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Investigation of a wet ethanol operated HCCI engine based on first and second law analyses

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ABSTRACT

In this paper, a conceptual wet ethanol operated homogeneous charge compression ignition (HCCI) engine is proposed to shift the energy balance in favor of ethanol. The investigated option, HCCI engine is a relatively new type of engine that has some fundamental differences with respect to other prime movers. Combined first and second law of thermodynamic approach is applied for a HCCI engine operating on wet ethanol and computational analysis is performed to investigate the effects of turbocharger compressor ratio, ambient temperature, and compressor adiabatic efficiency on first law efficiency, second law efficiency, and exergy destruction in each component. First law and second law efficiencies are found to be an increasing function of the turbocharger pressure ratio, while they are found to be a decreasing function of the ambient temperature. The effect of turbocharger pressure ratio on exergy destruction is found to be more significant than compressor efficiency and ambient temperature. Exergy analysis indicates that maximum exergy is destroyed in HCCI engine which represents about 90.09% of the total exergy destruction in the overall system. Around 4.39% exergy is destroyed by the process of heat transfer in fuel vaporizer and heat exchanger. Catalytic converter contributes about 4.08% of the total exergy destruction. This will provide some original information on the role of operating variables and will be quite useful in obtaining the optimum design of ethanol fuelled HCCI engines.

1. Introduction

The world is presently confronted with the twin crisis of fossil fuel depletion and environmental degradation. Indiscriminate extraction and lavish consumption of fossil fuels have led to reduction in underground carbon based resources. The need to exploit bio-origin based alternative fuels to quench the world energy thrust has long been realized. By now, it has been conclusively realized that internal combustion engines form an indispensable part of modern life style. They play a vital role in transportation and modern mechanized agricultural sector [1].

Ethanol has been promoted as a domestically produced bio fuel that could reduce both CO_2 and dependence on foreign oil. However, ethanol critics have reported that up to 70% more energy is required to produce ethanol than the energy it contains [2]. The energy balance of ethanol has been studied in detail. Recent publications [3,4] have reported a slight energetic advantage from producing and using ethanol from corn. These studies reported that the greatest fraction of the energy necessary for making ethanol is spent in water removal. Distillation and dehydration requires 37% of the total energy in ethanol and co products. Direct utilization of wet ethanol may therefore considerably shifts the energy balance in favor of ethanol because its use eliminates the amount of energy spent in dehydration.

Prime movers typically available for transportation and for distributed power generation (spark ignition and diesel engines) do not perform well with wet ethanol. Combustion in SI engines occurs through propagation of a flame that initiates at the spark plug and spreads through the combustion chamber. High concentration of water in fuel results in substantial dilution of the fuel air mixture. Excessive dilution slows down the propagation of the flame, resulting in misfire. In addition to this, SI engines need to operate at low compression ratio (less than ~ 11) and low intake temperature (less than $\sim 60°C$) to avoid knock. The low compression ratio limits the engine efficiency and the low intake temperature limits the humidity of the intake gases, because condensation

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in the intake may damage engine components or mix with oil and reduce its lubricity. It is also difficult to run a diesel engine on wet ethanol. Pure ethanol itself is not a good fuel for diesel engines due to its high resistance to auto-ignition. Resistance to auto-ignition increases if wet ethanol is used because water inhibits chemical reaction by cooling down the air inside the combustion chamber as it evaporates. Therefore it is difficult to obtain appropriate combustion of wet ethanol in a diesel engine in a short time available for combustion [5].

Homogeneous charge compression ignition (HCCI) engines have recently emerged as alternative prime movers for transportation and stationary applications. HCCI combustion is fundamentally different from combustion in SI engines and diesel engines. HCCI combustion is the thermal auto-ignition of a premixed fuel air mixture, with no flame propagation (as in SI engines) or mixing controlled combustion (as in diesel engines). HCCI engines are intrinsically fuel flexible. Any fuel can be used as long as it can be evaporated and then heated by compression to a temperature hot enough for auto-ignition. The operating limits for HCCI engines are therefore set by the requirement to heat up the fuel and not by the need to propagate a flame across the combustion chamber. HCCI engines can therefore burn wet ethanol that cannot be burned in either diesel or SI engines. Unlike SI engines, HCCI engines are not limited to low intake temperature or low compression ratio due to knock. HCCI engines can operate at high intake temperature and are likely to be more efficient than SI engine due to their high compression ratio, and do not require spark plugs or the three way catalyst and may therefore have lower maintenance costs than SI engines, and produces extremely low NO_X emissions [6].

Frias et al. [7] proposed the hypothesis that a HCCI engine can run efficiently on wet ethanol fuel and that utilizing wet ethanol fuel in HCCI engines improves the energy balance of ethanol production . They carried out the first law of thermodynamics or energy balance analysis of wet ethanol operated HCCI engines and reported that direct utilization of wet ethanol improves the energy balance for ethanol and maximizes the overall system efficiency. The first law of thermodynamics or energy balance method is the most commonly used, however, this is concerned only with the conversion of energy, therefore it cannot show how or where irreversibilities in the system or process occur. Thus while producing the final design result; energy balance analysis is incapable on its own of locating sources of losses. Second law analysis gives much more meaningful evaluation by indicating the association of irreversibilities or exergy destruction with combustion and heat transfer processes and allow a thermodynamic evaluation of energy conservation option in power cycles, and thereby provides an indication that points the direction in which engineers should concentrate their efforts to improve the performance of thermal power systems [9,10].

In the published literature, there is no study that was conducted to report the second law analysis of HCCI engines operating on direct utilization of wet ethanol. The present study deals with the combined application of first and second law analyses for the performance evaluation of wet ethanol operated HCCI engine. The exergy balances for cycle and its components are presented and compared to energy balances. One of the objectives of the present research is to calculate thermodynamic irreversibilities with the intention of identifying and quantifying malfunction of system or its components that result in an increase in the amount of resource required to obtain the same output, or in more general terms, in a decrease in overall performance of the system. The effect of ambient temperature, pressure ratio and turbocharger compressor efficiency have been investigated on the main thermodynamic parameters of the system, viz. first law efficiency, second law efficiency and irreversibility in each component of the system in order to determine system components where improvement could be made to enhance performance. Emphasis is placed throughout on realistic component modeling based on current technological constraints.

Comparison of fuel properties of ethanol, gasoline and diesel is given in Table 1. Ethanol has high octane number and high latent heat of vaporization. Ethanol also has simple molecular structure. It burns efficiently and improves combustion efficiency.

2. System description

Fig. 1 shows the schematic diagram of a HCCI engine system operating on 35% ethanol in water (by volume) as a fuel .The system is similar to engine which would typically be used in heavy trucks or stationary engine applications except that we have added a regenerator for heat transfer from the exhaust into the intake gases, and a vaporizer to evaporate the fuel prior to entering the combustion chamber. Ambient air enters the compressor which delivers air at high pressure and temperature followed by the regenerator, this raises the air temperature. Next, liquid ethanol in water is injected into the vaporizer, where it evaporates and mixes with air. The evaporation process in the vaporizer produces a homogeneous mixture of ethanol, water vapor and air, which then enters the HCCI engine. The ethanol water air mixture inducted into the cylinder heats up as it mixes with residual gases within the cylinder. After combustion, exhaust gases enter the catalytic converter at a higher temperature and exit the catalytic converter at further higher temperature due to heat release from conversion of unburned fuel, hydrocarbons (HC) and carbon monoxide (CO) that were not reacted in the engine combustion chamber. Gases from catalytic converter at higher temperature flow into the turbine, generating power that drives the turbocharger compressor. After circulating through the turbine, the exhaust gases exchange heat with the intake air in the regenerator and then leave the system at ambient pressure and higher temperature.

2.1. Assumptions used in the analysis

For the purpose of analysis, the following assumptions are made for wet ethanol operated HCCI engines [7,8,11].

Table 1
Comparison of fuel properties of ethanol, gasoline and diesel [14,15].

Property	Ethanol	Gasoline	Diesel
Chemical formula	C_2H_5OH	C_8H_{18} (mixture of hydrocarbons)	$C_{12}H_{26}$ (mixture of hydrocarbons)
Molecular weight (kg/kmol)	46	≈114	≈170
Oxygen percent (weight %)	34.8	Nil	Nil
Specific gravity	0.785	0.72–0.78	0.84–0.88
Boiling temperature at 1 atm (°C)	78.3	30–225	190–280
Latent heat of vaporization (kJ/kg)	840	305	270
Stoichiometric air/fuel ratio	9.0	14.6	14.5
Lower heating value of fuel (MJ/kg)	26.9	44.0	42.5
Research octane number (RON)	107	92–98	–
Motor octane number (MON)	89	80–90	–

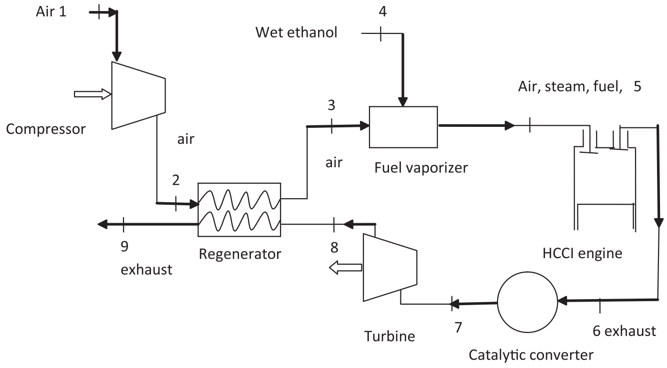

Fig. 1. Schematic diagram of wet ethanol operated HCCI engine.

- The HCCI combustion process is represented by a time dependent, variable volume well mixed reactor.
- The overall burn rates of HCCI combustion are typically fast and if correctly phased in the cycle could approximate the ideal Otto cycle. Fig. 2 illustrates the PV diagram for an ideal HCCI operating condition.
- Fuel and water are fully vaporized prior to entering the engine combustion chamber.
- The volumetric efficiency of the engine is 100% meaning the volume of air and fuel inducted into the engine cylinder is equal to the displacement of the cylinder.
- Thermal losses in ducts are negligible.
- Pressure drop in heat exchanger and duct is negligible.
- Turbocharger mechanical losses are neglected because most of recent researchers rejected these losses while reporting the thermodynamic analysis of internal combustion engines [7,8].
- Engine is assumed to operate at a constant engine speed (1800 RPM) and non-varying load.
- Displacement volume per cylinder is 2400 cm^3 and geometric compression ratio is 16:1. The residual gas fraction is assumed lower ($f = 0.03$) due to higher compression ratio.
- The minimum volume fraction of ethanol in water necessary for efficient HCCI operation is 35% by liquid volume. Consequently we have assumed 35% ethanol in water by volume mixture as the supplied fuel.

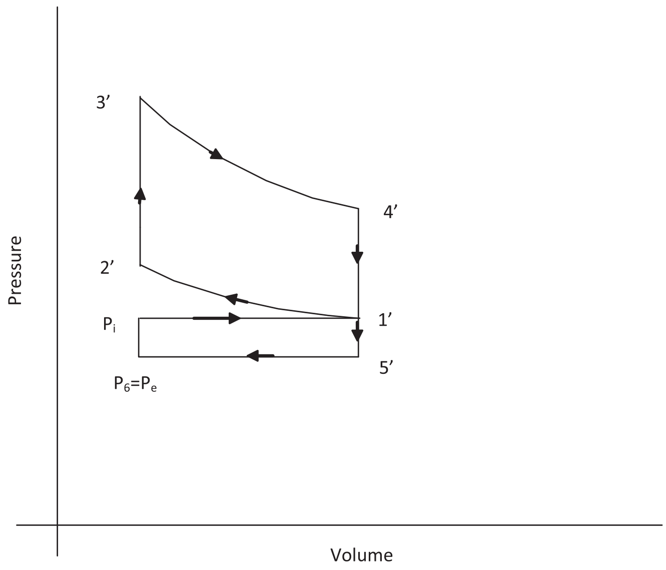

Fig. 2. PV diagram of an HCCI operating condition.

3. Irreversibility analysis

Exergy analysis of a process is a supplement to energy analysis, used to assess the work potential of input and output material and heat streams, and pinpoint irreversibility losses. Since the variation of kinetic and potential exergy is considered negligible in this study, physical and chemical exergies are taken into account at the system.

The physical exergy is obtained from thermal and mechanical processes, while the chemical exergy change is associated with chemical processes. These equations can be expressed on the basis of their definitions as following:

$$e^{\mathrm{ph}} = \sum_i [(h_i - h_0) - T_0(s_i - s_0)] \tag{1}$$

$$e^{\mathrm{ch}} = \sum_{i=1}^{i=n} x_i\left(\mu_i^* - \mu_{i,0}\right) \tag{2}$$

where, x_i is the mole fraction of chemical species, μ_i^* is the chemical potential of given species at restricted dead state and $\mu_{i,0}$ is the chemical potential at ultimate dead state.

In realistic cycles, there are irreversibilities associated with every component. The irreversibilities have negative effect on the system performance, and they can be evaluated via exergy analysis, which is a useful tool in determining the way, for achieving better performances. According to Bejan [13], the exergetic balance applied to a fixed control volume is given by the following equations.

$$\sum \dot{Q}_j\left(1 - \frac{T_0}{T_j}\right) - \dot{W} + \sum_{\mathrm{in}} \dot{m}e_{\mathrm{in}} - \sum_{\mathrm{out}} \dot{m}e_{\mathrm{out}} - E_{\mathrm{D}} = 0 \tag{3}$$

where $\dot{Q}_j$ is the heat transfer rate to the system, $\dot{W}$ the mechanical power supplied by the system, $\dot{E}_{\mathrm{D}}$ the exergy destruction rate because of irreversibilities and e is the exergy transfer associated with the stream of matter [12]. The exergy destruction is the amount of exergy that is lost to the environment and cannot be used anywhere.

3.1. Turbocharger compressor

Inducted air first flows through a turbocharger compressor which raises the outlet temperature and pressure. For a given compressor polytropic efficiency and pressure ratio $\{r_{\mathrm{p}} = (P_2/P_1)\}$. The pressure ($P_2$) at the compressor outlet can be calculated from the following equation.

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma\eta_{\mathrm{C}}}} \tag{4}$$

where γ is the specific heat ratio $c_{\mathrm{p}}/c_{\mathrm{v}}$. and η'_{c} is the polytropic efficiency

The energy balance yields compressor work w_{C} and the exergy balance for compressor gives the exergy destruction $e_{\mathrm{D,compressor}}$

$$e_{\mathrm{D,compressor}} = w_{\mathrm{C}} + \left(e_1^{\mathrm{ph}} - e_2^{\mathrm{ph}}\right) \tag{5}$$

$$w_{\mathrm{C}} = (h_2 - h_1) \tag{6}$$

$c_{\mathrm{p,air}}$ and $c_{\mathrm{v,air}}$ are evaluated from the following relations [12].

$$c_{\mathrm{p,air}} = \frac{8.314}{28.96}\left(3.653 - 1.337\,T10^{-3} + 3.294\,T^2 10^{-6} - 1.913\,T^3 10^{-9} + 2.763\,T^4\,10^{-13}\right) \tag{7}$$

$$c_{\mathrm{v,air}} = c_{\mathrm{p,air}} - 0.287 \tag{8}$$

3.2. Regenerator

Air flowing out of the compressor is directed to a regenerator (an exhaust to intake air heat exchanger) that preheats the intake air with the exhaust gases. Intake preheating is necessary for complete ethanol in water mixture evaporation in the vaporizer. The performance of the regenerator is specified through the effectiveness defined as

$$\epsilon_{\mathrm{h}} = \frac{(\dot{m}c_{\mathrm{p}})_2 (T_3 - T_2)}{(\dot{m}c_{\mathrm{p}})_{\min} (T_8 - T_2)} \tag{9}$$

where subscripts 2, 3, and 8 indicate state points in Fig. 1 and $(\dot{m}c_{\mathrm{P}})_2$ is the minimum of $(\dot{m}c_{\mathrm{P}})_2$ and $(\dot{m}c_{\mathrm{P}})_8$. In this case, the minimum heat capacity rate $(\dot{m}c_{\mathrm{Pmin}}) = (\dot{m}c_{\mathrm{P}})_2$ and the Eq. (9) can be expressed as

$$T_3 = T_8 \epsilon_{\mathrm{h}} + T_2 (1 - \epsilon_{\mathrm{h}}) \tag{10}$$

Pressure drop through regenerator is neglected. Therefore $P_3 = P_2$ and $P_8 = P_9$.

Applying the energy balance equation on the heat exchanger yields

$$H_3 - H_2 = H_8 - H_9 \tag{11}$$

The exergy balance of heat exchanger yields exergy destruction

$$e_{\mathrm{D,HE}} = \left(e_2^{\mathrm{ph}} - e_3^{\mathrm{ph}}\right) + \left(e_8^{\mathrm{ph}} - e_9^{\mathrm{ph}}\right) \tag{12}$$

3.3. Fuel vaporizer

HCCI engine operates with premixed fuel air intake charge. The wet ethanol fuel is evaporated in the vaporizer before reaching the engine. The vaporizer consists of a low-pressure injector, similar to those used in throttle body or port injected gasoline engines. The low-pressure injector provides good mixing between the fuel and the hot incoming air so that ethanol and water can fully evaporate in the vaporizer. A system such as this has demonstrated good performance in HCCI experiments [8]. Law of conservation of mass gives $\dot{m}_{\mathrm{O_2,5}} = \dot{m}_{\mathrm{O_2,3}}$, $\dot{m}_{\mathrm{N_2,5}} = \dot{m}_{\mathrm{N_2,3}}$, $\dot{m}_{\mathrm{H_2O,5}} = \dot{m}_{\mathrm{H_2O,4}}$ and $\dot{m}_{\mathrm{Ethanol,5}} = \dot{m}_{\mathrm{Ethanol,4}}$. Applying the mass balance equation yields

$$m_3 + m_4 = m_5 \tag{13}$$

where $\dot{m}$ is the mass flow rate of O_2, N_2, H_2O, and ethanol, respectively, and the number indicates the location in Fig. 1. Temperature at the end of evaporator can be calculated by energy balance as

$$H_3 + H_4 = H_5 \tag{14}$$

The relation between H and T is given by

$$\Delta H = mc_{\mathrm{p}} \Delta T \tag{15}$$

Exergy balance for fuel vaporizer gives exergy destruction and is given by.

$$e_{\mathrm{D,FV}} = e_3^{\mathrm{ph}} + e_4^{\mathrm{ph}} - e_5^{\mathrm{ph}} \tag{16}$$

3.4. Homogeneous charge compression ignition engine

The premixed ethanol–water mixture from vaporizer flows in the HCCI engine. The intake charge mixes with hot residual gases and is compressed by piston. High compression ratio $(r_{\mathrm{c}} = 16)$ during compression process heats up the charge, leading to ignition and combustion. Combustion in HCCI engine is a bulk auto-ignition process controlled by chemical kinetics [7,13]. The combustion efficiency $(\eta_{\mathrm{Combustion}})$ is assumed to be 94%. There is no external exhaust gas recirculation in this engine system model. The mixed temperature of residual gases and the fresh charge is calculated by the following equation [15] and inlet/exhaust pressure ratio is assumed to be 1.4 [16].

Intake process (i – 1′)

$$T_{1'} = \frac{T_i (1 - f)}{1 - \frac{1}{(nr_{\mathrm{c}})[P_{\mathrm{e}}/P_{\mathrm{i}} + (n-1)]}} \tag{17}$$

where 1′ indicates the location at the beginning of the compressor stroke (Fig. 2) and $P_{1'} = P_{\mathrm{i}}$

Polytropic compression stroke (1′–2′)

$$P_{2'} = P_{1'} \left(\frac{V_{1'}}{V_{2'}}\right)^n = P_{1'} (r_{\mathrm{c}})^n \tag{18}$$

$$T_{2'} = T_{1'} (r_{\mathrm{c}})^{(n-1)} \tag{19}$$

Constant volume heat addition (2′–3′)

$$T_{3'} = T_{2'} + q_{\mathrm{in}} (1 - f)/c_{\mathrm{v}} \tag{20}$$

$$P_{3'} = P_{2'} \left(\frac{T_{3'}}{T_{2''}}\right) \tag{21}$$

Polytropic expansion stroke (3′–4′)

$$P_{4'} = P_{3'} \left(\frac{1}{r_{\mathrm{c}}}\right)^n \tag{22}$$

$$T_{4'} = T_{3'} \left(\frac{1}{r_{\mathrm{c}}}\right)^{(n-1)} \tag{23}$$

Polytropic blow down (4′–5′)

$$T_{5'} = T_{4'} \left(\frac{P_{4'}}{P_{\mathrm{e}}}\right)^{\frac{(1-n)}{n}} \tag{24}$$

$$P_{5'} = P_{\mathrm{e}} \tag{25}$$

Constant pressure polytropic exhaust stroke (5′–6′)

$$T_{\mathrm{e}} = T_{5'} \tag{26}$$

$$P_6 = P_{5'} = P_{\mathrm{e}} \tag{27}$$

$$f = \frac{1}{r_{\mathrm{c}}} \left(\frac{P_{\mathrm{e}}}{P_4}\right)^{\frac{1}{n}} \tag{28}$$

Residual gas fraction $(f = 0.03)$ is lower due to higher compression ratio.

After making entropy balance over the compression process of HCCI engine using second law of thermodynamics, the exergy destruction during expansion process may be obtained as

$$e_{\mathrm{D,compression}} = T_0 S_{\mathrm{gen}} = T_0 \left[(s_2 - s_1) + \frac{q_{\mathrm{surrounding}}}{T_0}\right] \tag{29}$$

Exergy destruction during heat addition $e_{\mathrm{D,Heat\ addition}}$

$$e_{\mathrm{D,Heat\ addition}} = T_0 \left[(s_{3'} - s_{2'}) - \frac{q_{\mathrm{in}}}{T_{\mathrm{Source}}}\right] \tag{30}$$

The exergy destruction during expansion process $e_{\mathrm{D,Expansion}}$

$$e_{\mathrm{D,Expansion}} = T_0\left[(s_4 - s_3) + \frac{q_{\mathrm{surrounding}}}{T_0}\right] \tag{31}$$

Exergy destruction during exhaust

$$e_{\mathrm{D,Exhaust\ \ Blowdown}} = T_0\left[(s_{5'} - s_{4'}) + \frac{q_{\mathrm{surrounding}}}{T_0}\right] \tag{32}$$

The heat lost to the surrounding during exhaust and blow down is calculated by assuming ~9% heat lost during these processes [15].

Chemical exergy in HCCI engine is calculated as [17]

$$e^{\mathrm{ch}}_{\mathrm{HCCI}} = -\Delta g + \bar{R}T_0\left\{xO_2\ln\frac{P^{00}_{O_2}}{P^0} + yH_2O\ln\frac{P^{00}_{H_2O}}{P^0} - \sum_K x_k\ln\frac{p^{00}_k}{p^0}\right\} \tag{33}$$

where $\dot{m}$ the mass flow rate, s is the specific entropy at condition specified for the species and the subscript 0 represents environment conditions. x and y are is the molar fraction of species in the flow and e^{ch} is the reference specific chemical exergy. ΔG is the net change in the Gibbs energy in the direction of the reaction. For the calculation of chemical exergy, the standard partial pressure (bar) of different species in the environment is as: $(N_2) = 0.7583$, $(O_2) = 0.2040$, H_2O (g) = 0.0088, $(CO_2) = 0.000294$, $(Ar) = 0.00907$ [17].

In this study, the air fuel mixture is stoichiometric because water mixed with the ethanol provides all the needed dilution. The combustion equation for combustion of gases in the HCCI engine is assumed as follows.

$$\begin{aligned} C_2H_5OH + 6H_2O + 3(O_2 + 3.773N_2) &= 1.8CO_2 + 0.1CO \\ &\quad + 0.05C_2H_5OH + 8.85H_2O + 0.2O_2 + 11.319N_2 \end{aligned} \tag{34}$$

The procedure for obtaining the balanced reaction equation of an actual reaction where combustion is incomplete is not always straight forward. Actually combustion is the result of a series of very complicated and rapid chemical reactions and the product formed depend on many factors. When fuel is burned in the cylinder of an engine, the product of reaction vary with the temperature and pressure in the cylinder. The appearance of some carbon monoxide and unburned oxygen and fuel is due to incomplete mixing, insufficient time for complete combustion, and other factors. $e_{\mathrm{D,HCCI}}$ is calculated by assuming chemical reaction with 94% combustion efficiency.

The exergy balance for the HCCI engine gives the exergy destruction

$$\begin{aligned} e_{\mathrm{D,HCCI}} &= e_{\mathrm{D,Compression}} + e_{\mathrm{D,Heat\ addition}} + e_{\mathrm{D,Expansion}} \\ &\quad + e_{\mathrm{D,Exhaust\ \&\ Blowdown}} + e^{\mathrm{ch}}_{\mathrm{HCCI}} \end{aligned} \tag{35}$$

3.5. Catalytic converter

Catalytic converter is built in honey comb or pallet geometry to expose the exhaust gases to a larger surface. The HCCI produces very little NO_X so the oxidizing catalyst is required to control unburned hydrocarbons and carbon monoxide emission. Pd-loaded $SiO_2{-}Al_2O_3$ can burn the hydrocarbon and carbon monoxide that are not burned in the engine. The composition and temperature of outlet gases are calculated from the mass and enthalpy balances assuming complete combustion of hydrocarbon and carbon monoxide.

$$\sum_{\mathrm{Product}} n_{\mathrm{e}}\left(\bar{h}^{\circ}_{\mathrm{f}} + \Delta\bar{h}\right)_{\mathrm{e}} = \sum_{\mathrm{Reactant}} n_{\mathrm{i}}\left(\bar{h}^{\circ}_{\mathrm{f}} + \Delta\bar{h}\right)_{\mathrm{e}} \tag{36}$$

where i denotes the incoming air and fuel stream entering the catalytic converter and e denotes the exiting combustion products from the catalytic converter.[12]. The coefficients n_{i} and n_{e} correspond to the respective coefficients of the reaction equation giving the amount ratio of reactants or products per mole of ethanol. The assumed chemical equation of the catalytic converter is as follows.

$$\begin{aligned} 1.8CO_2 + 0.1CO + 0.05C_2H_5OH + 8.85H_2O + 0.2O_2 + 11.319N_2 \\ = 2CO_2 + 9H_2O + 11.319N_2 \end{aligned} \tag{37}$$

The exergy balance for the catalytic converter gives the exergy destruction as

$$e_{\mathrm{D,Converter}} = e^{\mathrm{ph}}_6 - e^{\mathrm{ph}}_7 + e^{\mathrm{ch}}_{\mathrm{converter}} \tag{38}$$

where $e^{\mathrm{ch}}_{\mathrm{converter}}$ is obtained after using Eq. (33) on the assumed catalytic converter chemical equation.

3.6. Turbine

Exhaust leaving the engine is further expanded through a turbine that drives a compressor. The benefits are two fold. (1) The engine is more efficient because energy that would have other wise been wasted is recovered from exhaust gas; and (2) the density of the incoming charge is greater.

The temperature at the exit of the turbine can be calculated by [7]

$$T_8 = T_7\left(\frac{P_8}{P_7}\right)^{\frac{(n-1)\eta_{\mathrm{T}}}{n}} \tag{39}$$

The energy balance yields the turbine work w_{T} and the exergy balance for the turbine gives the exergy destruction and is given by

$$e_{\mathrm{D,Turbine}} = \left(e^{\mathrm{ph}}_7 - e^{\mathrm{ph}}_8\right) - w_{\mathrm{T}} \tag{40}$$

$$w_{\mathrm{T}} = h_7 - h_8 \tag{41}$$

4. Performance parameters

The relevant parameters required for the wet ethanol operated HCCI engine may be considered as follows.

4.1. First law efficiency (η_I)

The ratio of all the useful energy extracted from the system to the energy of the fuel input is known as first law efficiency. By definition,

$$\eta_{\mathrm{I}} = \frac{w_{\mathrm{net}}}{q_{\mathrm{f}}} \tag{42}$$

4.2. Second law efficiency (η_{II})

Since exergy is more valuable than energy according to second law of thermodynamics, it is useful to consider both output and input in terms of exergy [13]. The amount of exergy supplied in the product to the amount of exergy associated with the fuel is more accurate measure of the thermodynamic performance of a system which is defined as the ratio of exergy contained in the product to the exergy associated with the fuel output.

$$\eta_{\mathrm{II}} = \frac{w_{\mathrm{net}}}{e_{\mathrm{f}}} \tag{43}$$

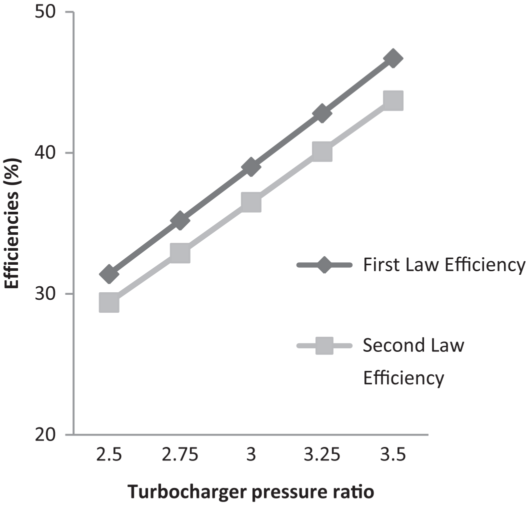

Fig. 3. Effect of variation of turbocharger pressure ratio on wet ethanol operated HCCI engine efficiencies.

5. Results and discussions

In this paper, the effect of turbocharger compressor pressure ratio, ambient temperature, and compressor polytropic efficiency on the first law efficiency of wet ethanol operated HCCI engine is obtained by energy balance approach or first law analysis of the cycle. However, the exergy destruction or irreversibility in each component and the second law efficiency has also been investigated under the exergy balance approach or second law analysis of the cycle. The exergy destruction in each of the component or the system as a whole is non-dimensional by expressing it as a percentage of the fuel exergy.

Fig. 3 shows the variation of turbocharger compressor pressure ratio on the first and second law efficiencies of wet ethanol operated HCCI engine for mean operating conditions of $T_0 = 300$ K, $\eta'_c = 0.8$. It is observed that as pressure ratio increases, the first and second law efficiencies of the system increase significantly. This is because increase in pressure ratio increases the density of the charge (wet ethanol and air) at the inlet of HCCI engine which increases the power output and hence the efficiency. Increase in pressure ratio was considered because HCCI engine requires high intake temperature for multi point ignition to occur. Fig. 3 also shows the variation of second law efficiency which is a more accurate measure of thermodynamic performance. Since the quality of fuel (i.e. exergy associated with the heat addition) is more than the heating value or energy of fuel because the exergy of fuel would increase while bringing it from the ambient pressure to combustion pressure at ambient temperature. Hence exergy associated with the heat addition will be equal to exergy associated with the heating value of fuel plus exergy increase, i.e., mechanical exergy due to increase of pressure of fuel from the ambient to combustion state. Therefore the second law efficiency of the cycle is slightly lower than the first law efficiency.

Fig. 4 shows the variation of first and second law efficiencies with the change in turbocharger compressor polytropic efficiency for $T_0 = 300$ K and $r_p = 3$. It is observed that for a given ambient temperature and pressure ratio, the first and second law efficiencies increase with the higher polytropic efficiency. This is because the compressor power reduces with the increase in polytropic efficiency that decreases the turbine power which results in reduction of exhaust temperature and hence more power delivered by the HCCI engine. This leads to increase in first law efficiency. First law efficiency of wet ethanol operated HCCI engine increases to its maximum value at polytropic efficiency (=0.9), Further increase in

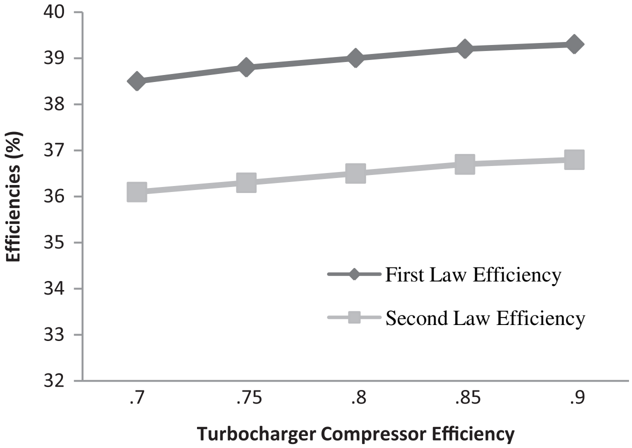

Fig. 4. Effect of variation of turbocharger efficiency on wet ethanol operated HCCI engine efficiencies.

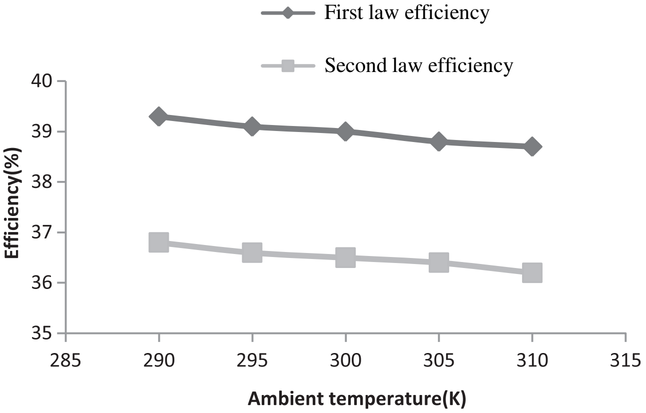

Fig. 5. Effect of variation of ambient temperature on wet ethanol operated HCCI engine efficiencies.

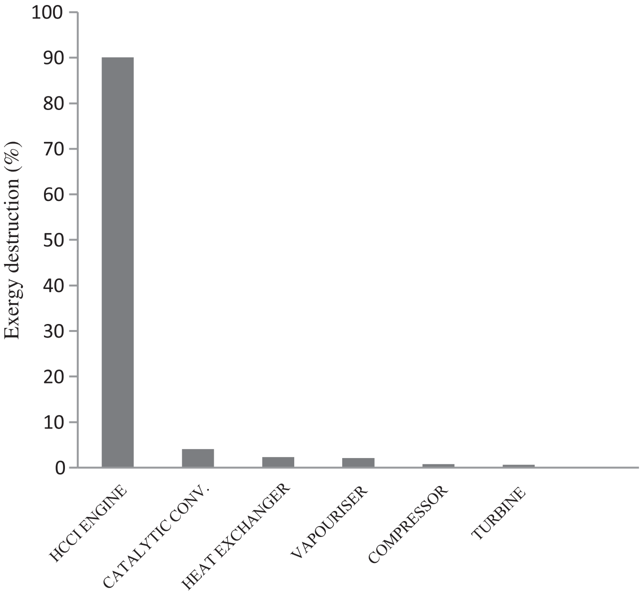

Fig. 6. Percentage of exergy destruction in each component of wet ethanol operated HCCI engine.

Table 2
Effect of variation of turbocharger compressor pressure ratio on exergy destruction in different components of ethanol operated HCCI engine. ($T_0 = 300$ K, $\eta'_c = 0.8$).

Turbocharger compressor pressure ratio, r_p	Compressor, Ed kJ/kg	Heat exchanger, Ed kJ/kg	Vaporizer, Ed kJ/kg	HCCI engine, Ed kJ/kg	Catalytic converter, Ed kJ/kg	Turbine, Ed kJ/kg
2.50	20.24	89.50	76.70	2814.00	128.14	14.65
2.75	22.45	80.25	70.55	2815.88	127.81	17.53
3.00	24.51	72.41	65.00	2817.59	127.67	20.24
3.25	26.43	65.58	60.10	2819.07	127.27	22.80
3.50	28.23	59.11	55.60	2820.43	127.06	25.23

Table 3
Effect of variation of ambient temperature on exergy destruction in different components of ethanol operated HCCI engine ($\eta'_c = 0.8$, $r_p = 3$).

Ambient temperature, T_0	Compressor, Ed kJ/kg	Heat exchanger, Ed kJ/kg	Vaporizer, Ed kJ/kg	HCCI engine, Ed kJ/kg	Catalytic converter, Ed kJ/kg	Turbine, Ed kJ/kg
290.00	23.54	75.73	67.02	2815.00	126.85	19.72
295.00	24.02	74.04	66.01	2816.32	127.34	19.98
300.00	24.51	72.41	65.00	2817.59	127.67	20.24
305.00	25.00	70.53	64.03	2818.82	127.80	20.49
310.00	25.50	68.96	62.98	2820.06	128.04	20.74

polytropic efficiency shows the flattened behavior for both first and second law efficiencies. Further increase in polytropic efficiency reduces the exhaust temperature below the required temperature for catalytic converter which reduces the converter efficiency and hence increases the emission. Second law efficiency is coming less than the first law efficiency due to the reason explained in Fig. 3.

Fig. 5 shows the variation of first law efficiency and second law efficiency with the change in ambient temperature for $r_p = 3$ and $\eta'_c = 0.8$. The increase in ambient temperature results in decreasing the first and second law efficiencies. This is because increase in ambient temperature increases the fuel air mixture intake temperature which results in reduced work output and hence the efficiency. It may be further verified from the fact that the increase in ambient temperature causes an adverse effect on the efficiency of the engine operating between two temperature limits.

Fig. 6 shows the fraction of exergy destruction in each components of wet ethanol operated HCCI engine for $T_0 = 300$ K $r_p = 3$, $\eta'_c = 0.8$ The exergy study reveals that the majority of system irreversibility occurs within the HCCI engine components where 90.09% of the total exergy destruction in the overall system is destroyed as shown in Fig. 6. The exergy destroyed in the remainder of the system; in the catalytic converter is 4.08% of the total exergy destruction. ≈Around 4.39% of the total exergy is destroyed by the processes of heat transfer in fuel vaporizer and heat exchanger. ≈About 1.43% of the total exergy is destroyed by the compressor and turbine. These results clearly show that exergy analysis is providing a ranking among the components of the system. Component with higher exergy destruction is more responsible to deteriorate the performance compared to the component of the system with the lower exergy destruction, and shows which component needs to be repaired or serviced first.

Table 2 shows the variation of the magnitude of exergy destruction of each component of the system when changing the turbocharger compression ratio for $T_0 = 300$ K, $\eta'_c = 0.8$. It is found that the exergy destruction in HCCI engine dominates the exergy destruction picture because of the irreversibility due to combustion process. The exergy destruction in catalytic converter comes next. As the pressure ratio increases, the exergy destruction in heat exchanger decreases significantly. This is because the higher pressure ratio results in the lower turbine exit temperature. The exergy destruction in compressor and turbine increases with the increase in pressure ratio. The exergy destruction in fuel vaporizer decreases significantly with the increase in pressure ratio. This is because entropy generation due to mixing of wet ethanol and air in fuel vaporizer is lower at higher pressure ratio.

Table 3 shows the variation of magnitude of exergy destruction in each component of wet ethanol operated HCCI engine with the increase in ambient temperature for $r_p = 3$, $\eta'_c = 0.8$. As the ambient temperature increases, the exergy destruction in the compressor increases. The exergy destruction in catalytic converter and turbine also increases slightly with the increase in ambient temperature. The exergy destruction in vaporizer and heat exchanger decreases significantly with the increase in ambient temperature because the temperature difference between the two heat exchanging fluid, i.e. the compressed air and wet ethanol decreases which results in smaller finite time heat transfer irreversibility and hence the exergy destruction. Similarly the maximum possible temperature difference in the heat exchanger decreases with increase in ambient temperature which causes a decrease in exergy destruction.

Table 4 shows the variation of magnitude of exergy destruction in each component of the system with the change in turbocharger compressor efficiency for $r_p = 3$, $T_0 = 300$ K. It is shown that the exergy destruction in turbocharger compressor decreases significantly with the increase in turbocharger efficiency due to smaller temperature difference across the compressor which results in lower heat transfer irreversibility and hence the exergy destruction. As the turbocharger efficiency increases, the exergy destruction in the heat exchanger increases because the temperature gradient of heat transfer in it increases with higher compression efficiency. The

Table 4
Effect of variation of turbocharger compressor efficiency on exergy destruction in different components of ethanol operated HCCI engine ($T_o = 300$ K, $r_p = 3$).

Turbocharger compressor efficiency, $\eta'_{C\ \text{variation}}$	Compressor, Ed kJ/kg	Heat exchanger, Ed kJ/kg	Vaporizer, Ed kJ/kg	HCCI engine, Ed kJ/kg	Catalytic converter, Ed kJ/kg	Turbine, Ed kJ/kg
0.70	41.81	67.80	67.15	2816.31	127.80	20.13
0.75	32.57	70.03	66.02	2816.99	127.65	20.19
0.80	24.51	72.41	65.00	2817.59	127.67	20.24
0.85	17.42	74.16	64.22	2818.06	127.40	20.28
0.90	11.12	75.86	63.49	2818.50	127.30	20.32

exergy destruction in fuel vaporizer decreases with increase in turbocharger efficiency. This is because the increase in turbocharger efficiency causes lower compressor exit temperature and hence the lower temperature gradient for heat transfers in the vaporizer which results in smaller exergy destruction. The exergy destruction in HCCI engine and catalytic converter are almost constant because the effect of chemical exergy in these components dominates over the effect of physical exergy.

6. Conclusion

Exergy analysis which is widely gaining acceptance over traditional energy methods in both industry and academia because it clearly tells about the efficient utilization of resources (fuels) and explains the benefits of sustainable energy and technology. This method of exergy analysis has been applied to wet ethanol operated HCCI engine and its application leads to the following conclusions.

- The first and second law efficiencies are found to be an increasing function of the turbocharger pressure ratio while they are found to be a decreasing function of the ambient temperature.
- The effect of turbocharger pressure ratio is found to be more than that of the compressor efficiency and ambient temperature on exergy destruction in each component of the cycle.
- Exergy destruction in HCCI engine, catalytic converter and turbine changes very little as the turbocharger compressor efficiency increases.
- The highest irreversibility is in the HCCI engine (90.09% of total irreversibility).
- Heat transfer processes in fuel vaporizer and heat exchanger, and catalytic converter accounts for about 4.39% and 4.08% of the total exergy destruction, respectively.

Inclusion of losses in an irreversibility analysis approaches the performance of real cycle. The present study is expected to help researchers and engineers for a better design and analysis of ethanol fuelled HCCI engine.

Nomenclature

C	carbon atom
c_v	specific heat at constant volume (kJ/kg K)
c_p	specific heat at constant volume (kJ/kg K)
E	exergy (kJ)
E_D	exergy destruction (kJ)
e	specific exergy (kJ/kg)
e^{ph}	physical exergy (kJ/kg)
e^{ch}	chemical exergy(kJ/kg)
e_D	exergy destruction(kJ/kg)
f	residual gas fraction
ΔG	change in Gibb's function for reaction (kJ/kg)
H_i	lower specific heating value
H	enthalpy (kJ)
H	hydrogen
ΔH	change in molar enthalpy (J/mol)
h	specific enthalpy (kJ/kg)
$\bar{h}_f$	enthalpy of formation on molar basis (kJ/kg)
Δh	change in specific enthalpy (J/kg)
M	molar mass (kg/mol)
m	mass (kg)
N	nitrogen
n	polytropic index
O	oxygen
P	pressure (bar)
r_c	compression ratio
r_p	pressure ratio
P_B^{00}	partial pressure of component B (bar)
q_{in}	heat supplied during heat addition process of HCCI engine (kJ/kg)
$q_{surrounding}$	heat rejected during exhaust & blow down process of HCCI engine (kJ/kg)
q_f	heat supplied by fuel (kJ/kg)
R	molar gas constant (J/K mol)
r_c	compression ratio
S	entropy (kJ/K)
s	specific entropy (kJ/kg K)
T	absolute temperature (K)
T_0	ambient temperature (K)
t	temperature (°C)
V	volume (m^3)
v	specific volume (m^3/kg)
w	work (kJ/kg)

Subscripts

C	compressor
c	compression
ch	chemical
D	destruction
e	expansion
e	exhaust
f	fuel
FV	fuel vaporizer
g	gas
HE	heat exchanger (regenerator)
i	inlet
k	components of products of combustion
ph	physical
Q	heat
T	turbine
W	work
x, y	reactant coefficients
I	first law
II	second law
1–9	state points.
1′, 2′, 3′, 4′, 5′	state points of HCCI cycle.

Greek symbol

η	efficiency (%)
γ	ratio of specific heat capacities
ϵ_h	effectiveness of regenerator

References

[1] W. Korbitz, Biodiesel production in Europe and North America, an encouraging prospect, Renew. Energy (1999) 1078–1083.

[2] D. Pimentel, The Limit of Biomass Energy, Encyclopedia of Physical Science and Technology, third ed., vol. 2, Academic, New York, 2001, p. 159–171.

[3] H. Shapouri, J.A. Duffield, M.S. Graboski, Estimating the Net Energy Balance of Corn Ethanol Report No. AER-721. USDA Economic Research Service, Washington D.C., 1995.

[4] H. Shapouri, J.A. Duffield, M. Wang, The exergy balance of corn ethanol revisited, Trans. ASME 46 (4) (2003) 959–968.

[5] P.F. Flynn, G.L. Hunter, R.P. Durrett, L.A. Farrell, W.C. Akinyemi, Minimum engine flame temperature impact on diesel and spark ignition engine NO_X production, SAE Paper No. 2000-01-1177 (2000).

[6] K. Epping, S.M. Aceves, R.L. Bechtold, J.E. Dec, The potential of HCCI combustion for high efficiency and low emission, SAE Paper No. 2002-01-1923 (2002).

[7] J.M. Frias, S.M. Aceves, D.L. Flower, Improving ethanol life cycle energy efficiency by direct utilization of wet ethanol in HCCI engine, Trans. ASME, J. Energy Res. Technol. 129 (2007) 332–337.

[8] J.E. Dec, M. Sjoberg, A parametric study of HCCI combustion – the sources of emissions at low loads and the effects of GDI fuel injection, SAE Paper No. 2003-01-0752 (2003).
[9] S.D. Oh, H.S. Pang, S.M. Kim, H.Y. Kwak, Exergy analysis for a gas turbine cogeneration system, ASME J. Eng. Gas Turbine Power 118 (1996) 782–791.
[10] A.A. Hasan, D.Y. Goswami, S. Vijayaraghavan, First and second law analysis of a new power and refrigeration thermodynamic cycle using a solar heat source, Solar Energy 73 (5) (2002) 385–393.
[11] R.J. Osborne, G. Li, S.M. Sapsford, J. Stokes, T.H. Lake, Evaluation of HCCI for future gasoline power trains, SAE Paper No. 2003-01-0750 (2003).
[12] M.J. Moran, H.N. Shapiro, Fundamentals of Engineering Thermodynamics, sixth ed. Wiley, New York, 2008.
[13] A. Bejan, Fundamentals of exergy analysis, entropy generation minimization, and the generation of flow architecture, Int. J. Energy Res. 26 (2002) 545–565.
[14] D.W. Green, R.H. Perry, Perry's Chemical Engineer's Hand book, eighth ed. Mcgraw Hill, New York, 2008.
[15] J.B. Heywood, Internal Combustion Engine Fundamentals. McGraw Hill, New York, 1988.
[16] C.R. Ferguson, A.T. Kirkpatrick, Internal Combustion Engines, second ed. Wiley, New York, 2004.
[17] T.J. Kotas, The Exergy Method of Thermal Plant Analysis. Butterworth, London, 1985.