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EXERGY ANALYSIS OF A DIESEL ENGINE RUNNING ON STRAIGHT VEGETABLE OIL

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Abstract*-An experimental investigation has been carried out for exergy analysis of a diesel engine running on straight soybean oil (SVO) preheated at the temperatures of 50, 75, and 100*°C *with different loads at varying speeds of 1750, 2000, 2250 rpm. The results show that the brake thermal efficiency of the engine increases with preheat temperature of SVO and at 2250 rpm the values obtained are approximately 35, 36, 37.5% corresponding to 50, 75, 100*°C*, respectively. At 50 and 75*°C*, SVO provides lower efficiency than that with diesel fuel at room temperature of 30*°C *but at 100*°C*, the efficiencies both for SVO and Diesel fuel agree relatively well. The exergic efficiency also increases with preheat temperature, but the value of exergic efficiency is less by 1 to 1.15% in comparison with the brake thermal efficiency. The amount of exergy destroyed with exhaust gases is approximately 10% of the available energy. However, the major part of the available energy may be wasted out with coolant and with friction in the moving parts of the engine.*

Keywords: Brake Thermal Efficiency, Exergic Efficiency, Straight Soybean Oil, Preheat Temperature, Diesel Engine

1. INTRODUCTION

Diesel or compression-ignition engine has a wide range of applications due to their relatively high efficiency and their capability to meet current environmental and health standards. Straight vegetable (soybean) oil is used in this study in order to check its suitability as an alternative fuel to the conventional diesel oil. The methodology which utilizes both first and second laws of thermodynamics is called 'exergy analysis' (also known as 'availability analysis') and this methodology is usually used for better energy utilization. The exergy of a substance in a given state is a measure of the maximum obtainable work as the substance proceeds to the dead state while exchanging heat solely with the environment.

Investigations that have used the second law of thermodynamics to study internal-combustion engines in detailed manner date back to the late 1950s. Traupel [1] studied the availability destruction during combustion process of a naturally aspirated diesel engine and turbocharged diesel engine and found these values as 22.5% and 21.9% of the fuel availability, respectively. However, Patterson and van Wylen [2] described the availability for the compression and expansion strokes of spark-ignition engines incorporating the entropy values. Clarke [3] examined the Otto, Joule and Atkinson air-standard cycles from the perspective of availability and its associated destruction. Edo and Foster [4] reported an availability analysis for an engine which utilized dissociated methanol. Flynn et al. [5] showed in a turbocharged, intercooled diesel engine that of the original fuel availability about 46% was delivered as useful indicated work, 26% was destroyed, 10% was transferred as heat and 18% was exhausted. The performance of a diesel engine using the second law of thermodynamic was also analyzed further by Primus and Flynn [6, 7]. Al-Najem and Diab [8] reported in a turbocharged diesel engine that about 50% of the fuel availability is destroyed due to unaccounted factors such as combustion, 15% is removed via exhaust and cooling water and 1% is destroyed in the turbocharger.

Haq [9] studied the suitability of vegetable oils as diesel oil substitute. He reported that the calorific values of vegetable oils are slightly less than diesel values. Calorific values and energy densities of vegetable oils are very close to diesel values. However, viscosities of straight vegetable oils were found significantly higher than that of diesel. With adequate preheating, the viscosity of straight vegetable oil becomes comparable with that of the diesel fuel. Same inference was found for density. Rakopoulos and Giakoumis [10] reported on the use of a computer analysis to assess the performance of a turbocharged, indirect-injected, multi-cylinder diesel engine operated over a range of engine speeds and loads. Nwafor [11] studied the emission characteristics of diesel engine running on vegetable oil with elevated fuel inlet temperature. He reported that the hydrocarbon emissions were significantly reduced when running on plant oils. The CO production with heated plant oil is a little higher than the diesel fuel at higher loads. The heated vegetable oil showed marginal increase in CO_2 emission compared to diesel fuel.

In summary, most of the studies based on the second law analyses have used some type of engine simulation, although several based their results on measurements of the principal energy terms. The majority of the previous works have been completed for diesel engines. A few studies include some non-conventional characteristics like the use of alternative fuels. However, availability analysis of engine processes running with straight vegetable oil is still not that much focused area. In addition, the vegetable oils have a high potentiality of as alternative fuel for heat engines. So, it claims more methodical and elaborate investigations to ensure its efficient use in heat engines.

Therefore, in the present study, we have performed an experimental investigation for the exergy analysis of a diesel engine running on straight vegetable oil preheated at the temperatures of 50, 75, and 100°C with different loads at varying speeds of 1750, 2000, 2250 rpm.

2. EXPERIMENT

2.1 Experimental Facility

The experimental setup consists of engine test bed with diesel and soybean oil supply system, different metering and measuring devices along with test engine. Soybean oil feed systems consists with an electric heater placed in a heater box. Soybean oil is allowed to pass through the heater which is controlled by a temperature controller circuit which senses the inlet soybean oil before it enters into the fuel injection pump. A thermocouple is placed at the soybean oil inlet line just before the fuel pump which is connected with the overload relay of the temperature controller circuit. Schematic diagram of the experimental set up is shown in Fig. 1.

Experiments are carried out using a 33.75 kW (45 hp) Motor Diesel (VM) Engine of model 1053 SU manufactured in Italy. It is air cooled high speed direct injection four stroke diesel engine. It has three cylinders each having a bore of 105 mm and a stroke length of 110 mm. Its rated output is 33.5 kW at a rated speed of 2250 rpm. The engine is started by means of a self-starter motor run by a 12 V battery. It has a total swept volume of 2858 liter. The speed of the engine is controlled by lever control. The engine has an injection pressure of 230 kg/cm^2 and an injection order of 1-3-2. A water-brake dynamometer of model TFJ 250 manufactured by Tokyo Meter Corporation, Japan is used to apply desired loads on the engine to measure the torque. The maximum brake power of dynamometer is 250 hp and the revolutions at maximum braking hp point are 2500 to 5000 rpm. The dynamometer has the maximum revolutions of 5000 rpm and the maximum braking torque of 71.6 kg-m. The brake power is calculated from this torque and the corresponding speed which is recorded by a built-in tachometer. The dynamometer is connected with the engine by a universal joint and adequate care is taken so that no eccentricity might occur between them. Water is supplied to cool the dynamometer. A thermocouple is used to monitor the brake water temperature so that it can be kept within the limit.

Fuels (both diesel and soybean oil) are fed to the injector under gravity. A heating coil is dipped into the soybean oil container. In order to raise the temperature of the fuel up to a desired limit, the heater is controlled by a controller circuit which cut off or connects on the electric current by sensing the temperature of the fuel before it enters the injector. For this purpose, a sensor is used at the fuel line just before the fuel injector rail whose other end is connected to the controller circuit through an overload relay. The overload relay is very sensitive to voltage change. Whenever the sensor senses any rise in the fuel temperature over the set temperature its voltage difference changes and the relay breaks the circuit at that instant.

2.2 Experimental Procedure

Initially the test engine is run by diesel fuel at three different speeds of 1750, 2000 and 2250 rpm. Then the engine is run by 100% pure soybean oil preheated by an electric heater at the temperatures of 50, 75 and 100°C. At each pre-heating temperature, the engine is operated at 1750, 2000 and 2250 rpm by the soybean oil under different loading conditions. Engine speeds are maintain-

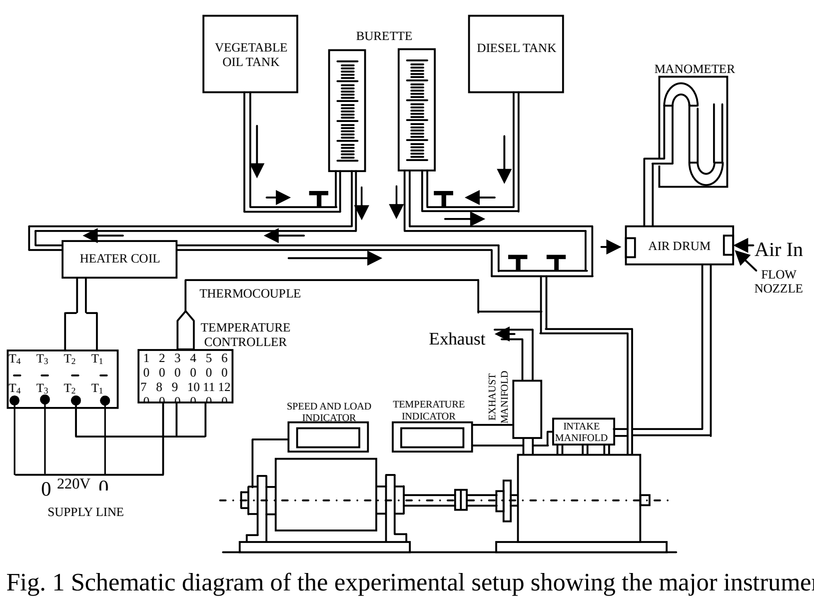

Fig. 1 Schematic diagram of the experimental setup showing the major instrumentation

ed within ±10 rpm of the desired speeds at different loads and the preheat temperature of the soybean oil is maintained within ±2°C of the desired temperature.

In the present study, BS standards for engine performance test BS 5514: Part I: 1982, equivalent to ISO 3046 and J 1349, ISO and SAE standards for the same respectively, is followed in the experimental and measurement procedures. Any other additional guidelines required are taken from the procedures used by Plint and Böswirth [12].

The values of the power and fuel consumption rate are properly derated using the derating procedure of BS 5514: Part I: 1982. For the purpose of determining the power and fuel consumptions of engines, the following reference conditions are set by the BS 5514 as: total barometric pressure = 100 kPa, air temperature = 27°C, relative humidity = 60% and charge air coolant temperature = 27°C. In this connection, it is mentioned worthy here that in finding the derating factors the mechanical efficiency (η_m) of the engine is assumed to be 80%. Because, according to the clause 10, note 4 of 5514: Part I: 1982, "The value of mechanical efficiency shall be stated by the engine manufacturer. In absence of any such statement, the value of η_m = 80% will be assumed."

All measuring and metering instruments are calibrated according to the above standards prior to record any parameter. After the calibration of the instruments, the parameters such as engine brake power and speed, fuel consumption rate, air flow rate and pre heat temperatures are measured so as to analyze the performance evaluation of the engine.

3. DATA REDUCTION

3.1 Brake Power Output

The brake power of the test engine is calculated using the following equations:

$$P(\mathrm{hp}) = \frac{W \cdot N}{2500} \tag{1}$$

$$P(\mathrm{kW}) = 0.746 \cdot P(\mathrm{hp})$$

Where, P(hp) is the brake power output in hp, P(kW) denotes brake power output in kW, W is load on the dynamometer in kg, and N is the speed of the shaft connected to the dynamometer in rpm.

3.2 Standardized Brake Power

The engine brake power are standardized or derated according to the BS 5514: Part 1 1982 as follows:

$$P_b = \frac{P(\mathrm{kW})}{\alpha} \tag{2}$$

Where, P_b is the adjusted brake power output in kW, P (kW) denotes brake power output in kW and α denotes the power adjustment factor.

3.3 Fuel Consumption Rate

The fuel consumption rate of the engine is calculated as below:

$$\dot{m}'_f = \frac{(\sigma \cdot \rho_w \cdot V)}{t} \times 60 \times 10^{-6} \tag{3}$$

Where, $\dot{m}'_f$ is the fuel consumption rate in kg/hr, V is the volume of fuel flown in time t in ml, ρ_w is the density of water in kg/m^3, σ is the specific gravity of the fuel and t time to measure V (ml) fuel flow in minutes.

3.4 Standardized Fuel Consumption Rate

The fuel consumption rate are standardized or derated according to the BS 5514: Part 1 1982 as follows:

$$\dot{m}_f = \frac{\dot{m}'_f}{\beta} \tag{4}$$

Where, $\dot{m}_f{}'$ is the experimental measurement of fuel flow rate and β is the correction factor for standardizing the value of fuel flow rate.

3.5 Brake Specific Fuel Consumption

The brake specific fuel consumption is computed as follows:

$$bsfc = \frac{\dot{m}_f}{P_b} \tag{5}$$

Where, $bsfc$ denotes the brake specific fuel consumption in kg/kW-hr, $\dot{m}_f$ is the adjusted fuel consumption rate in kg/hr and P_b is the adjusted brake power output in kW.

3.6 Brake Thermal Efficiency

Brake thermal efficiency is a dimensionless parameter that relates the engine output to the necessary input (fuel flow). The ratio of the brake power output to the rate of fuel energy input is known as brake thermal efficiency η_b. It is given by

$$\eta_b = 3600 \times \frac{P_b}{\dot{m}_f \cdot LHV} = \frac{3600}{bsfc \cdot LHV} \tag{6}$$

Where P_b denotes the brake power measured at fly wheel in kW, $\dot{m}_f$ is the mass flow rate of fuel in kg/h, and LHV denotes the lower heating value of fuel in kJ/kg.

3.7 Brake Mean Effective Pressure

Brake mean effective pressure is a constant pressure, which acting on the piston area through stroke would produce brake power at the flywheel. Therefore,

$$P_b = bmep . \frac{V_d N}{2} \tag{7}$$

This mean pressure is fiction, but is useful as it is roughly comparable even in very different engines, as these different engines burn same fuel, necessarily under approximately the same conditions and hence similar pressures. In SI engines, typical WOT *bmep* available nowadays is between 0.9 and 1.1 MPa. However, CI engines have 25-30% lower value of bmep because of much leaner combustion.

3.8 Fuel Chemical Exergy or Exergy Input

For diesel and soybean oil fuel, the fuel exergy input is calculated by the following relation:

$$A_{in} = Q_{in} \left[1.0374 + 0.0159 \frac{y}{x} + 0.0567 \frac{z}{x} + 0.5985 \frac{w}{x}\left(1 - 0.1737 \frac{y}{x}\right) \right] \tag{8}$$

In the present study, the formula of diesel is taken as

C14.4H24.9 [13]. So, putting x = 14.4 and y = 24.9, the above correlation for diesel fuel becomes

$$A_{in} = 1.06489\, Q_{in} \tag{9}$$

In our present study we have taken the formula of soybean oil fuel as C56H102O6 [14]. So, putting x = 56 and y = 102, z = 6 and w = 0, the above correlation for soybean oil fuel becomes

$$A_{in} = 1.0724\, Q_{in} \tag{10}$$

3.9 Exergy of the Exhaust Gas

The availability of the exhaust gases is calculated from the relation:

$$A_{exst} = Q_{eg} + \dot{m}_{eg} T_0 \left[C_{p,e} \ln\left(\frac{T_0}{T_{e,0}} \right) \right] \tag{11}$$

Where $\dot{m}_{eg}$ is the mass flow rate of exhaust gases $= \dot{m}_f (1 + A/F)$, Q_{eg} is the heat energy of exhaust gases $= C_{p,e} \dot{m}_f (1 + A/F)(T_{exst} - T_{db})$, and $C_{p,e}$ is the specific heat of the exhaust gas.

The specific heat of exhaust is calculated using the formula of specific heat of mixture of gases as follows:

$$C_{p,e} = \sum_{i=1}^{n} N_{i,00} C_{p,i} \Big/ \sum_{i=1}^{n} N_{i,00} = \frac{N_{CO_2} C_{p,CO_2} + N_{H_2O} C_{p,H_2O}}{N_{CO_2} + N_{H_2O}} \tag{12}$$

Assuming that complete combustion of both of the diesel fuel and soybean oil fuel produces only carbon dioxide (CO_2) and water vapor (H_2O). The constant-pressure specific heat equations of the individual components of exhaust gas are adapted from the data in NASA SP-273. These are as follows:

$C_{p,CO2} = (2.401 + 8.735\times10^{-3}T - 6.607\times10^{-6}T^2 + 2.002\times 10^{-9}T^3)\, R_u$

$C_{p,H2O} = (4.0 - 1.108\times10^{-3}T + 4.152\times10^{-6}T^2 - 2.964\times10^{-9}T^3 + 0.807\times10^{-12}T^4)\, R_u$

Where, R_u is the universal gas constant.

The percent availability destruction in exhaust gases is calculated as below:

$$A_{eg} = \frac{A_{exst}}{A_{in}} \tag{13}$$

The percent availability destruction from other sources like friction, cooling, etc. is calculated as below:

$$A_{uncounted} = \left[100 - \left(A_{shaft} + A_{eg}\right)\right] \tag{14}$$

3.10 Second Law Efficiency

The second-law (or availability) efficiency is defined as the ratio of the maximum possible useful work output from the heat engine cycle to the fuel exergy input to the engine. The term 'percent availability output at shaft' (A_{shaft}) is used instead of the term 'second-law efficiency'. So, in the present study the second-law efficiency is calculated as:

$$A_{shaft} = \text{(brake power output/availability input)} = \frac{P_b}{A_{in}} \tag{15}$$

4. RESULTS AND DISCUSSION

The availability (or exergy) input to an internal combustion engine is contained in its fuel chemical availability. In CI engines, the input availability contained in fuel is converted into (i) useful brake output availability, (ii) availability transferred to cooling medium, (iii) availability transferred to exhaust gases, (iv) availability destroyed in engine accessories, turbocharger, cooling fan, etc. and (v) availability destroyed due to friction and radiation heat loss to surroundings.

The availability is destroyed or lost due to different irreversibilities such as combustion losses, friction losses, heat loss to lubricating oil, power consumed by auxiliary equipment (axial blower in the present test engine), radiation losses, fluid flow losses, etc. Availability transfer to cooling medium (air in this case) has been included into the above category because of the lack of facilities to compute the heat loss in the air-cooling system by axial blower. Availability destruction due to all these sources is combinedly expressed by $A_{uncounted}$. This is justified by the fact that availability transfer to cooling medium in an air-cooled engine is only a fraction (less than 2.5%) of the availability input to the engine [8]. Availability destruction in exhaust gases, A_{eg} is evaluated separately.

The brake thermal efficiency, η_b with percent availability output at shaft of an engine, A_{shaft} running on diesel fuel are demonstrated against brake mean effective pressure, *bmep* in Fig. 2. In the figure, two efficiencies are represented in one graph to have a clear comparison. Furthermore, to obtain a more precise contrast between these two efficiencies, Fig. 3 is plotted where both the efficiencies are demonstrated against *bmep* for the rated speed of 2250 rpm. It can be revealed that A_{shaft} is somewhat less than η_b throughout almost the entire range except at lower values (< 0.15 MPa) of *bmep* as shown in Fig. 3. Both of the graphs urge the fact that the capability of an engine to successfully utilize the available energy is rather less than that articulated in brake thermal efficiencies. The major reason behind this is that the fuel chemical availability, A_{in} is about 3.35% to 7.25% as higher (depending on the chemical formula of the fuel) than the heat input, Q_{in} calculated from the lower (or higher) heating value of the fuel. And this available energy (or exergy) cannot be interpreted into shaft work due to inherent irreversibilities associated with pertaining engine processes.

Figure 4 shows the comparison between the brake thermal efficiencies, η_b using soybean oil fuel at pre-

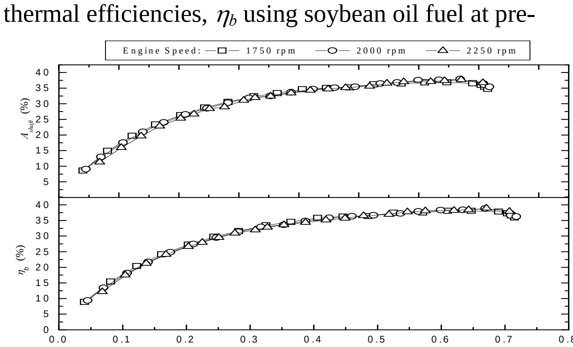

Fig. 2: Comparison of brake thermal efficiency with percent availability output at shaft as a function of brake mean effective pressure run by diesel fuel

heated temperature of 100°C and diesel fuel at room temperature (30±2°C). It can be seen that brake thermal efficiency using soybean oil at 100°C has become comparable to that obtained using diesel fuel at room temperature. The heavier the fuel the less it is being atomized by the fuel injector. So, at lower temperatures the explosive mixture of air and fuel remains non-homogeneous which affects the burning efficiency, thus resulting in poor η_b. However, penetration rates of soybean oil spray is increased and cone angle decreased as the viscosity is reduced by increasing the temperature of the oil. This contributes in improving the engine performance at elevated fuel inlet temperatures. Almost identical elucidation is associated with the Fig. 5 which shows a comparison between the percent availability output at shaft, A_{shaft} using soybean oil fuel at 100°C and diesel fuel at room temperature (30±2°C).

The availability destructions from different sources using soybean oil fuel at 100°C and diesel fuel at room temperature (30±2°C) are demonstrated against *bmep* in Fig. 6. Although the availability destruction in exhaust gases, A_{eg} does not vary that much the availability destruction in friction, cooling, etc., $A_{uncounted}$ using soybean oil drops slightly compared to that using diesel fuel. This is due to the fact that increased fuel inlet temperature helps to reduce some of the loss heads like combustion loss, fluid flow loss, etc., but the magnitude is not that significant (< 3.5%). Therefore, the major part (almost 50%) of the available energy is wasted and lost due to uncounted factors. The availability destroyed in exhaust gases is about 10% of the available energy. This may be utilized directly for preheating purpose, thereby improving the actual efficiency of the engine.

Figures 7 and 8 are presented to show the outcomes of preheating of soybean oil fuel whether it improves the efficiency (both η_b and A_{shaft}) of the engine or not. Shown in Fig. 7 is a comparison among the brake thermal efficiencies, η_b of the test engine run by diesel fuel at room temperature (30±2°C) and soybean oil fuel at three different preheat temperatures (50, 75 and 100°C), all at the rated speed of 2250 rpm. It can be revealed that the the brake thermal efficiency of the engine increases with preheat temperature of SVO and at 2250 rpm the values obtained are approximately 35, 36, 37.5% corresponding to 50, 75, 100°C, respectively. Engine with soybean oil at50°C preheat temperature exhibit lower efficiency than that with diesel fuel. As the preheat temperature is incre-

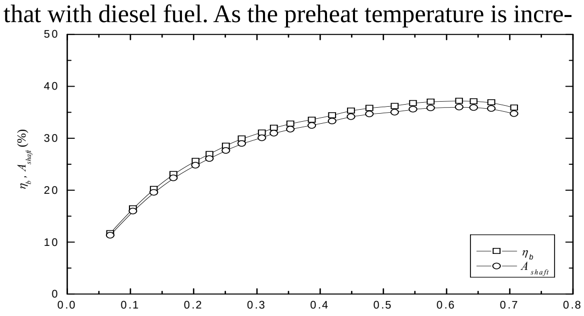

Fig. 3: Comparison of percent availability output at shaft with corresponding brake thermal efficiency as a function of brake mean effective pressure run by diesel fuel at 2250 rpm

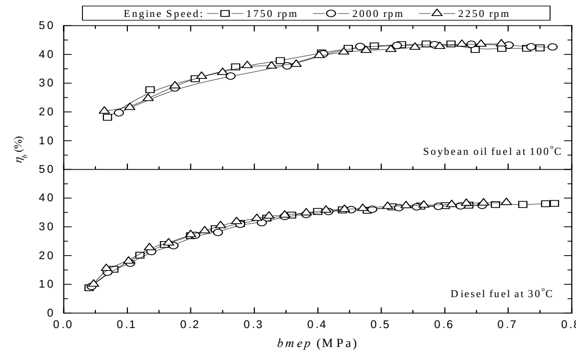

Fig. 4: Comparison of brake thermal efficiency as a function of brake mean effective pressure run by soybean oil fuel at 100°C preheat temperature and diesel fuel at 30±2°C

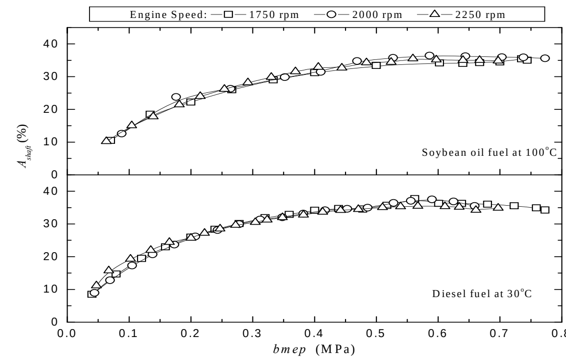

Fig. 5: Comparison of percent availability output at shaft as a function of brake mean effective pressure run by soybean oil fuel at 100°C preheat temperature and diesel fuel at 30±2°C

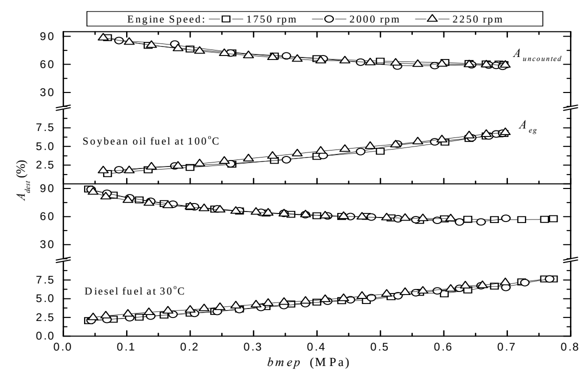

Fig. 6: Comparison of availability destruction as a function of brake mean effective pressure run by soybean oil fuel at 100°C preheat temperature and diesel fuel at 30±2°C

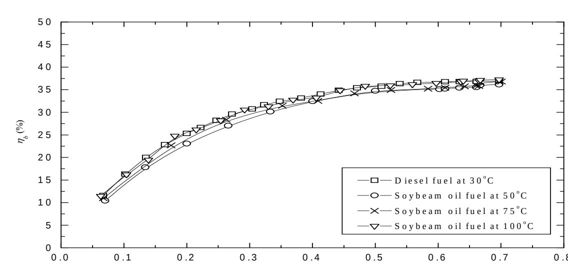

Fig. 7: Comparison of thermal efficiency as a function of brake mean effective pressure run by preheated soybean oil fuel and diesel fuel both at 2250 rpm

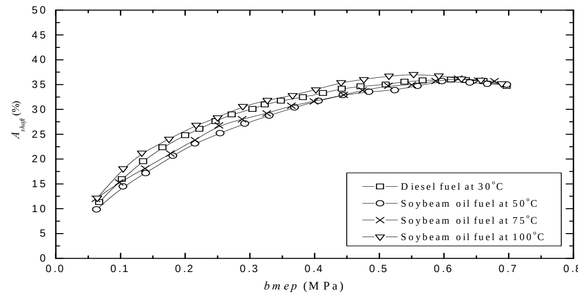

Fig. 8: Comparison of percent availability output at shaft as a function of brake mean effective pressure run by preheated soybean oil fuel and diesel fuel both at 2250 rpm

ased (75 and 100°C) the performance of the engine using soybean oil fuel gets closer to that using diesel fuel. The performance curve of soybean oil fuel at 100°C preheat temperature becomes comparable with that of diesel fuel. This happens due to the fact that at 100°C, both the density and viscosity of soybean oil become very much closer to those values of diesel fuel [9]. So, at this temperature the spray pattern for soybean oil fuel bear a resemblance to that for diesel fuel. Also, the penetration of fuel injection becomes comparable.

The percent availability output at shaft, A_{shaft} using soybean oil fuel at three different preheat temperatures (50, 75 and 100°C) and diesel fuel at room temperature (30±2°C) also show equivalent behavior, as seen in Fig. 8, to those of brake thermal efficiency.

5. CONCLUSION

Experimental study of a diesel engine running on diesel fuel and preheated soybean oil fuel is carried out. The thermodynamic analysis of preheated SVO as the alternative of the conventional diesel fuel for CI engine is analyzed. Engine performance parameters are obtained using both the conventional practice and the availability analysis. The latter is carried out using empirical correlations presented by other researchers. The major inferences that may be drawn from the present study are listed as follows.

(1) Preheated straight soybean oil may be a practical replacement of the conventional diesel fuel with a small power and efficiency drop.

(2) The brake thermal efficiency of the engine increases with preheat temperature of SVO and at 2250 rpm the values obtained are approximately 35, 36, 37.5% corresponding to 50, 75, 100°C, respectively. At 50 and 75°C, SVO provides lower efficiency than that with diesel fuel at room temperature of 30°C but at 100°C, the efficiencies both for SVO and Diesel fuel agree relatively well.

(3) The exergic efficiency also increases with preheat temperature, but the value of exergic efficiency is less by 1 to 1.15% in comparison with the brake thermal efficiency.

(4) The major part (almost 50%) of the available energy is wasted and lost due to uncounted factors. The availability destroyed in exhaust gases is about 10% of the available energy. This may be utilized directly for preheating purpose, thereby improving the actual efficiency of the engine.

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