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ISSN (Online) 2456-1290
International Journal of Engineering Research in Mechanical and Civil Engineering (IJERMCE)
Vol 2, Issue 7, July 2017

Performance, Emission and Exergy Analysis of Diesel Engine for Varying Engine Coolant Temperatures

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***Abstract*— Only one third of the total fuel energy supplied to the conventional IC engine is converted to useful work, whereas a major part of the energy input is rejected to the exhaust gas and the cooling system. The idea of a low heat rejection (LHR) engine also called ''adiabatic engine" was introduced in its potential for improving engine thermal efficiency by reducing the heat losses. In this study, the LHR operating condition is implemented by increasing the engine coolant temperature (ECT). Experimentally, the engine is operated at varying ECTs of 25°C, 50° C, 70° C and 90° C in an effort to get trend-wise behavior without exceeding safe ECTs. And performance, emission and exergy analysis were conducted on the engine. The study uses a single cylinder Mitsubishi Kubota diesel engine operating at 1500 rpm to examine the cases having different ECTs. The study revealed that increasing ECT yields improvements in performance and emission characteristics of the engine while an increase in NOx emission was observed.**

***Index Terms*—Coolant temperature, Exergy analysis, Diesel engine, LHR engines.**

I. INTRODUCTION

From the perspective of thermodynamics, the IC engine may theoretically approach 100% efficiency since the IC engine converts chemical energy to mechanical energy; chemical energy is fully available to do useful work. The IC engine will still be limited, however, by second law considerations such as irreversible processes. Actual IC engines generally convert only approximately one-third of the fuel energy to useful work; the rest is rejected typically in the form of thermal energy to the coolant and exhaust.

The LHR concept has been of interest since the 1980s, when a substantial number of programs investigated the ''adiabatic engine". These programs aimed to improve engine efficiency with partial/complete suppression of heat loss through the combustion chamber walls. Thermal barrier coatings have been extensively used in LHR engine designs, which increase the thermal resistance of the combustion chamber walls and consequently increase the level of temperatures inside the cylinder. From the first law of thermodynamics, it can be expected that any retained energy by reducing heat losses through the chamber walls can be converted to useful work and consequently improve the fuel conversion efficiency.

Experimental studies reveal the fuel economy superiority of LHR engines over conventionally cooled engines. However, the facts do not necessarily substantiate a conclusion that LHR engines will outperform the conventional engine. In fact, some of the previous experimental work where the results are mixed show that a number of engine operating parameters are inter-related which can negatively affect the efficiency of a LHR engine. For instance, the higher operating temperature conditions decrease the volumetric efficiency, which in turn adversely influences the energy conversion efficiency.

So far, relatively little attention has been devoted to investigating the possibility of LHR engines with the approach of altering coolant temperature, which does not require significant modifications on engine. Reductions in heat losses to the coolant jacket can be supposedly implemented by raising the engine coolant temperature (ECT), due to the smaller temperature difference between the coolant and the wall. The main finding was that a higher ECT reduced the heat rejection to the coolant or the net in-

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cylinder heat transfer rate, the associated gains also include enhancing fuel evaporation and therefore mitigating hydrocarbon (HC) and CO emissions. It seems therefore logical to expand on the investigations of LHR concept through varying engine coolant temperature.

Typically, the operation conditions with ECT beyond 110^{O} C are unlikely to be realized due to the needs of avoiding excessive oil temperature. The maximum ECT is also restricted by the occurrence of nucleate boiling causing strong convection when the wall surface temperature goes beyond the coolant saturation temperature. In this case, the simulation method allows for extended studies at ECTs beyond the limited values on an actual engine.

Comparison test between experimental and simulation results of the effects of ECT on fuel conversion efficiency conducted by Li et al [1] demonstrated that increasing ECT yields slight improvements in net indicated fuel conversion efficiency, with larger improvements observed in brake fuel conversion efficiency. Effects of biodiesel in the LHR engine on its performance characteristics have been investigated by Hasimoglu et al [2] and improvements in the specific fuel consumption was observed. Partial reductions were found in CO, HC and smoke emissions, and increase in NOx emissions was observed while Aydin et al [3] coated piston and valves with NiCrAl and a mixture that consists of %88 ZrO2, %4 MgO and %8 Al2O3.

Buyukkaya et al [4] suggested that the method of injection timing retardation can be used to reduce NOx emissions released by LHR diesel engines. Coating piston surface and valves by using zirconium oxide (ZrO2) ceramic coating material by ISCAN [5] and molybdenum (Mo) by Hazar [6] also promised improvements in engine performance and emissions except for NOx emission.

Mortaza Aghbashlo et al [7] and Mustafa Canakcia et al [8] concluded that first law of thermodynamics is insufficient for evaluating energy utilization. In order to understand thermodynamic details of the operation and to determine the locations, causes, and magnitudes of energy waste in a thermal system, an exergy analysis must also be performed. Chaudhary [9] suggested that using the data from exergy analysis, waste heat recovery systems can be designed to utilize the heat and can improve performance

II. MATERIALS AND METHODS

Test rig setup**:** The Performance and exergy analysis test was conducted on MITSUBISHI KUBOTA diesel engine to find its variation with change in temperature of cooling water. The specifications of the engine are given in Table 1. The single cylinder Mitsubishi Kubota engine is modified for conducting the Performance and exergy analysis. Digital temperature indicator was fitted with the engine to get the temperatures of the cooling water inlet, cooling water outlet, and exhaust gas. A tank was installed in order to collect the cooling water and recirculate it through cooling water jacket. A ½ HP centrifugal pump was fitted in order to circulate the cooling water through the water jackets. Temperate of inlet side of cooling water was controlled by a bypass valve which mixes hot cooling water with cold water to obtain desired temperature for engine inlet cooling water.

Emission analysis**:** In the emission test the exhaust gas is analyzed by using an AVL Digas analyzer 444. The amount of hydrocarbon emission (%), NOx emission (ppm), carbon monoxide (%) and carbon dioxide (%), hydro carbon (%) and oxygen (%) emission can be found out.

Exergy analysis**:** Exergy is defined as the maximum theoretical useful work that can be obtained from a system before it coming to equilibrium with its surroundings. Exergy is generally not conserved as energy but destroyed in the system. It is an extensive property of the system and depends on both the state of the system and on the properties of the environment. The state of the environment is referred to as the dead state, defined by the environmental temperature, pressure and composition. In availability analysis of thermal systems, availability content of a system is divided into two parts:

I. Physical exergy: It is also called as thermo-mechanical availability (exergy), refers to the maximum useful mechanical work extractable as the system comes into thermal and mechanical equilibrium with the surrounding atmosphere

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II. Chemical exergy: It is also called as chemical availability (exergy), one part of the chemical availability of a system concerns only the system‟s species that are also present in the environment, known as diffusion availability. Whereas, the other part, called reactive availability, concerns the amount of work developed by allowing species of the system to chemically react with substances of the environment in order to form also environmental species.

It was assumed that the engine runs at steady state, the combustion air and exhaust gas are ideal gas mixtures, and potential and kinetic energy effects of the combustion air, exhaust gas and fuel stream are ignored. Reference environment is taken as P_0 = 1atm, T_0 = 30°C.

Assumptions are

- The engine operates at steady-state.
- The whole engine, including the dynamometer, is selected as a control volume.
- The combustion air and the exhaust gas each forms ideal gas mixtures.
- Potential and kinetic energy effects of the incoming and outgoing fluid streams are ignored.

The performance of engine is analyzed in light of the 2nd law of thermodynamics, which narrates the quality of energy and determines the lost opportunities to do work. An exergy balance is the availability of fuel energy used in different ways that includes availability in shaft, cooling water, exhaust, and destructed. Exergy efficiency is the ratio between exergy in product to total exergy input.

The available energy (A.E.) referred to a cycle is the maximum portion of energy which could be converted into useful work by ideal processes which reduces the system to a dead state.The minimum energy that has to be rejected to the sink by the second law is called the “Unavailable Energy (U.E.)”

The available energy refers to a diesel engine:

Q_1 = A.E. + U.E.,

W_{max} = A.E. = $Q1 -$ U.E.

Exergy balance of a control volume can be obtained using equation

$E_{x\,heat} + E_{xw} = \Sigma m_{in}\,\varepsilon_{in} - \Sigma m_{out}\,\varepsilon_{out} + E_{x\,dest}$

Where E_{xheat} - the exergy transfer associated with heat transfer at temperature T,

E_{xW} - exergy work rate, m is mass flow rate, ε is specific flow exergy

E_{xdest} - exergy destruction (irreversibility) rate.

(i) Availability of Fuel (A_{in}) in kW.The specific chemical exergy of liquid fuel on a unit mass basis can be evaluated as

$$A_{in} = [m_f \times LCV \times \{1.0401 + 0.1728\,(\tfrac{H}{C}) + 0.0432\,(\tfrac{O}{C}) + 0.2169\,(\tfrac{S}{C}) \times (1 - 2.0268\,(\tfrac{H}{C}))\}]$$

Where H, C, O, S are mass fraction of hydrogen, carbon, oxygen, sulphur

(ii) Shaft availability (A_s) = brake power of the engine in kW.

(iii) Cooling water availability (A_{cw}) in kW is

$$A_{cw} = Q_{cw} - [m_{we} \times C_{pw} \times T_a \times \ln(\tfrac{T2}{T1})]\ ,$$

where m_{we} is the mass of cooling water circulated through the cooling jacket, kg/s. C_{pw} is the specific heat of water kJ/kg K. $T1$ is the inlet water temperature passing through the cooling jacket, K. $T2$ is the outlet water temperature of cooling jacket, K. Ta is the ambient temperature, K.

$$Q_{cw} = m_{we} \times C_{pw} \times (T2 - T1)\ ,$$

(iv) Availability of exhaust gas (Aex), in kW is

$$A_{ex} = Q_{ex} - [m_{ge} \times T_a \times \{C_{pe} \ln(\tfrac{T5}{Ta}) - R_e \ln(\tfrac{Pe}{Pa})\}]$$

where Re is the specific gas constant of the exhaust gas in kJ/kg K. Pa is the ambient pressure, N/m^2. Pe is the final pressure, N/m^2. Ta is the ambient temperature, K. mge is the mass of exhaust gas, kg/s. T2 is the exhaust gas to calorimeter inlet temperature, K.

The properties of air can be taken for diesel exhaust gas calculations as an approximation. The error associated with it will be usually no more than about 2% by neglecting the combustion products.

(v) Destructed availability (A_d) in kW is

$$A_d = A_{in} - (A_s + A_{cw} + A_{ex})$$

(vi) Exergy efficiency (η_A) in %:

$$\eta_A = [1 - (\tfrac{Ad}{Ain})] \times 100.$$

III. RESULTS AND DISCUSSION

Brake specific fuel consumption

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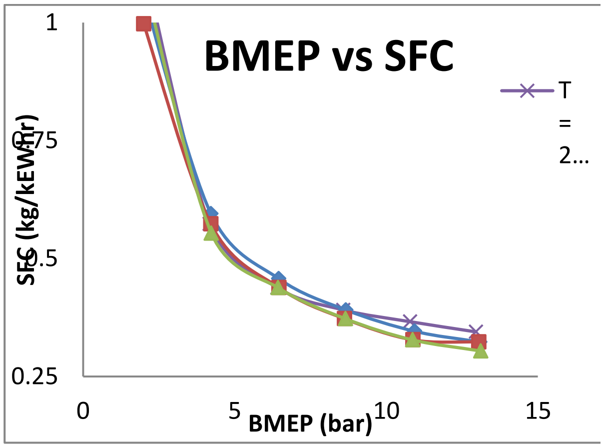

Fig: 1 SFC v/s Brake mean effective pressure

The specific fuel consumption at different cooling water inlet temperatures at 25° C, 50° C, 70° C, 90° C with diesel fuel are shown in Fig 1. From that it is clear that cooling water temperature, T= 90° C has the lowest SFC of 0.304 kJ/kW-hr followed by T= 70° C, T= 50°C with 0.323 kJ/kW-hr and T= 25° C- 0.344 kJ/kW-hr. A reduction 11.6% of SFC (from 0.344 kJ/kW-hr to 0.304 kJ/kW-hr) is achieved as ECT changes from 25°C to 90° C.

Brake thermal efficiency

The graph shows the variation of brake thermal efficiency (BTE) of the Kubota CI engine with brake mean effective pressure (BMEP) at varying ECTs of 25°C, 50° C, 70° C and 90° C. For all range of loads, the brake thermal efficiency of the engine at ECT 90° C was more than the efficiency of the engine at ECTs of 25°C, 50° C, 70° C. The maximum brake thermal efficiency at ECTs of 25°C, 50° C, 70° C and 90° C are 26.4%, 24.85%, 24.85%, 23.3% respectively. Reduced in cylinder heat transfer caused retained fuel energy which is converted into useful work.

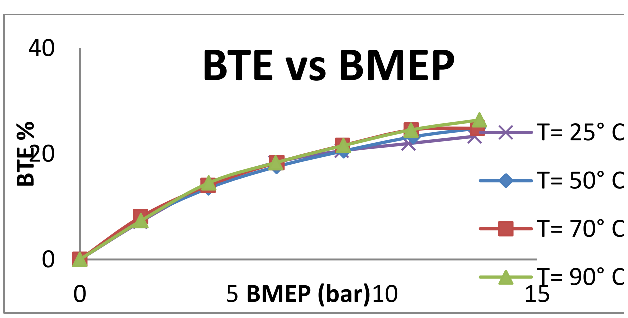

Fig: 2 BTE v/s Brake mean effective pressure

EMISSION CHARACTERICS

HC Emissions

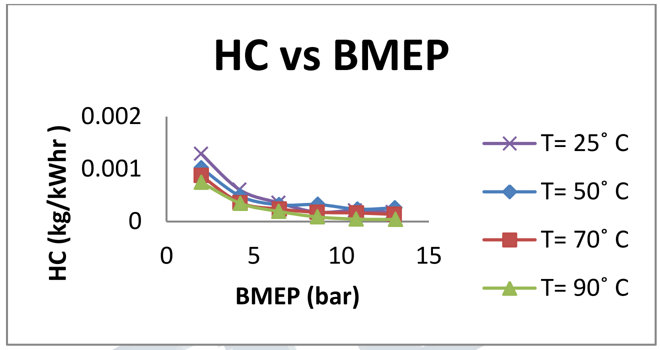

Fig: 3 HC v/s Brake mean effective pressure

The fig: 3 show the variation of unburned hydrocarbon emission of diesel at different ECTs of the CI engine with brake mean effective pressure. It has been observed that increasing ECT reduces HC pollutants, they serve as indicator of increase in combustion efficiency. The increased ECT reduces the amount of heat transfer to engine coolant that results in reduced in-cylinder heat transfer. This results in higher gas temperatures promoting the combustion towards completeness.

NOx Emissions

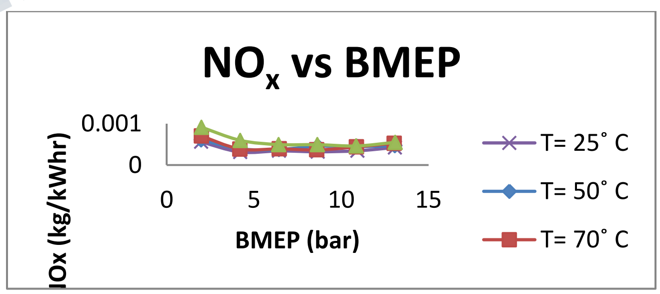

Fig: 4 NOx v/s brake mean effective pressure

The variations of NOx emission at different ECT's on CI engine with brake mean effective pressure are shown in fig: 4. The NOx is created mostly from nitrogen in the air and found in the fuel. NOx emissions increase with the increase in load. At low loads the nitrogen, N_2 is stable diatomic molecule and when load increases the temperature inside the cylinder also increases, this will dissociate the N_2 into

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ISSN (Online) 2456-1290
International Journal of Engineering Research in Mechanical and Civil Engineering (IJERMCE)
Vol 2, Issue 7, July 2017

monoatomic nitrogen and the more NOx will be formed. Here the NOx emission for ECT's 90° C, 70°C, 50° C, 25° C are 0.000549 kg/kWhr, 0.000526 kg/kWhr, 0.000466 kg/kWhr, 0.000425 kg/kWhr respectively. ECT 90° C have the highest NOx emission value due to the high in cylinder temperature and higher intensity of premixed combustion.

CO Emissions

The fig: 5 show the CO emissions at different ECT's with brake mean effective pressure. Carbon monoxide is a colourless and odourless poisonous gas. Poor mixing, local fuel rich regions and incomplete combustion etc. are the reason for CO emission. Here ECT= 90° C shows lowest CO emission. And ECT= 25° C shows the highest CO emission levels. The CO emission decreases with increase in ECT. At

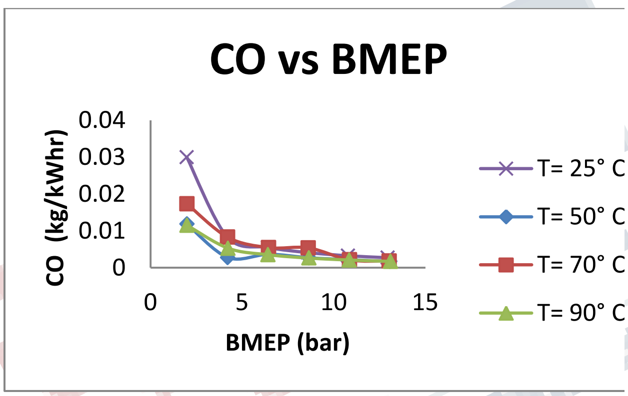

Fig: 5 CO v/s brake mean effective pressure

full load condition the CO emissions ECT's 90° C, 70° C, 50° C, 25° C are about 0.00172 kg/kWhr, 0.00178 kg/kWhr, 0.00173 kg/kWhr, 0.00273 kg/kWhr respectively.

IV. EXERGY ANALYSIS

Exergy analysis on diesel engine running on varyings ECT's was performed by using experimental data obtained at constant engine speed and six different loads applied. This analysis of diesel engines could provide comprehensive and deep information about the fuel combustion process and subsequent shaft work production, exergetic efficiency, exergy destructed etc.

Fuel Exergy

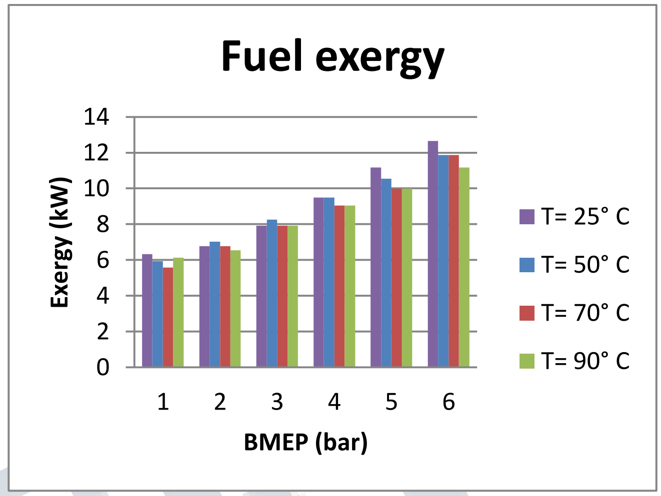

Fig: 6 Fuel exergy

Increasing engine load led to an increase in the fuel consumption and accordingly enhanced the rate of fuel exergy feeding into the engine. From the fig: 6, it is known that ECT at 90° C has lowest fuel exergy relating to lowest fuel consumption at 90° C ECT. ECT 25°C has highest fuel exergy denoting that the fuel consumption is larger compared to other ECTs for same fuel.

Exhaust gas exergy

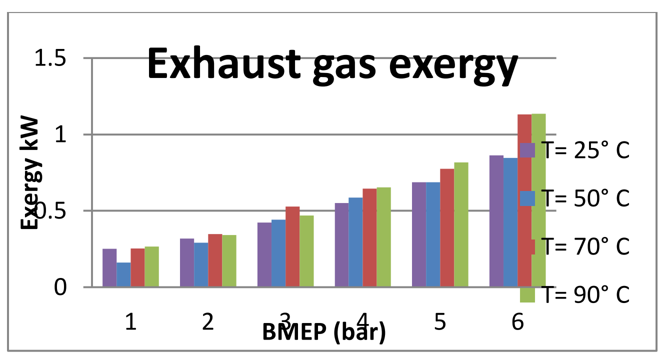

Fig: 7 Exhaust gas exergy

Exhaust gas exergy at varying ECTs are shown in the above fig: 7. All ECTs have the increasing exhaust gas exergy from no load to full load due to reduce in cylinder heat transfer. The increase in ECT increases

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ISSN (Online) 2456-1290
International Journal of Engineering Research in Mechanical and Civil Engineering (IJERMCE)
Vol 2, Issue 7, July 2017

exergy rate to the exhaust gases. It can be noted that exhaust gas exergy of ECT 90° C is 31.9% more than ECT 25°C. Heat loss through exhaust gases increased. The increasing ECT yields reduction in cylinder heat transfer but almost equivalent increase in the exhaust losses is observed.

Exergy of cooling water

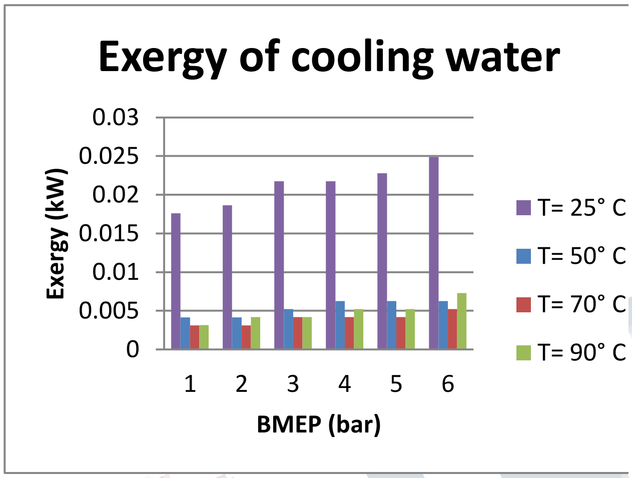

Fig: 8 Exergy of cooling water

Exergy transfer to cooling water is decreased as ECT is increased, reducing heat transfer through the walls. The retained energy by reducing heat losses through the chamber walls can be converted to useful work. It is observed that 70% reduction in coolant load was achieved for ECT 90° C compared to the baseline (ECT 25° C) engine.

Exergy destructed

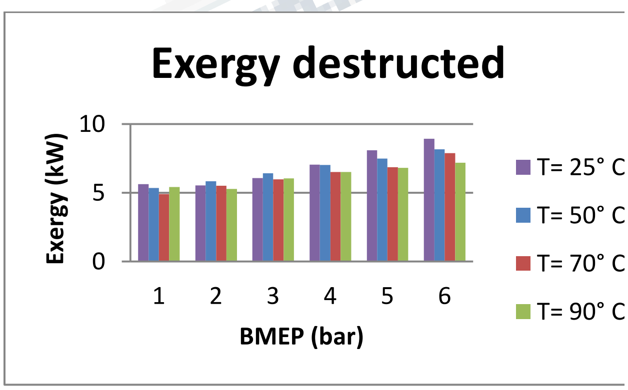

Fig: 9 Exergy destructed

Destructed exergy has an important role in the exergy analysis. The obtained results after calculations are shown in fig: 9. It is the highest loss of exergy from the total exergy followed by exhaust gas exergy, cooling water exergy. Destructed exergy is more for ECT 25° C and it reduces as ECT is increased to 90° C for medium and high loads. A reduction in 19.5% in destructed exergy is obtained when ECT is increased form 25° C to 90° C.

Exergy efficiency

Exergetic efficiency for varying ECTs with diesel fuel are shown in fig: 10. The exergy efficiency for ECT 90° C was found to be maximum as 35.77% followed by 70° C, 50° C, 25° C as 33.61%, 31.22% and 29.54% respectively. Exergy efficiency was increased upto 21% compared to baseline engine.

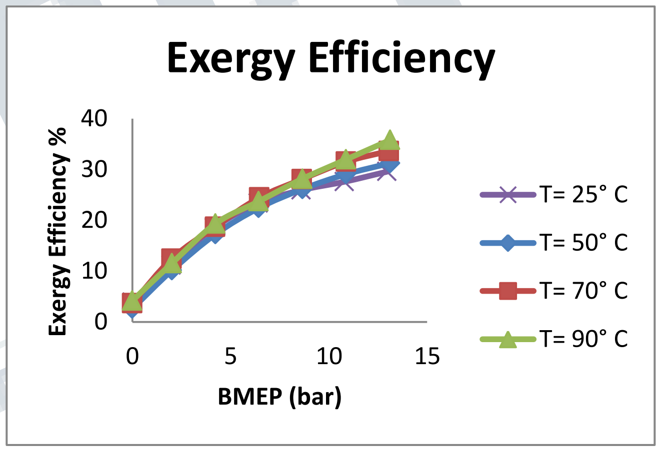

Fig: 10. Exergetic efficiency

V. CONCLUSION

The second law of thermodynamics was implemented to examine the strategy of altering engine coolant temperature to devise a version of LHR application in a conventional diesel engine. Analysis was performed on Mitsubishi Kubota diesel engine for ECT varying 25° C, 50° C, 70°C and 90° C with diesel as fuel. Experimental results showed reduction in specific fuel consumption as ECT is increased. Reduced in-cylinder heat transfer resulted in higher gas temperatures which promoted combustion toward completeness that resulted reduction of CO and HC pollutants, as they are sign of increase in combustion efficiency. NOx emission value increases with ECT due to the high in cylinder temperature and higher intensity of premixed

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ISSN (Online) 2456-1290
International Journal of Engineering Research in Mechanical and Civil Engineering (IJERMCE)
Vol 2, Issue 7, July 2017

combustion. Exergy analysis were showed increasing ECT yields reduction in cylinder heat transfer but almost equivalent increase in the exhaust losses, which explained why varying ECT achieves slight improvements in net indicated fuel consumption.

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