fuel burning in (kW) and λ is the factor lambda measured during test. The amount of heat rejected to the environment by convection and radiation $Q_{environment}$ is:

$$Q_{environment} = Q_{fuel} - Pb_{shaft} - Q_{water} - Q_{exhaust} - Q_{oil} - Q_{incconbustion} \tag{8}$$

The Table 1 shows the thermal properties of the specific mass of water, specific heat of water, lower heating power of the gasoline fuel, specific heat of air, specific heat of oil and specific mass of oil used in the calculation of the energy balance.

Table 1 – Thermal properties of water, gasoline fuel, air and oil lubricant used in the calculation of the energy balance, (Bergman et al., 2011) and (ANP, 2012).

Specific mass of water [kg/m³]	Specific heat of water [kJ/kg. K]	Lower heating power of the gasoline fuel [kJ/kg]	Specific heat of air [kJ/kg.K]	Specific heat of oil [kJ/kg.K]	Specific mass of oil [kg/m³]
970.00	4.20	40546.00	1.07	2.26	835.00

The properties of water and oil were evaluated respectively, at temperature of 85 and 103 C°. These temperatures represent the arithmetic averages of the outlet and inlet temperatures measured during the test at full load, according to table 1. The property of air was evaluated at temperature of 419 C°, assuming the specific heat of air, at mean exhaust temperature, as the average specific heat of the gases. The fuel used in this work is the commercial gasoline C with 27% anhydrous ethanol homologated for the Brazilian market.

The tests were carried out on a four-cylinder flex-fuel spark-ignition engine with the follow specification presented in Table 2.

Table 2 – Specification of spark ignition engine used in test.

Parameter	Type or value
Cycle	Four strokes
Compression ratio	10.35: 1
Bore × stroke	72.00 × 84.00 mm
Number of cylinders	4, in line
Total displacement	1368 cm³;
Intake system	Naturally aspirated
Maximum power	58.84 KW at 5750 rpm with gasoline
Maximum torque	112.80 N.m at 4000 rpm with gasoline

3. RESULTS AND DISCUSSION

This section will be present and discuss the results of application of the first law of thermodynamics the engine operating in the dynamometer working at full and no load conditions. In this work, the concept of control volume will be applied, which is represented by a certain closed space and this concept establishes that the net variation of the energy rates entering and leaving the control volume has to be equal to zero, (Heywood,2003).

The Table 1 shows the experimental engine data measured on the dynamometer during the test at full load condition.

Table 1 - The experimental engine data measured during the test at full load condition.

Engine rotation [rpm]	Engine torque [kg.m]	Water flow [m³/h]	Oil flow [l/min]	Air flow [kg/h]	Fuel flow [kg/h]	Inlet water temp [°C]	Outlet water temp [°C]	Inlet oil temp [°C]	Outlet oil temp [°C]	Inlet air temp [°C]	Outlet gas temp [°C]	Air/Fuel ration	Factor lambda
1500	12.90	1.46	12.50	85.73	6.43	82.05	89.48	84.50	96.50	22.20	660.30	13.33	1.00
2000	14.55	1.96	13.40	120.16	9.10	82.72	89.20	86.20	101.70	22.50	696.90	13.21	1.00
2500	14.73	2.45	14.30	164.16	12.51	83.07	89.40	88.80	105.40	22.80	751.60	13.18	1.00
3000	15.39	2.96	15.40	188.64	14.35	83.09	89.07	92.20	110.20	22.70	775.20	13.15	1.00
3500	16.23	2.98	16.60	222.87	16.96	82.45	89.40	95.50	113.30	23,20	806.30	13.14	0.99
4000	16.28	3.04	18.20	284.28	21.67	84.12	91.59	99.40	117.40	23.00	824.40	13.12	0.99
4500	16.49	3.05	19.10	331.32	25.27	80.31	89.79	103.30	121.50	24.30	864.20	13.11	0.99
5000	16.21	3.08	19.60	362.15	27.65	79.00	89.64	108.60	125.70	25.70	882.20	13.10	0.99
5500	13.71	3.15	19.80	384.59	29.40	78.37	89.84	113.80	131.50	25.00	897.80	13.08	0.98
6000	12.19	3.23	19.90	384.13	29.44	79.73	91.17	116.30	135.90	23.60	910.50	13.05	0.97

The Table 2 shows the experimental engine data measured on the dynamometer during the test at no load condition.

Table 2 - The experimental engine data measured during the test at no load condition.

Engine rotation [rpm]	Engine torque [kg.m]	Water flow [m³/h]	Oil flow [l/min]	Air flow [kg/h]	Fuel flow [kg/h]	Inlet water temp [°C]	Outlet water temp [°C]	Inlet oil temp [°C]	Outlet oil temp [°C]	Inlet air temp [°C]	Outlet gas temp [°C]	Air/Fuel ration	Factor lambda
1500	0.60	1.45	12.60	21.51	1.63	82.19	88.32	87.30	94.78	22.18	579.89	13.20	1.00
2000	0.70	1.94	13.30	23.62	1.79	83.46	88.24	88.76	96.64	22.20	582.68	13.20	1.00
2500	0.70	2.42	14.30	29.43	2.23	83.78	88.60	91.48	100.78	22.30	609.57	13.20	1.00
3000	0.80	2.97	15.20	37.75	2.86	83.24	88.60	93.14	102.39	22.40	634.21	13.20	1.00
3500	0.80	3.01	16.30	41.18	3.12	83.40	88.72	96.78	105.89	22,67	636.49	13.20	1.00
4000	0.90	3.04	17.90	53.20	4.03	83.77	90.67	99.89	109.54	22.34	645.87	13.20	1.00
4500	1.00	3.02	19.10	60.32	4.57	83.32	90.45	101.78	113.30	23.12	670.89	13.20	1.00
5000	0.90	3.06	19.30	73.64	5.58	83.02	90.64	105.10	115.78	23.43	695.34	13.20	1.00
5500	1.02	3.18	19.40	77.06	5.84	84.15	90.21	108.78	119.58	23.76	707.32	13.20	1.00
6000	1.05	3.25	19.60	96.62	7.32	84.37	90.78	111.89	123.54	23.78	723.15	13.20	1.00

Figure 3 shows the thermal energy rate calculated at full load condition.

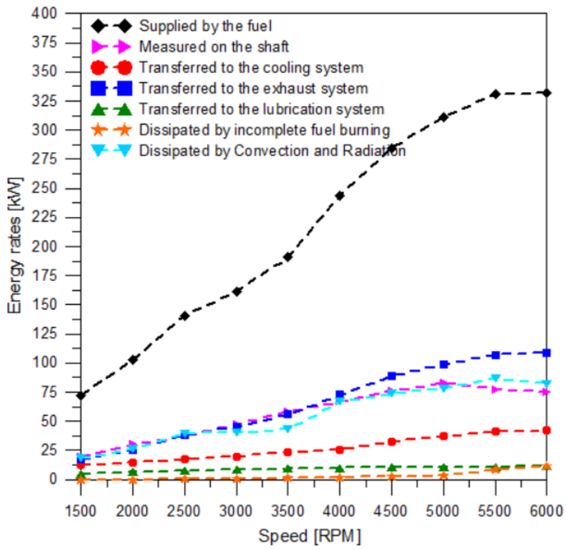

Figure 3 - Thermal energy rate calculated at full load condition.

At full load condition, the maximum energy rate supplied by the gasoline fuel was 331.52kW at 6000rpm, the average energy rate measured on the axis was 62.54kW and the maximum measured rate was 83.12kW at 5000 rpm, the average energy rate transferred to the cooling system was 24.56kW and the maximum transferred rate was 37.05kW at 5000 rpm, the average energy rate transferred to the exhaust system was 64.23kW and the maximum transferred rate was 108.82kW at 6000 rpm, the average energy rate transferred to the oil lubrication system was 5.00kW and the maximum transferred rate was 7.10kW at 6000 rpm, the average energy rate dissipated by the incomplete fuel burning was 1.55kW and the maximum transferred rate was 11.60kW at 6000 rpm and the average energy rate dissipated by convective and radiation heat was 59.65kW and the maximum transferred rate was 90.91kW at 5500 rpm. The rate of energy released by burning the gasoline inside the combustion chamber depends on the mass flow rate and the lower heating power value of the gasoline, as shown in equation 2. The increasing of this rate of energy released as the engine speed increases during the test can be justified because the gasoline flow increases with the increase in engine speed. The power measured on the engine crankshaft depends of the engine speed and torque, as shown in equation 3. The maximum power reached at 5000 rpm can be justified because the engine torque increases with the engine speed until reaching a maximum value between 3500 to 5000 rpm. The energy rates transferred to the cooling, exhaust and lubrication oil systems depend on the specific heats and temperature difference, as shown in equations 4, 5 and 6 respectively. The rate of energy lost due to incomplete combustion depends on the lambda factor, as shown in equation 7. This rate is zero at low speeds because the lambda factor is 1 and high at high speeds because the lambda factor is less than 1.

Figure 4 shows the thermal energy percentage calculated at full load condition.

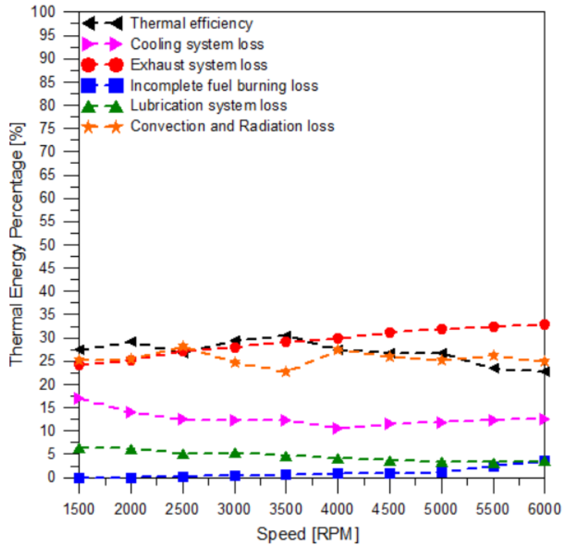

Figure 4 - Thermal energy percentage calculated at full load condition.

At full load condition, the average thermal efficiency of engine operated with gasoline fuel was 27.08% and the maximum thermal efficiency was 30.51% at 3500 rpm, the average thermal energy percentage rejected for cooling system was 12.33% and the maximum was 16.96% at 1500 rpm, the average thermal energy percentage rejected for exhaust system was 29.40% and the maximum was 32.82% at 6000 rpm, the average thermal energy percentage rejected for lubrication oil system was 2.40% and the maximum was 3.95% at 1500 rpm, the average thermal energy percentage rejected for environment by convection and radiation was 27.36% and the maximum was 30.45% at 4000 rpm and the average thermal energy percentage lost due to incomplete combustion was 0.7% and the maximum was 3.5% at 6000 rpm. (Heywood, 2013) describe that for the Spark ignition engines at maximum power the brake power ranges from 25 to 28%, the cooling heat loss ranges from 17 to 26%, the exhaust heat loss ranges from 34 to 45% and the miscellaneous heat loss ranges from 4 to 10%. Colin and Allan (2016) describe that for the small spark ignition automobile engine the shaft work term reveals that the engine has a brake thermal efficiency of about 30%. About 45% of the energy is rejected in the exhaust, 10 to 15% is dissipated by friction, and 10 to15% is dissipated with heat loss.

Figure 5 shows the thermal energy rate calculated at no load condition.

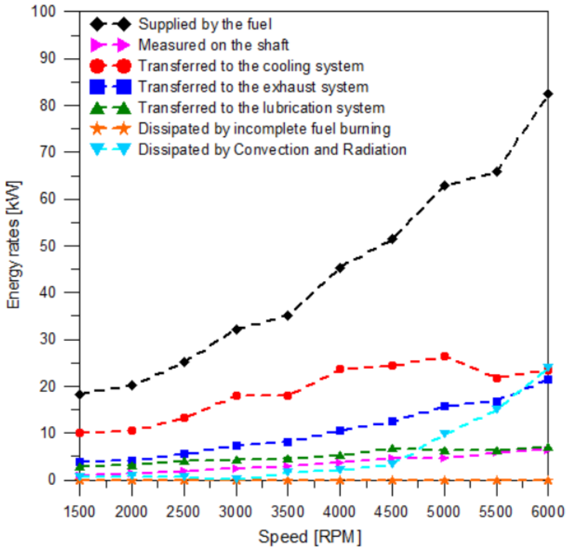

Figure 5 - Thermal energy rate calculated at no load condition.

At no load condition, the maximum energy rate supplied by the gasoline fuel was 82.44kW at 6000 rpm, the average energy rate measured on the axis was 3.28kW and the maximum measured rate was 6.46kW at 6000 rpm, the average energy rate transferred to the cooling system was 19.96kW and the maximum transferred rate was 26.39kW at 5000 rpm, the average energy rate transferred to the exhaust system was 9.33kW and the maximum transferred rate was 21.57kW at 6000 rpm, the average energy rate transferred to the oil lubrication system was 4.92kW and the maximum transferred rate was 5.99kW at 6000 rpm and the average energy rate dissipated by convective and radiation heat was 1.81kW and the maximum transferred rate was 23.85kW at 6000 rpm.

Figure 6 shows the thermal energy percentage calculated at no load condition.

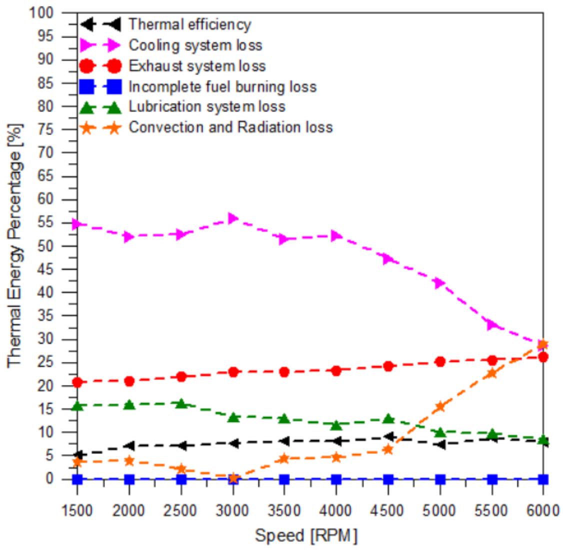

Figure 6 - Thermal energy percentage calculated at no load condition.

At no load condition, the average thermal efficiency of engine operated with gasoline fuel was 7.74% and the maximum thermal efficiency was 8.97% at 4500 rpm, the average thermal energy percentage rejected for cooling system was 51.81% and the maximum was 54.79% at 1500 rpm, the average thermal energy percentage rejected for exhaust system was 23.14% and the maximum was 26.16% at 6000 rpm, the average thermal energy percentage rejected for lubrication oil system was 13.01% and the maximum was 16.21% at 2500 rpm and the average thermal energy percentage rejected for environment by convection and radiation was 4.48% and the maximum was 28.93% at 6000 rpm. The rate of energy lost due to incomplete combustion was zero because at no load condition, the lambda factor is 1, consequently the thermal energy percentage lost due to incomplete combustion was zero too.

Figure 7 shows the distribution of the thermal energy percentage distribution at full load condition and 3500 rpm.

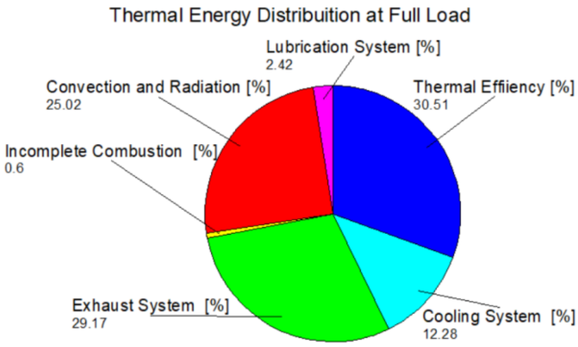

Figure 7 - Thermal energy percentage distribution at full load condition and 3500 rpm.

At full load condition and 3500 rpm (region of better engine thermal performance), the thermal efficiency was 30.51%, the thermal energy percentage rejected for cooling system was 12.28%, the thermal energy percentage rejected for exhaust system was 29.17%, the thermal energy percentage rejected for lubrication oil system was 2.42%, the thermal energy percentage rejected for environment by convection and radiation was 25.02% and the thermal energy percentage lost due to incomplete combustion was 0.6%. The tests in the present study were compared with the tests performed by (Yuksel and Ceviz, 2013) that used a four-stroke spark ignition engine and four cylinders with gasoline to investigate the energy balance and performance of engine coupled to hydraulic dynamometer. At 3534 rpm the brake power measured was 41.45kW, the heat loss through the exhaust system was 32.82kW, heat loss to the cooling water system was 18.88 kW, heat loss to the lubricating oil system was 12.71kW, radiation heat loss was 105.88kW and the engine thermal efficiency was 39%. In the same conditions of fuel used in teste (gasoline), type of engine (four-stroke spark ignition engine and four cylinders), load (full load) and speed (3500 rpm), these comparisons presented the energy rates and efficiencies of energy balance in the same order of magnitude.

Figure 8 shows the distribution of the thermal energy percentage distribution at no load condition at 3500 rpm.

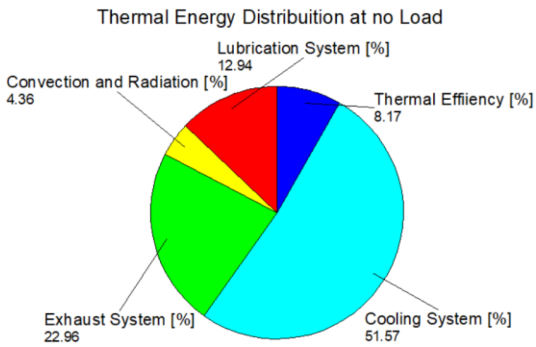

Figure 8 - Thermal energy percentage distribution at no load condition at 3500 rpm.

At no load condition and 3500 rpm, the thermal efficiency was 8.17%, the thermal energy percentage rejected for cooling system was 51.57%, the thermal energy percentage rejected for exhaust system was 22.96%, the thermal energy percentage rejected for lubrication oil system was 12.94% and the thermal energy percentage rejected for environment by convection and radiation was 4.36%. This experimental study confirms that the distribution of energy rates in the energy balance of a spark ignition four-stroke engine was influenced by the engine speed and the load. Up to 3500 rpm thermal efficiency increases with increasing rotation and after 3500 rpm thermal efficiency decreases with increasing rotation. Operating at lower loads, the engine achieves low thermal efficiency and operating at maximum load (maximum opening the throttle), the engine achieves maximum thermal efficiency, in the present study the maximum efficiency of 30.51% was achieved in the 3500 rpm engine speed. The energy rate transferred to the cooling system, exhaust system, oil lubrication system, energy rate dissipated by convective and radiation heat and rate of energy lost due to incomplete combustion decrease at low speeds and low loads. The thermal energy percentage rejected for cooling system and lubrication oil system increase and the thermal energy percentage rejected for exhaust system, rejected by convection and radiation and lost due to incomplete decrease at low speeds and low loads. This can be justified because at low speeds and low loads the thermal energy rate supplied by the gasoline fuel was lower. Kimura and Murakami (2012) investigated the effect of cylinder bore temperature and lubricant oil temperature on friction of gasoline engine. The results of this study demonstrated that the cylinder bore temperature was the main parameters that influenced the piston friction.

4. CONCLUSIONS

The results showed that, at full load conditions, the average thermal efficiency of engine operated with gasoline fuel was 27.08% and the maximum thermal efficiency was 30.51% at 3500 rpm, the average thermal energy percentage rejected for cooling system was 12.33% and the maximum was 16.96% at 1500 rpm, the average thermal energy percentage rejected for exhaust system was 29.40% and the maximum was 32.82% at 6000 rpm, the average thermal energy percentage rejected for lubrication oil system was 2.40% and the maximum was 3.95% at 1500 rpm, the average thermal energy percentage rejected for environment by convection and radiation was 27.36% and the maximum was 30.45% at 4000 rpm and the average thermal energy percentage lost due to incomplete combustion was 0.7% and the maximum was 3.5% at 6000 rpm. This experimental study confirms that the distribution of energy rates in the energy balance of a spark ignition four-stroke engine was influenced by the engine speed and the load. In the present study, up to 3500 rpm thermal efficiency increases with increasing rotation and after 3500 rpm thermal efficiency decreases with increasing rotation. The energy rate transferred to the cooling system, exhaust system, oil lubrication system, energy rate dissipated by convective and radiation heat and rate of energy lost due to incomplete combustion decrease at low speeds and low loads because at low speeds and low loads the thermal energy rate supplied by the gasoline fuel was lower. The thermal energy percentage rejected for cooling system and lubrication oil system increase and the thermal energy percentage rejected for exhaust system, rejected by convection and radiation and lost due to incomplete decrease at low speeds and low loads.

5. ACKNOWLEDGEMENTS

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