Renewable and Sustainable Energy Reviews 51 (2015) 382–395

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Power generation from waste of IC engines

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ARTICLE INFO

Article history:
Received 26 August 2014
Received in revised form
8 April 2015
Accepted 26 May 2015

Keywords:
Renewing waste energy system
SI engine
Micro-facet exhaust gas recirculation
Thermal electric generator
Emission
Fuzzy intelligent system

ABSTRACT

Fuel consumption of IC engine could be improved significantly by harvesting waste thermal energy. Several methods for waste thermal energy recovery from internal combustion engine (ICE) have been studied by using supercharger or turbocharger and /or combined. This study presents an innovative approach on power generation from waste of IC engine based on coolant and exhaust. The waste energy harvesting system of coolant (weHS^{c}) is used to supply hot air at temperatures in the range of 60–70 °C directly into the engine cylinder, which would be useful to vaporize the fuel into the cylinder. The waste energy harvesting system of exhaust system (weHS^{ex}) has been developed with integrating fuzzy intelligent controlled Micro-Faucet emission gas recirculation (MiF-EGR) and thermoelectric generator (TEG). In this study the MiF-EGR (micro-facet exhaust gas recirculation) will be used to maintain the intake temperature 70 °C by keeping flow of the exhaust to the engine cylinder chamber and to increase the engine volumetric efficiency. The TEG produces electrical power from heat flow across a temperature gradient of exhaust and delivers DC electrical power to the vehicle electrical system which could reduce the load of the alternator by as much as 10%. The performance of weHS equipped engine has been investigated by using GT suite software for optimum engine speed of 4000 rpm. The result shows that specific fuel consumption of engine has improved by 3% due to reduction of HC formation into the engine combustion chamber causes significantly improved the emission. While, the brake power has been increased by 7% due to the fuel atomization and vaporization at engine intake temperature 70 °C.

Contents

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http://dx.doi.org/10.1016/j.rser.2015.05.077
1364-0321/

1. Introduction

Thermal management of IC engines is considered a serious problem as it develops high temperatures due to combustion of the fuel but needs to keep the temperature at a controllable level in order to operate the engine safely. Once the temperature in the combustion chamber has reached intolerable values the engine block and components may suffer damage. Therefore, it is essential to have a heat removal process which will maintain the engine at a safe operating condition. Heat losses to the atmosphere through the exhaust are unavoidable. The heat lost to the exhaust is determined by the temperature within the cylinder when exhaust begins. The large amount of energy from the stream of exhaust gases could potentially be used for waste heat recovery to increase the work output of the engine [4]. Hatazawa et al. [5] and [7], Stabler [6], Yang [9], Yu and Chau [8] stated that the waste heat produced from thermal combustion process generated gasoline could get as high as 30–40% is lost to the environment through exhaust pipe. Sam [23] reported that the engine energy distribution: shaft power 25–40%; heat rejection- coolant heat rejection 10–35%, exhaust enthalpy loss 20–45%, and engine external surface loss 2–10%. Conklin and Szybist [10] investigated that the percentage of fuel energy converted into useful work only 10.4% and also found that 27.7% energy lost through exhaust pipe. Dolz et al. [11] reported that the value of exhaust gases is 18.6% of total combustion energy. [12] stated that by installing heat exchanger exhaust energy can be saved up to 34%. Based on the above researches output it could be concluded that the engine waste energy can be turned into useful energy by using several techniques which are discussed in the following sections of this manuscript.

An accurate estimate of the heat transfer between cylinder gases and cylinder wall of a combustion engine is necessary for a precise calculation of power, efficiency and emissions during engine development [1]. Several models exist for evaluating the heat transfer coefficient, of which the correlations of [2] and [3] are the most widely used. The heat transfer involved in the intake system occurs when air or an air–fuel mixture enter into the manifold. The intake manifold is hotter than the air–fuel mixture because of its proximity to the engine components or the design of the manifold. The intake manifold can be designed to heat the air–fuel mixture so that the mixture can start to vaporize once it has entered the combustion chamber. One way of heating the manifold is to put it in close proximity with other hot components. The manifold will heat through convective heat transfer. Electricity and hot coolant flow are other ways in which the manifold can also be heated. After the manifold is heated the air–fuel mixture receives heat through convective heat transfer process [18]. Gu et al. [15] found that the engine cycle (Rankine Cycle) efficiency of several working fluids is very sensitive of evaporative pressure but insensitive to expander inlet temperature. Boretti [16] stated that in a given temperature gradient for optimizing the work output, the working fluid's evaporation enthalpy should be as high possible. Engine intake air temperature needs to increase to 60 °C by using hot spots at the intake manifold for 60% fuel vaporization, which helps the engine to do the complete burning of fresh air fuel mixture Willard [32]. Willard [32] has reported that in order to get the proper combustion it is necessary to make the fuel after injection: *atomization: the fuel droplets break into smaller droplets*; *vaporization: the small droplets of fuel vaporize in the chamber due to high temperatures.* About 90% of the fuel injected into the cylinder needs to be vaporized within 0.001 s after injection.

The maximum power, an engine can deliver, is limited by the amount of fuel that can be burned efficiently inside the engine cylinder . This in turn is limited by the amount of air that is introduced into each cylinder in each cycle [1]. So by introducing a greater amount of air into the cylinder, the opportunity for combustion increases, leading to increase in power of the engine. Different types of waste energy from exhaust can be captured using different energy harvesting materials. The most promising technologies in development include heat that can be captured and transformed into electrical power using thermoelectric and piezoelectric materials. Thermoelectric materials can capture some of this heat, and produce electricity. Thermocouple was first used by [20], to measure gun bore temperatures. Stobart et al. [13] explored the possibility of thermoelectric generator (TEG) in vehicles in which they found that the 1.3 kW output of thermoelectric device could potentially replace the alternator of small vehicle. By improving thermocouples, it would be possible to convert 3 to 5% of the waste heat into electricity which would be efficient enough to recharge a vehicle's 12 V battery. Therefore, the load on the engine is reduced thereby improving fuel efficiency by as much as 10%. However, a 10% efficient thermal electric generator can require at least 500 °C [21]. An increase of 20% of fuel efficiency can be easily achieved by converting about 10% of the engine waste heat into electricity [9,14]. TEG could be coupled with various other devices to maximize its potentiality. Yu and Chau [8] has proposed and implemented an automotive thermoelectric waste heat recovery system by adopting a Cuk converter and a maximum power point tracker (MPPT) controller as a tools for power conditioning and transfer.

2. Materials and methodology

This section has presented the engine heat transfer to the cylinder wall and variation of cylinder wall temperature with crank angle, the effect of heat transfer on the volumetric efficiency and emission such as CO, HC and NO_x. This section also presents the theoretical models of weHS[c] and the role of MiF-EGR on maintaining the emission and influences the performance of weHS[c] on engine power enhancement and finned type TEG performance on the conversation of waste heat energy of engine into electrical energy.

2.1. Thermodynamic analysis of internal combustion SI engine

Energy is supplied to the engine in the form of chemical energy of the fuel and producing useful power and losses as heat through exhaust and coolant. First Law of thermodynamics states that energy is conserved. Therefore, the energy balance equation for the engine can be represented as,

$$\dot{m}_f h_f + \dot{m}_a h_a = P_{brake} + \dot{Q}_c + \dot{Q}_{exh} + (\dot{m}_a + \dot{m}_f)\ h_e + \dot{Q}_{rad} \tag{1}$$

with

$$\dot{Q}_C = \dot{m}_c c_{pc} \Delta T \text{ and } \dot{Q}_{exh} = \dot{m}_{exh} c_{exh} \Delta T$$

where, $\dot{m}_a$ and $\dot{m}_f$ is the rate of flow of air and fuel to the engine as initial energy, $\dot{Q}_c$ and $\dot{Q}_{exh}$ is the heat transfer rate of the engine to the coolant and exhaust respectively, h_e is the enthalpy of unburned gas mixture.

$$p_{brake} = \dot{m}_f h_f + \dot{m}_a h_a - \dot{Q}_c - \dot{Q}_{exh} - (\dot{m}_a + \dot{m}_f)\ h_e - \dot{Q}_{rad} - \dot{Q}_{fric} \tag{2}$$

Eq. (2) indicates that the P_{brake} will be increased if % of $\dot{Q}_c$ and $\dot{Q}_{exh}$ can be recovered and can be reduced the unburned gas mixture of $(\dot{m}_a + \dot{m}_f)\ h_e$. Recover of waste heat energy can increase the brake power, improve the fuel consumption and reduce the CO_2 emission. This study presents waste energy harvesting system (weHS) for coolant and exhaust which could be potentially used to harvest % of waste energy. Heat transfer through a cylinder wall per unit surface area can be estimated by using the equation of Willard [32]:

$$\dot{q} = (T_g - T_c)/[(1/h_g) + (\Delta x/k) + (1/h_c)] \tag{3}$$

where, T_g is average temperature of burn charge in the combustion

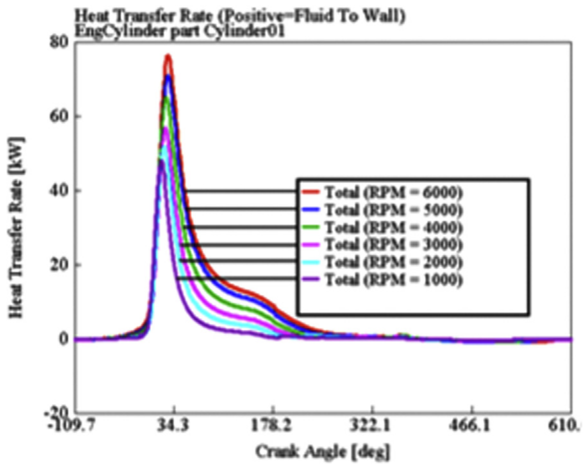


Fig. 1. Heat transfer with crank angle.

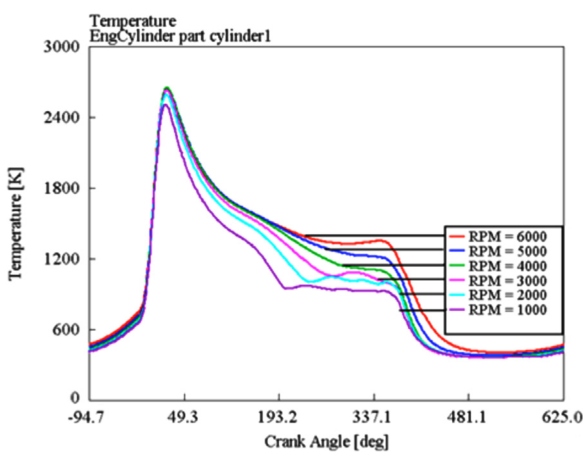


Fig. 2. Cylinder temperature at crank angle.

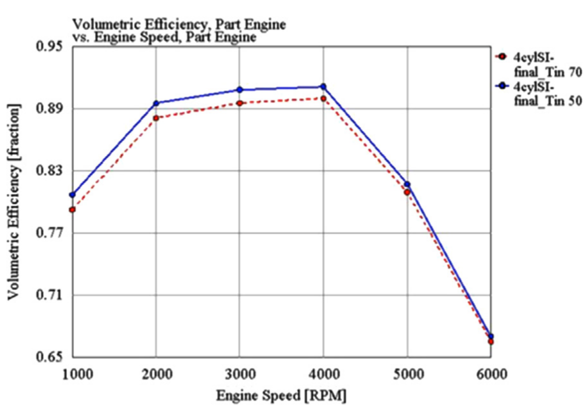


Fig. 3. Volumetric efficiency with engine speed.

chamber, T_c is the coolant temperature, h_g and h_c are the convection heat transfer coefficient gas in cylinder and coolant side, Δx is the combustion chamber wall thickness, and k is the thermal conductivity of the cylinder wall. Heat transfer coefficient can be computed by using the equation:

$$h_g = Nu\frac{k_g}{B} = C_1(Re)^{C_2}\frac{k_g}{B} \tag{4}$$

with

$$Re = \frac{[(\dot{m}_a + \dot{m}_f)B]}{A_p \mu_g}$$

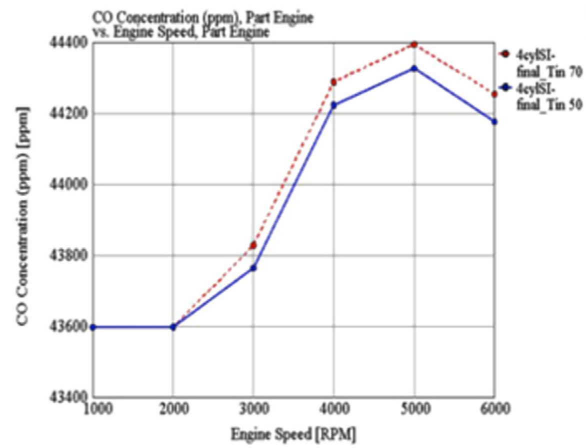


Fig. 4. CO emission with heat transfer.

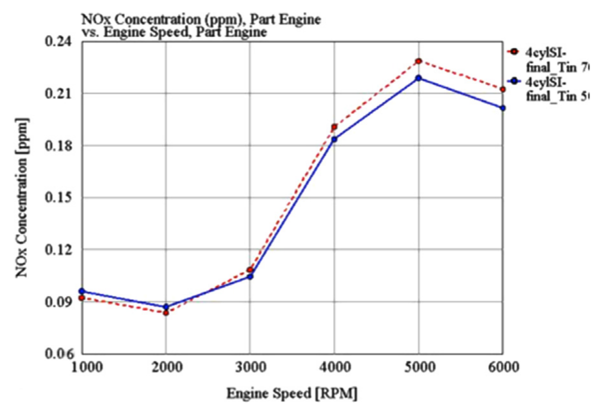


Fig. 5. NO*x* emission with heat transfer.

where, *Nu* is the Nusselt number, *Re* is the Reynold number, μ_g is the dynamic viscosity of gas in cylinder, k_g is the thermal conductivity of gases in cylinder. C_1 and C_2 are constant, B is the cylinder bore size in m, and A_p is the area of the piston face. When the air–fuel mixture is introduced into the cylinder chamber, complex motion such as tumbling and swirling flows are developed. Due to the unsteadiness and local changes caused turbulence for the Reynold numbers, tumbling, swirling motions and interactions with valve motion, engine heat transfer undergoes unsteadiness and local changes affecting volumetric efficiency and combustion.

2.2. Simulation of thermodynamic analysis—Using GT-suite software

The simulation study on the effect of engine speed (rpm) to the heat transfer in cylinder wall and the volumetric efficiency has been conducted by using GT-suite software. At higher engine speeds, there is less time per cycle. Combustion occurs over about the same engine rotation at all speeds, so the time of intake and combustion is less at higher speeds. The less time for intake and ignition and less time for heat transfer per cycle, causes the engine runs hotter as shown in Figs. 1 and 2. Therefore, the wall of the cylinder becomes hotter which might affect the fuel lost. Reitz [22] reported that up to 30% of fuel energy is lost to the wall heat transfer. Therefore, the exhaust temperature will be high and it could be in the range of 300–900 °C. By using a thermoelectric generator waste heat energy of exhaust can be harvested. Fig. 3 shows that the volumetric efficiency of the engine will be less if the engine heat transfer is less which might occur for higher speed (i.e., turbulence effect). The volumetric efficiency of the engine will be maximum at engine speed of 4000 rpm which might be good

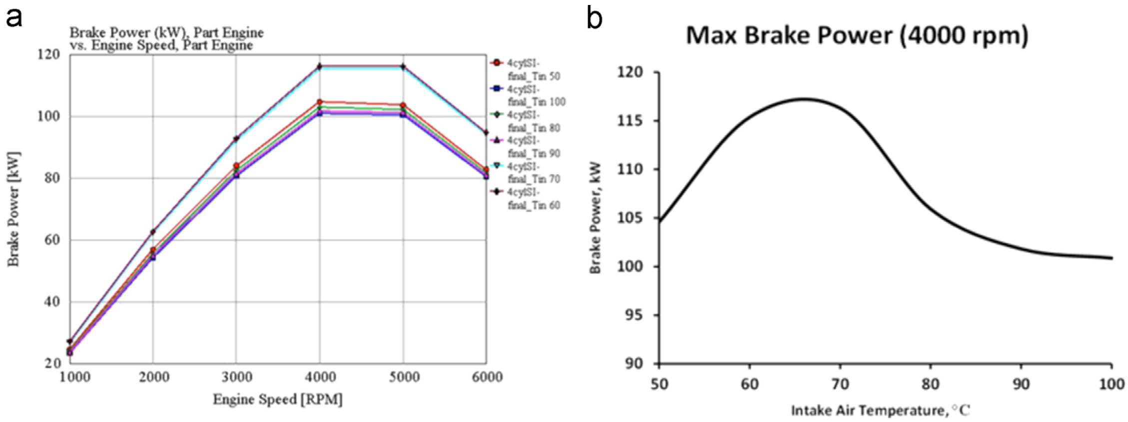


Fig. 6. Engine brake power at (a) different rpm, (b) different intake temperature.

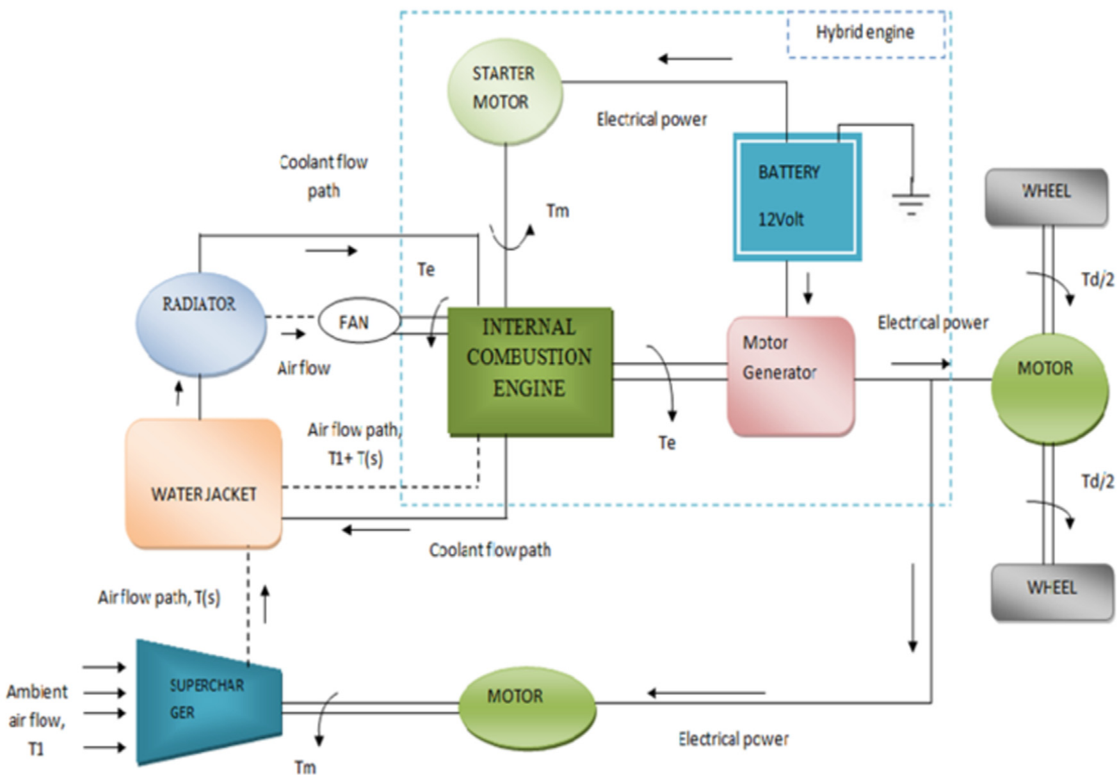


Fig. 7. weHS for coolant (*Source*: [17]).

for good combustion and there is less amount of unburnt HC that could remain in the cylinder chamber. Heat transfer has a strong influence on exhaust emissions [19]. Formation of NO_x has an exponential dependence on temperature. Figs. 5 and 6 show the emission of CO and HC with the heat transfer from engine. The engine emission control and waste energy recovery has been conducted by developing model of $weHS^c$ and $weHS^{ex}$. Figs. 4 and 5 shows the concentration of CO and NO_x.

The $weHS^c$ and $weHS^{ex}$ has presented in this study to improve the engine performance in the range of 10–15% with reducing the engine emission. The $weHS^{ex}$ is consisted of fuzzy adaptive control micro facet exhaust gas recirculation (MiF-EGR) and finned type thermal electric generator (TEG). The basic purpose of $weHS^c$ is to supply the sufficient air with temperature in the range of 60–70 °C into the engine cylinder chamber to vaporize the fuel and to avoid the formation of CO and HC into the residual gas, as suggested by [32,17]. The fuzzy adaptive control MiF-EGR is used to keep down the combustion temperature of the engine and to reduce the formation of NO_x. TEG has been used to convert the heat energy of exhaust gases into electrical energy due to the temperature gradient. The FAC activates the MiF-EGR if the intake air of $weHS^c$ is more than 70 °C. The effect of intake air temperature on engine brake power has been analyzed by using software GT Suite. The output power of the engine as shown in Fig. 6, varies with the variation of intake temperature from 50 to 100 °C. The maximum output brake power 117 kW at intake temperature (T_{sca}) 70 °C. The brake power of engine increases 11% for the variation of engine intake temperature from 50 to 70 °C.

3. Waste energy harvesting system

The main concern of the development of hybrid engine is to reduce the engine size and use 40–45% of energy generated due to combustion as useful energy. Energy harvesting system from the cooling system is one of the best methods to increase the engine performance by decreasing fuel consumption and emission. This is the section discusses the model of weHS^c and weHS^{ex} for hybrid engine. Model for weHS^c

The theoretical model of weHS^c as shown in Fig. 7, is developed with a coolant flow jacket and a metal pipe (duct). One end of the metal pipe placed inside the coolant jacket is connected with an electro small scale supercharger while the other end is connected with the intake manifold of the engine. Hot coolant from the engine flows through coolant jacket and releases heat to the super—charged air into the inner metal duct. Thus the supercharged air gains heat and enters the engine cylinder chamber with temperatures in the range of 60–70 °C, which could atomize and vaporize the fuel. Therefore, the complete combustion into the engine could occur and reduce the formation of CO and H_2 into the residual gas of the cylinder chamber significantly.

3.1. Mathematical model for weHS^c

The following assumptions are made for the dynamic modeling and simulation of weHS^c performance:

- Based on Baker [24].
- The physical properties of the coolant are constant over the range of temperatures employed.
- The heat losses to the surroundings are negligible and the two end plates of the weHS serve as adiabatic walls.
- The film coefficient for heat transfer is dependent principally upon the fluid velocity and is proportional to an exponential function of the flow rate.
- The heat transfer within the coolant in any channel is by convection only.
- The thermal capacity of the weHS^c wall is negligible compared with the thermal capacity of the coolant in the plate.
- The variation of temperature along the length of the duct is neglected.
- Based on Yunus and Michael [27].
- The temperature of the supercharged compressed air at the inlet of weHS^c is considered to be 59 °C for mass rate of air 0.178 kg/s and the specific heat of air at constant pressure 1.001 kJ/kg K.
- Based on Incropera [28], Yunus [31],
- the coolant-ethylene glycol (50% ethylene glycol and 50% water), the density of ethylene glycol 1109 kg/m^3, the specific heat of exhaust pressure 1.077 kJ/kg K and boiling point of coolant 111 °C.
- The coolant temperature at the inlet and outlet of the weHS^c are 89 °C and 70 °C, respectively.
- Based on Khan et al. [25]
- The coolant pump's mass flow rate of coolant 2.5 kg/s for the normal operating temperature of the engine.

The heat gaining by the coolant from the engine can be represented by using the equation of Rahman et al. [17],

$$Q = \frac{V\int_0^{P_{com}} dP + \gamma\, P\int_0^V dV}{(\gamma-1)} - PV\frac{PV}{(\gamma-1)^2}\int_0^{\gamma} d\gamma \tag{5}$$

where, γ is the ratio of specific heat was calculated from the composition and mean temperature of the in-cylinder gas and treated as a variable, P is the maximum engine combustion pressure in kPa, V is the displacement volume of the cylinder at the time of combustion completed. The specific heat ratio can be defined as, $\gamma = c_p/c_v$, where, c_p is the specific heat at constant pressure in J/(kg K) and c_v is the specific heat at constant volume in J/(kg K). The rate of heat (energy) released by the coolant to the inner duct of the weHS can be represented as,

$$\dot{Q}_{coolant} = \dot{m}_c C_p \left(T_{c(o)} - T_{c(i)}\right) \tag{6}$$

where, $T_{c(o)}$ and $T_{c(i)}$ are the coolant inlet and outlet temperature.

The nonlinear evolution equation for the coolant is estimated under the maximum heating condition. The length of the coolant flow into the weHS^c is very large with respect to the wave amplitude. For dynamic studies of the nonlinear waste energy recovery from the coolant, the flow rate of coolant is considered as an input variable while the inlet-temperature of supercharged air is consider as constant and vice-versa. Based on the advanced vehicle thermal management systems, the temperature of the coolant which taken away from the engine is set as 70 °C [26]. In weHS^c, the hot coolant releases heat to the cold supercharged air. The coolant temperature at the outlet point of the weHS^c is about 60 °C ([26]). Temperature of the supercharged air into the duct can be estimated as,

$$T_{sca} = \frac{h_c}{h_{sca}}\left(T_{c(i)} - T_{wc(i)}\right) + T_{sca(wall)} \quad \text{with } i = 1,\ 2,\ 3\ldots, n. \tag{7}$$

where, T_{sca} is the supercharged air, T_{wc} is the release temperature of the coolant to the weHS's inner wall duct. The T_{wc} is the function of mass rate flow of coolant and the length of the inner duct of the weHS^c, $T_{wc} = \oint(\dot{m}_c, L)$, if mass flow rate of supercharged air is kept constant. The released temperature of the coolant per unit length of the duct of weHS supercharged air can be modeled by simplifying the equation of Gerardo C. Diaz [29]:

$$T_{wc} = \left[T_{c(i)} - \frac{1}{\dot{m}_c C_p}\frac{\dot{Q}}{\left[\exp\left(-\frac{h_c dL}{\dot{m}_c c_p}\right) - 1\right]}\int_0^L dl\right] \tag{8}$$

while, the gaining temperature of the supercharger compressed air into the duct of weHS^c per unit length can be estimated by using the equation of Rahman et al. [17]:

$$\Delta T_{sca(duct)} = \left[T_{c(i)} - \frac{1}{\dot{m}_c c_p}\frac{\dot{Q}}{\left[\exp\left(-\frac{h_c dL}{\dot{m}_c c_{p(c)}}\right) - 1\right]}\int_L^0 dl\right] \tag{9}$$

where, $T_{c(i)}$ is the inlet temperature of the coolant, T_{wc} is the temperature release by the coolant to the inner duct of weHS, d is the thickness of the inner wall, L is the length of the duct, h_c is the heat transfer coefficient of the coolant and $T_{sca(duct)}$ is the supercharged air temperature into the duct. Temperature of the supercharged air at the outlet port of the weHS can be computed by using the equation of Rahman et al. [17]:

$$T_{sca(o)} = T_{sca(i)} + \Delta T_{sca(duct)} = T_{sca(i)} + \left[T_{c(i)} - \frac{1}{\dot{m}_c c_p}\frac{\dot{Q}}{\left[\exp\left(-\frac{h_c dL}{\dot{m}_c c_{p(c)}}\right) - 1\right]}\int_L^0 dl\right] \tag{10}$$

The weHSc supercharged air temperature is controlled in the range of 60–70 °C in the engine cylinder chamber. If the temperature goes above 70 °C, it needs to control by controlling the cooled EGR with response of a sensor. Temperature of the $T_{sca(o)}$ is the function of the coolant temperature which significantly increases the heat transfer to the duct and finally to the supercharged air ($T_{sca(o)}$). Therefore, the intake temperature will be higher. It is stated previously as the intake temperature reaches 100 °C or more, engine volumetric efficiency drastically discreases. So that the engine BMEP and efficiency will be lower. This study presents a fuzzy controlled EGR which will be used to maintain the temperature of intake in the range of 600–700 °C.

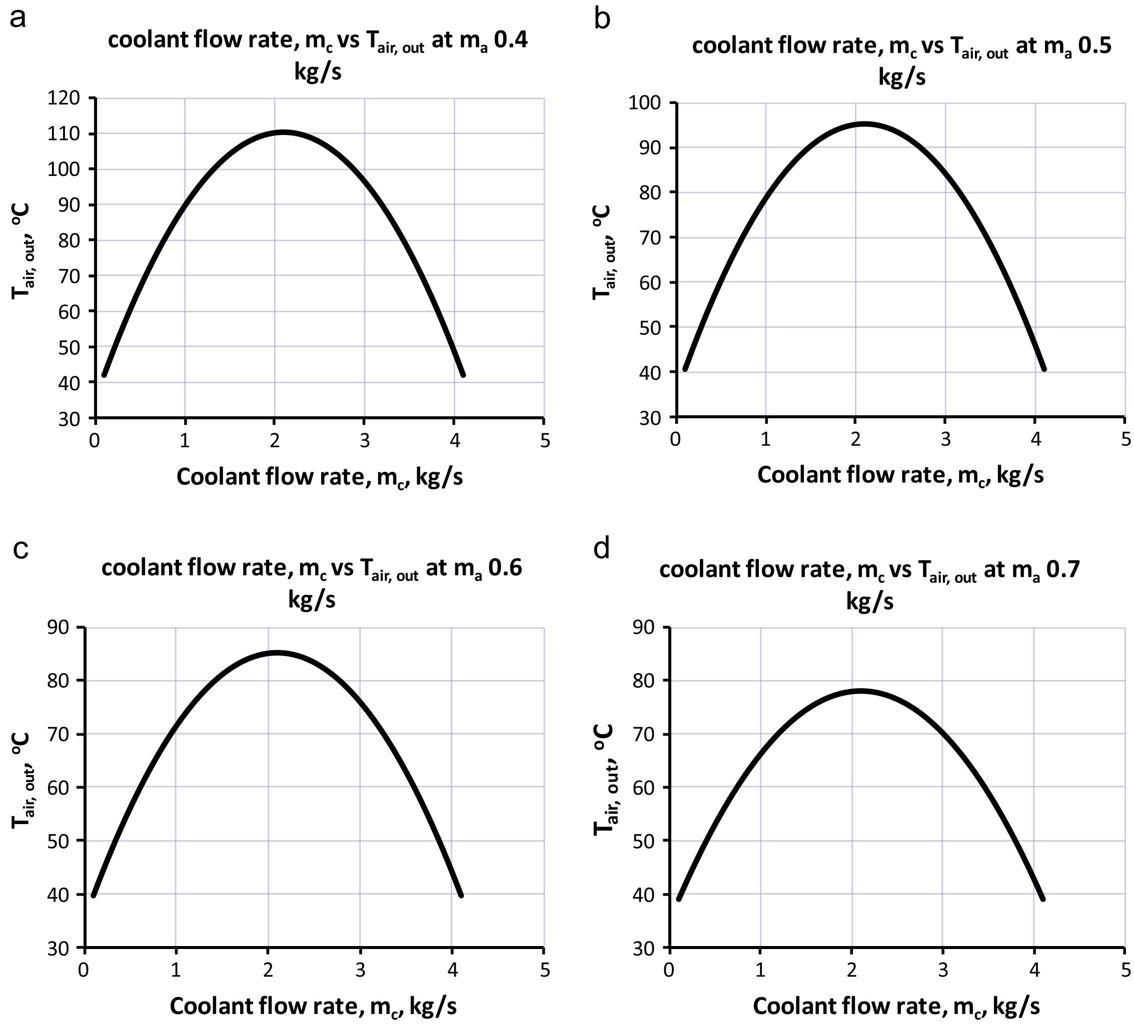


Fig. 8. Temperature of supercharged air for the variation of mass flow rate of coolant.

The work transfer rate or power required to drive the supercharger can be estimated,

$$\dot{W} = \frac{\dot{m}_{a(i)} c_p T_\alpha}{\eta_c}\left[\left(\frac{p_{sc(0)}}{p_\alpha}\right)^{(\gamma-1)/\gamma} - 1\right] \tag{11}$$

where, $\dot{m}_{a(i)}$ is the mass flow rate of air to the supercharger, T_α is the surrounding atmospheric temperature, $P\alpha$ is the atmospheric pressure of air, $P_{sc(o)}$ is the outlet pressure of the supercharged air, η_c supercharger efficiency.

3.2. *Performance of WeHS^c—Theoretically*

The supercharged air temperature before the inlet port of weHS is 35 °C if the supercharged air gains 100% of the heats that rises across the supercharger. However, in practice it is not. Theoretical model of waste energy harvesting system coolant based has been developed to enhance the engine performance and preventing the engine from detonation. For the optimum temperature of the supercharged air at the outlet port of the weHS^c is considered to be 70 °C. The mass flow rate of coolant has been investigated theoretically for the different mass flow rate of supercharged air and vice-versa. Figs. 8 and 9 shows the value of the output temperature of the weHS^c which is called temperature of supercharged air or engine intake temperature ($T_{sca(o)}$ or T_{in}) and vice-versa. Fig. 8 shows that the coolant flow rate needs to maintain in the range of 3 to 3.7 kg/s for getting the optimum temperature 70 °C for the mass flow rate of air in the range of 0.4 to 0.7 kg/s. While, Fig. 9 shows that the outlet temperature of supercharged air ($T_{sca(o)}$ or $T_{air(out)}$) has changed significantly with the variation of coolant flow for the variation of flow rate of supercharged air in the range of 0.1 to 0.7 kg/s. Result shows that $T_{air(out)}$ has changed from 72 to 87 °C, 84 to 104 °C, 98 to 120 °C, 107 to 147 °C, 118 to 155 °C, and 132 to 172 °C. The results indicate that it is difficult to maintain the optimum temperature 70 °C of weHS^c by doing the optimum operation of supercharger and coolant pump. Therefore, without interrupting the supercharger and coolant pump, we have introduced a fuzzy intelligent controlled micro facet exhaust gas recirculation (MiF-EGR) system with intake manifold thermistor (temperature sensor) to control the weHS^c supercharged air temperature in the range of 60–70 °C.

3.3. *Model for weHS^{ex}: MiF-EGR*

The MiF-EGR controls a small passageway between the intake and exhaust manifolds. The MiF-EGR has been controlled by a Fuzzy Adaptive Controller (FAC). The FAC valve activates based on the response of the thermal sensor which is placed just at the top of the intake valve. If the intake air temperature goes to 100 °C the FAC valve will be activated and allows the cool exhaust enters the intake manifold and cool down the supercharge air

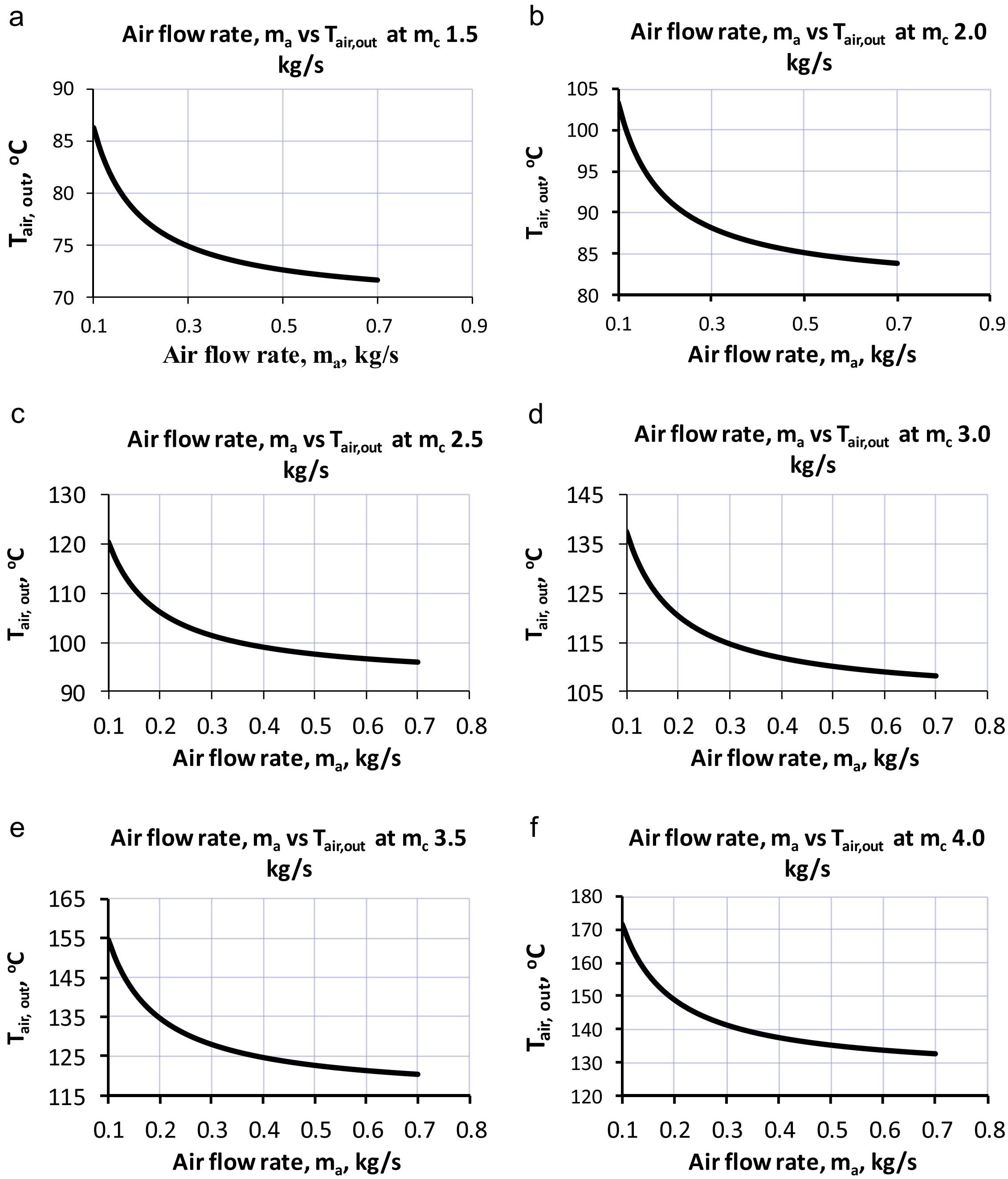


Fig. 9. Temperature of weHSc at different air flow rate.

temperature and reduces the generation of NO_x. The weHS^c air temperature will maintain in the range of 60°–70 °C by EGR in the range of 10–15% of the total mass of exhaust which will be used to decrease the combustion temperature.

3.3.1. Mathematical model for weHS^{ex}-MiF-EGR

The effectiveness of the weHS^{ex} is verified by estimating the 4-stroke cycle engine efficiency which is considered as ideal cycle engine. To simplify the analysis of weHS^{ex} certain conditions are assumed. Such as:

(i) Engine cycle is ideal, the pressure of the mixture during the intake and exhaust stroke is the pressure of atmosphere, and hence there is no "pumping loss".

(ii) Conservation of mass, mass flow of air + mass flow of fuel = mass flow of exhaust gas.

(iii) MiF-EGR valve is considered as a standard orifice and it opens for the intake air temperature at intake manifold > 70 °C.

The MiF-EGR valve's opening and closing operation is conducted by using FAC with the response of the thermal sensor. The % of exhaust gas flow by EGR can be computed by using the equation as follows:

$$0/0\mathrm{EGR} = \frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_a + \dot{m}_f + \dot{m}_{\mathrm{EGR}}} \tag{12}$$

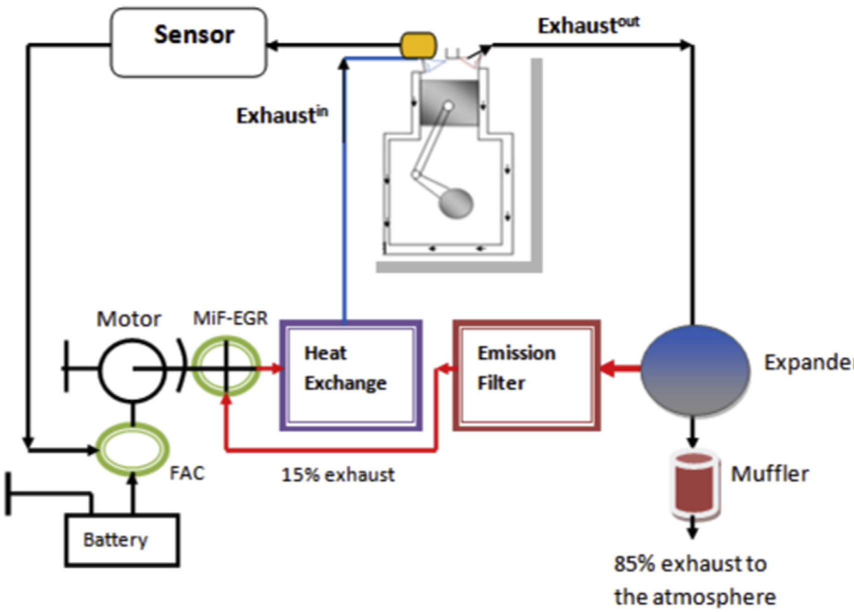


Fig. 10. Ideal model EGR system for this study.

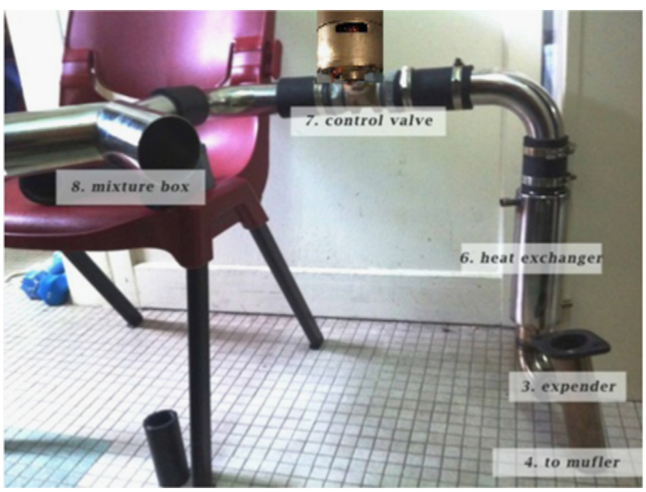


Fig. 11. Develop FAC-EGR prototype.

$$\%\mathrm{EGR} = \begin{vmatrix} 0\% & \text{if } T_{im} < 60\,^{\circ}\mathrm{C} \\ 5\% & \text{if } 60\,^{\circ}\mathrm{C} < T_{im} \leq 70\,^{\circ}\mathrm{C} \\ 20-30\% & \text{if } 70\,^{\circ}\mathrm{C} < T_{im} \leq 75\,^{\circ}\mathrm{C} \\ 30-50\% & \text{if } 75\,^{\circ}\mathrm{C} < T_{im} \leq 80\,^{\circ}\mathrm{C} \\ 50-75\% & \text{if } 80\,^{\circ}\mathrm{C} < T_{im} < 90\,^{\circ}\mathrm{C} \\ 100\% & \text{if } T_{im} = 100\,^{\circ}\mathrm{C} \end{vmatrix} \tag{13}$$

where, $\dot{m}_{\mathrm{EGR}}$ is the mass flow rate of EGR, $\dot{m}_a$ is the mass flow rate of air, and $\dot{m}_f$ is the mass flow rate of fuel respectively in kg/s, T_{im} is the air temperature at intake manifold. To maintain the % of EGR flow, the program into the FAC has been adapted based on Eq. (13).

3.4. Development of MiF-EGR prototype

Temperature of the exhaust gases of a typical SI engine is average 400 °C to 600 °C. This drops to about 400 °C at idle conditions and goes up to about 900 °C at maximum power and forms NO_x which causes emission and detonation problems. To reduce the formation of NO_x, combustion temperatures must be kept below 1300 °F [32]. This is done by recirculation a small amount of exhaust through the EGR valve. The proposed EGR system in this study has been developed by developing an intake system for introducing a fuel–air mixture into engine, an exhaust system for discharging exhaust gases from engine to the atmosphere. Fig. 10 shows the theoretical model of EGR. The exhaust gas passes through the bypass pipe, expender, muffler, filter exhaust gas box, the EGR heat exchanger, flap via control valve and mixing chamber. The throttle flap in the intake channel allows the differential pressure to rise and thereby the EGR quantity be changed. The exhaust gases leave the engine through exhaust manifold and goes through exhaust ducting, partial of the exhaust passes through intermediate expander and the rest is move through muffler to be expelled out the exhaust pipe. The electronic circuit signals relay in the control valve to control the flow of exhaust recirculation to the engine intake manifold in response to engine intake air temperature. The exhaust gas is taken directly downstream of the cylinders and cooled. The EGR valve then regulates the subsequent mixing of exhaust gas with the intake air. Since harmful nitrogen oxides (NO_x) are mainly produced at high temperatures. This process can reduce the quantities of NO_x and CO_2 emissions and reduce the fuel consumption and recover the waste heat of the engine. Fig. 11 shows prototype of EGR system which will be used to control the weHS[c] air temperature in the range of 60–70 C. Control Strategy of MiF-EGR

Temperature of the engine partially control with fuzzy logic controller is designed to realize the exhaust through the heat exchanger and thus optimize the engine temperature. The controller can overcome the disadvantage of the conventional PID with un-adjustable parameter setting. It has the advantage of fuzzy controller being simple (relations between input and output variables can be explained in a linguistic-based rule base), robust (performance is not depending on training and new input variables and rules can be easily added) and not requiring precise mathematical model (Rahman et al. [17]), Ataur et al. [30]. In the control system of the engine emission of NO_x, temperature T is selected as controlled variable, and exhaust flow rate Q as regulated variable through the change in valve position. The block diagram of the developed control system is shown in Fig. 12.

Based on the difference between measured value ($T_{cy(i)}$) and reference value (T_r), the temperature of the engine is controlled by regulating flow rate of the exhaust ($\dot{m}_{\mathrm{EGR}}$). The reference temperature T_r is calculated based on the maximum allowable temperature of the engine and then is compared with the measured temperature using temperature sensor. Hence, the resultant deviation, i.e., position error (PE), $e = T_{cy(i)} - T_r$ and rate of position error (RPE) $(\partial e/\partial t) = \dot{e} = \partial(T_{cy(i)} - T_r)/\partial t$ are continuously measured in operation. For implementation of fuzzy values into the system by using fuzzy logic system (FLS), MiF-EGR opening (MiF-EGR$^{(O)}$), supply current (I) are used as input parameters and temperature of the cylinder chamber is used as output parameter. The MiF-EGR$^{(O)}$ in % has been maintained by controlling the power flow of the motor. For fuzzification of these factors, the linguistic variables low (L), medium (M), high (H) and very high (VH) are used for the input parameter of MiF-EGR opening and low (L), medium (M), high (H), and very high (VH) are used as output parameters of EGR. The MIF-EGR$^{(O)}$ (H) indicates that MiF-EGR opening in the range of 95–98% for the temperature of the engine cylinder chamber closed to 70 °C (i.e., $60^\circ < T_{cyc} \geq 70$ °C). While, MiF-EGR opening will be 100% for maintaining 15% of exhaust gas when $T_{cyc} > 70$ °C. In earlier, it is mentioned that only 5–15% exhaust is considered to recycle. Cylinder chamber temperature can be estimated as, $T_{cyc} = T_{cy(i)} + T_{r(emission)}$, where, $T_{r(emission)}$ is the remaining temperature of the emission such as CO and HC. Total of 27 rules are formed. Parts of the rules are shown as follows. For example, Rule 1 and 13 can be interpreted as follows:

$$\text{Rule 6 : If MiF} - \text{EGR}^{(O)} = L \text{ and } I = L, \text{ then } T_{cyc} = L.$$

$$\text{Rule 13 : If MiF} - \text{EGR}^{(O)} = M \text{ and } I = M, \text{ then } T_{cyc} = M$$

$$\text{Rule 27 : If MiF} - \text{EGR}^{(O)} = H \text{ and } I = H, \text{ then } T_{cyc} = VH$$

Fuzzifications of the MiF-EGR$^{(O)}$, supply current (I) and T_{cyc} are made by aid follows functions. These formulas are determined by using measurement values.

$$I(i_1) = \left\{ \begin{array}{ll} i_1; & \text{if } 60^\circ < i_1 \geq 70\ ^\circ\text{C} \\ 0; & \text{if } T_{cyc} \leq 60\ ^\circ\text{C} \end{array} \right\} \tag{14}$$

$$\text{MiF\_EGR}(i_2) = \left\{ \begin{array}{ll} i_2; & \text{if } 60^\circ < T_{cyc} > 70\ ^\circ\text{C} \\ 0; & \text{if } T_{cyc} \leq 60\ ^\circ\text{C} \end{array} \right\} \tag{15}$$

$$T_{cyc}(o_1) = \left\{ \begin{array}{ll} o_1; & \text{if EGR } \leq 15\% \\ 0; & \text{if EGR} = 0\% \end{array} \right\} \tag{16}$$

In Eqs. (14)–(16), i_1 is the first input variable (MiF-EGR$^{(O)}$), i_2 is the second input variable (I), and o_1 is the first output variable (T_{cyc}) are the fuzzy sets for the fuzzy variables which are set up using MATLAB FUZZY Toolbox. In order to establish the relative

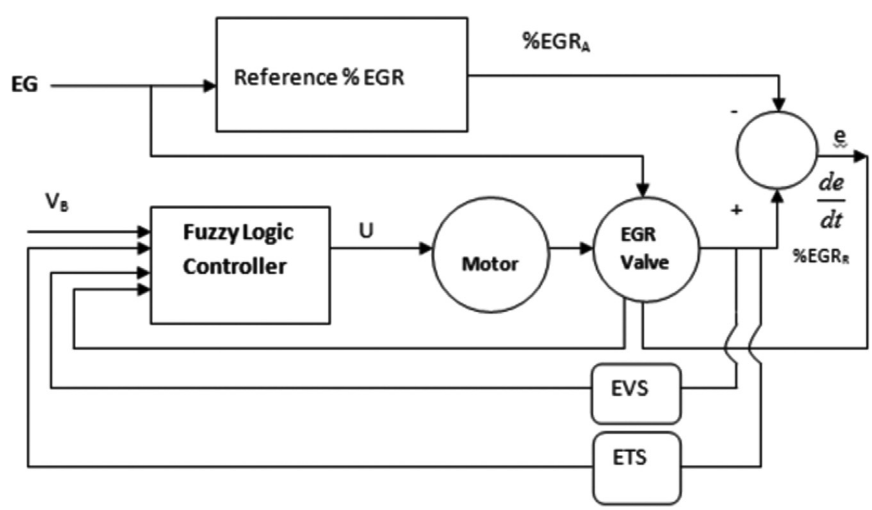


Fig. 12. Block diagram of the control system.

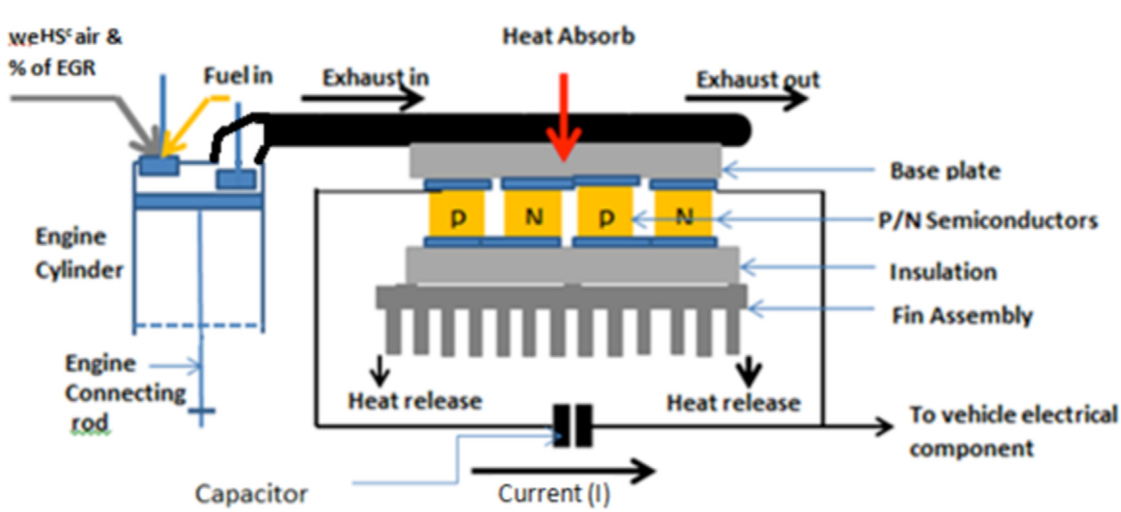
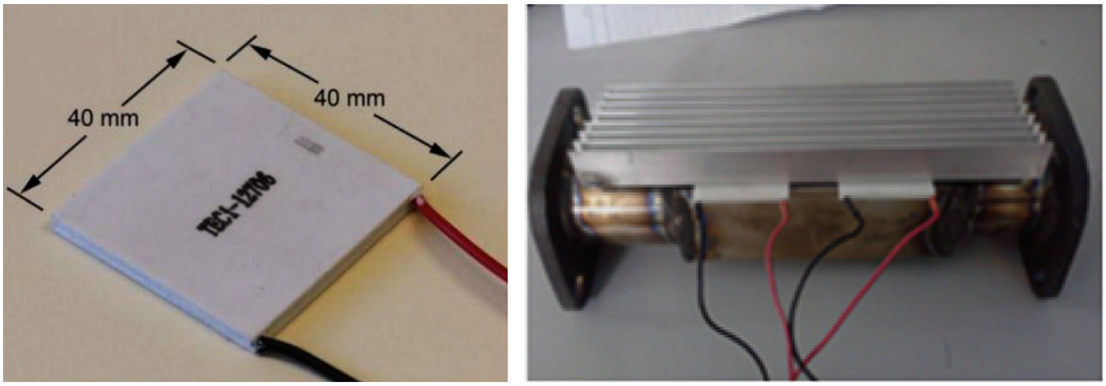


Fig. 13. Model of TEG with EGR and weHS^c.



P/N module

Prototype of TEG

Fig. 14. Sample of thermal electric generator.

error (ε) of structure, the subsequent equation is used:

$$\varepsilon = \sum_{i=1}^{n} \left| \frac{y - \hat{y}}{y} \right| \frac{100\%}{n} \tag{17}$$

The goodness of fit (η) of the predicted system is calculated by the following equation:

$$\eta = \sqrt{1 - \frac{\sum_{i=1}^{n} (y - \hat{y})^2}{\sum_{i=1}^{n} (y - y_{mean})^2}} \tag{18}$$

where, n is the number of interpretations, y is the measured value, $y|widehat$ is the predicted value, and y_{mean} is the mean of measured value. The relative error provides the difference between the predicted and measured values and it is necessary to attain zero. The goodness of fit also provides the ability of the developed system and its highest value is 1.

4. Model for weHS^{ex}: Finned type TEG

Fig. 13 shows the model of finned type TEG with EGR and weHS^{c}. The main focus of the thermoelectric generator is to generate electricity based on temperature gradient. The direct conversion of heat energy into electrical energy can be accomplished through the *Seebeck* effect. The TEG is developed by covering the exhaust pipe with the thermocouples. The P/N thermocouple consists of a single pellet of P-type and N-type thermoelectric material each that are connected electrically in series. Heat carries the majority carriers from one junction to the other producing electrical power. By placing P/N couples in series electrically and in parallel thermally, TEG module has been constructed that generates a voltage proportional to the temperature gradient across the elements.

Two thermoelectric modules as shown in Fig. 14 are placed with the smooth surface of hollow pipe by using a fin in each sides of the exhaust pipe which can be considered as the heat exchanger of the TEG. The combined of the exhaust pipe, thermoelectric module and fin is called Thermal Electric Generator (TEG). All of the thermoelectric modules are connected in series in order to increase the voltage of the TEG. Positive terminal of the thermoelectric module is connected with the positive terminal of a super-capacitor. The super-capacitor can be able to save even a fraction of voltage which could be considered as the power source of the few electrical components (loads) of the vehicle. The thermoelectric module classified as TEC1-12712 is built with Bismuth Telluride (Bi_2Te_3). The electromagnetic responses of TEG can be enhanced by using an inductive filtered HVDC converter transformer with field circuit coupling [33,34]. The resulting voltage (V_v) is proportional to the temperature difference (ΔT) via the *Seebeck* coefficient, α. By connecting an electron conducting (N-type) and hole conducting (P-type) material in series, a net voltage is produced that can be driven through a load connected with supercapacitor. A good thermoelectric material has a *Seebeck* coefficient between 100 μV/K and 300 μV/K. Thus, in order to achieve a few volts at the load, many thermoelectric couples has been connected in series to make the thermoelectric device. The electric current through the TEG couple, connected to an external load of resistance R_L can be estimated as,

$$I = \frac{V_v}{(R_L + R_{\mathrm{TEG}})} \tag{19}$$

with

$$V_v = \alpha(T_h - T_c)$$

Therefore, the heat transfers from the hot side and heat sink side of the TG thermocouple junction can be modeled as,

$$Q_h = \frac{[\alpha(T_h - T_c)]^2}{(R_L + R_{\mathrm{TEG}})} - \frac{1}{2} I^2 R_{\mathrm{TEG}} + \gamma_{\mathrm{TEG}}(T_h - T_c)) \tag{20}$$

with

$$\gamma_{\mathrm{TEG}} = \frac{\lambda_p \rho_p}{L_p} + \frac{\lambda_n \rho_n}{L_n} \text{ and } R_{\mathrm{TEG}} = \frac{L_p \rho_p}{A_p} + \frac{L_n \rho_n}{A_n}$$

A thermoelectric generator converts heat energy (Q) into electrical power (P) with efficiency η. The amount of heat, Q_h, that can be directed though the thermoelectric materials frequently depends on the size of the heat exchangers used to harvest the heat on the hot side and reject it on the cold side. The efficiency of a thermoelectric generator depends heavily on the temperature difference, $\Delta T = T_h - T_c$ across the device. It is noted in earlier that the TEG has been developed with finned which is considered as the heat exchanger is used to reduce the temperature on the cold side (T_c) significantly to increase the temperature gradient. So that the temperature gradient ΔT will increase consequently the voltage of the TEG will be high as the *Seebeck* coefficient (α) is constant. This is because the thermoelectric generator, like all heat engines, cannot have efficiency greater than that of a Carnot cycle ($\Delta T / T_h$). The efficiency of a thermoelectric generator is typically defined as

$$\eta = \frac{\Delta T}{T_h} \cdot \frac{\sqrt{1 + ZT_{ex}} - 1}{\sqrt{1 + ZT_{ex}} + (T_c / T_{ex})} \tag{21}$$

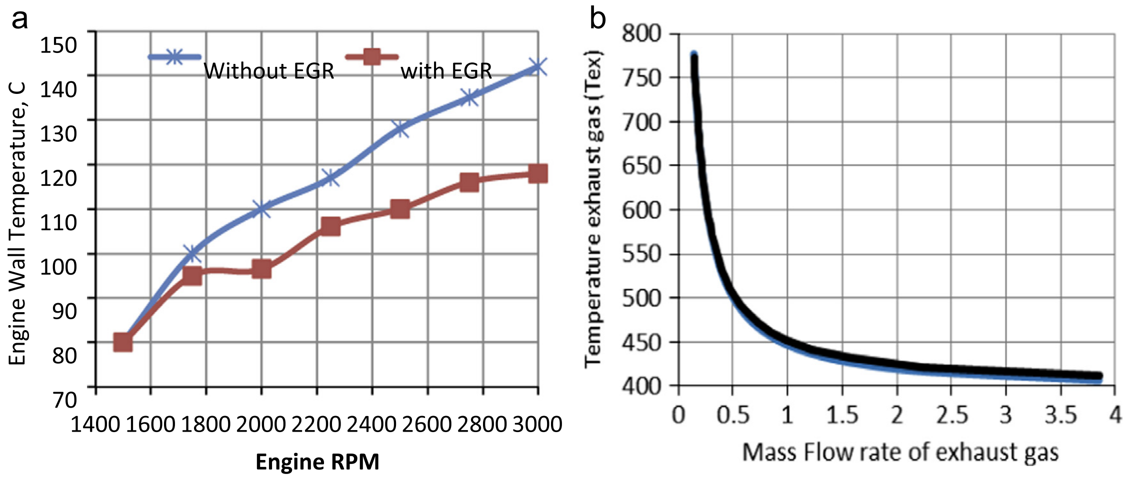


Fig. 15. Performance of EGR with engine RPM.

with

$$zT_{ex} = \frac{\alpha^2 T_{ex}}{\rho\, k}$$

where, the first term is the Carnot efficiency and ZT_{ex} is the merit for the device, α is the *Seebeck* coefficient, ρ is the electrical resistivity, and κ is the thermal conductivity, and T is the temperature. The temperature in the exhaust system is the ideal-gas isentropic expansion relationship between the temperature and pressure. The temperature of the exhaust (T_4 or T_{ex}) can be estimated by using the equation:

$$T_{ex} \text{ or } T_4 = T_{cy}\left(\frac{P_{ex}}{P_{cy}}\right)^{(\gamma-1/\gamma)} \tag{22}$$

with

$$P_{cy} = p_3\left(\frac{v_3}{v_{cy}}\right)^{\gamma} \text{ and } T_{cy} = T_{sca(i)} + \left[T_{c(i)} - \frac{1}{\dot{m}_c c_p}\frac{\dot{Q}}{\left[\exp\left(-\frac{h_c dL}{\dot{m}_c c_{p(c)}}\right)-1\right]}\int_L^0 dl\right] + T_{ruf}$$

$$v_{cy} = v_c\left[1+\frac{1}{2}(r_c-1)\left(R+1-\cos\theta-\sqrt{R^2-\sin^2\theta}\right)\right],\quad R=\frac{r_c}{a},\ \text{and}\quad a=\frac{S}{2},$$

The maximum temperature of the exhaust can be computed by applying the energy conservation law:

$$T_{\max} = T_{ex} + \Delta T = T_{cy}\left(\frac{P_{ex}}{P_{cy}}\right)^{(k-1/k)} + \frac{V_{ex}^2}{2gc_p} \tag{23}$$

where, T_{ex} and P_{ex} and T_{cy} and P_{cy} are the temperature and pressure of the exhaust and cylinder respectively, c_p is the specific heat at constant pressure, v_{cy} is the cylinder volume, v_c is the cylinder clearance volume, θ is the crank angle, r_c is the compression ratio, a is the crank offset and S is the stroke length. It could be mentioned that blowdown of the exhaust occurs once the exhaust valve opens before BDC. The flow of the exhaust at beginning of the exhaust manifold has high sonic velocity and kinetic energy. The kinetic energy of the exhaust is converted to additional enthalpy with an increase in temperature. V_{ex} is the velocity of the exhaust in m/s. The velocity of exhaust can be estimated as, $V_{ex} = (E_{cap} \times N)/({}^{exs}n \times 60)\left[1/(100A_{\exp})\right]$ where, V_{ex} is the speed of the exhaust in m/s, A_{exp} is the cross section of the exhaust pipe in cm^2, E_{cap} is the capacity of engine in litre (or cm^3), ${}^{exs}n$ is the revolution to get exhaust stroke (${}^{exs}n=2$), N is the rpm of the engine. Steady state exhaust temperature of SI engines is generally in the range of 400°–600 °C, with extremes of 300–900 °C. The temperature drops in exhaust flow between the engine and catalytic converter about 156 °C [32].

5. Result and discussion

The weHS^c and weHS^{ex} have been developed for the LHRE@208 CA engine. The EGR's motor controller has been connected with the thermistor of the intake stroke and performed the demonstration test for testing the effectiveness of the EGR. The heat exchanger of the EGR has been filled with thermal fluid 'Duratherm XLT-120' with optimal temperature range of −84° to 65 °C and tests its performance to drop the exhaust gas temperatures from 600 to 20 °C. The prototype of finned TEG is fixed around the exhaust pipe just at few centimeters away from the exhaust manifold and connects the eight TEG modules in series to get the higher voltage which could make the rating voltage of 12 V for the vehicle electrical system such as fuel pump motor or the engine's primary ignition coil.

The engine has been operated at maximum 6000 rpm for investigating the performance of weHS^c and weHS^{ex}. Fig. 15 represent the variation of the energy balance for engine speed in the range of 1700–6000 rpm with FAC-EGR valve opening of 25–100%. The exhaust energy has increased by 12% with the low operating speed of LHRE. However, the exhaust gas energy increased to 15% above the standard engine at medium load. In the LHRE, the exhaust gas energy increased by more than 21% at high load condition. In the LHRE configuration, there is a decrease of volumetric efficiency. Engine intake temperature increases with increasing the cylinder wall temperature with increasing the engine rpm. Fig. 15(a) shows the engine wall temperature increasees upto 142 °C without EGR and 118 °C with EGR when the engine operating speed was maximum 3000 rpm. Fig. 15(b) shows the changing of exhaust temperature with EGR. If the engine operating speed is considered to maximum 6000 rpm, the engine wall temperature will be 224 °C without EGR and 142 °C with EGR. Willard [32] reported that the engine wall temperature should not exceed 180–200 °C to assure thermal stability of the lubricating oil and structural strength of the wall for the operational speed of the engine in the range of 1000–6000 rpm. Therefore, weHS^c equipped engine must be used with MiF-EGR in order to maintain the engine wall temperature in the range of 140–160 °C for ensuring the lubrication stability and reduce the NO_x emission and CO and HC formation into the cylinder chamber. Furthermore, NO_x formation can reduce in modern engine with fast burn combustion chamber and this can be made by using weHS^c which allow the fuel automization and vaporize about 90% by keeping the intake temperature in the range of 60–70 °C. The importance of MiF-EGR is has been used in this study to maintain the intake desiredtemperature. So that engine wall temperature will be in the

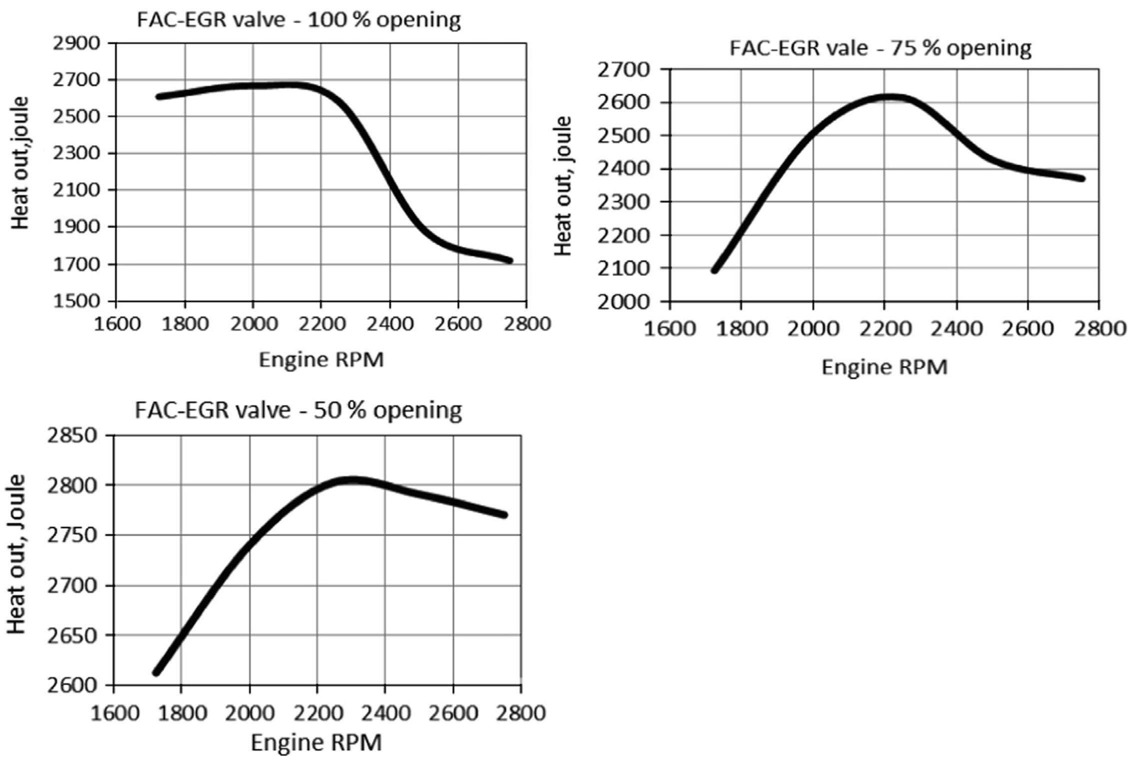


Fig. 16. Performance of FAC-EGR for different engine RPM.

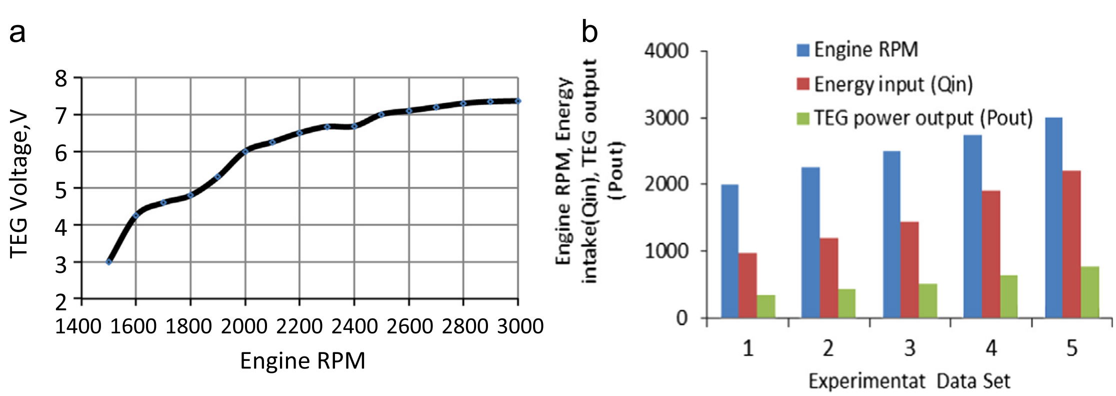


Fig. 17. Performance of TEG with engine RPM: (a) Electrical current, (b) TEG output with engine RPM and intake.

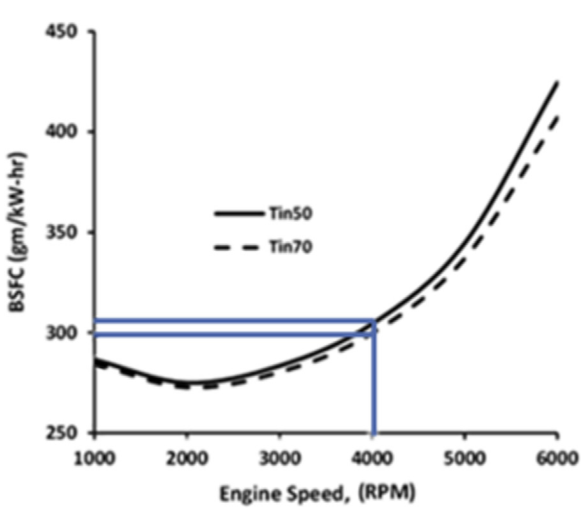


Fig. 18. Fuel consumption for weHS equipped engine.

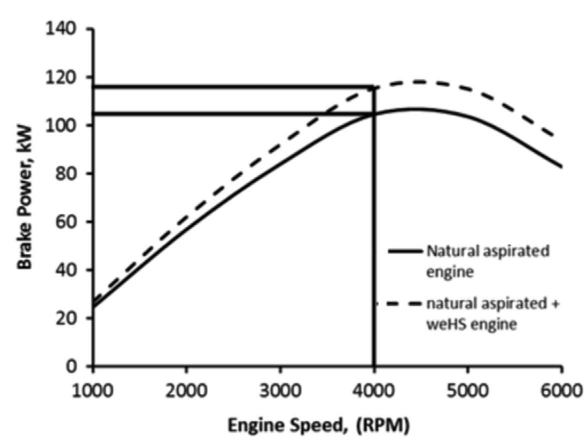


Fig. 19. Brake of weHS equipped engine.

Table 1
Benifit of weHS engine.

(i) Economically	Car engine without weHS	Car engine with weHS	Comparism (%) Engine with weHS over engine without weHS
Fuel required (l) per 100 kM	10	8.0	20
Fuel cost (RM) per 100 kM	32.00	25.60	20
(ii) Environtally			
CO_2 emission (g) per 100 kM	12,000	9,360	22
NO_x emission (g) per 100 kM	50	41	18

range of 140–160 °C and save the fuel energy lost and safe the envirnment from harmful NOx emission. This conclusion can be supported by Reitz [22]. Reitz reported that up to 30% of fuel energy is lost to the wall heat transfer.

Fig. 16 shows the performance of MiF-EGR on the heat out of the engine. The results show that the heat out of the engine decreases with increasing the opening of MiF-EGR. It could be due to the effect of fuel vaporization and engine's volumetric efficiency.

The TEG performance has been conducted with engine RPM range from 2000 to 3000 rpm. Fig. 17(a and b) show that the current and voltage development by the TEG increases with increasing the RPM of the engine which is due to the increment of fuel heating value. This is due to the dynamic effect of weHS^c combined with MiF-EGR. It could be mentioned that weHS^c heated intake air in the range of 60–70 °C which allows the fuel atomization and vaporization up to 90% and fuel ignite almost completely in the combustion chamber. So that the engine brake power will increase and reduce the fuel consumption and improve emission. It is conluded that the TEG efficiency decreases with increasing the engine rpm and start to decreases from 2700 rpm. This is becasue of the engine consumps more fuel to speed up from 2000 to 2700 rpm with overcoming the engine stall torque and accelerate the vehicle.

Figs. 18 and 19 shows the performance of weHS equipped engine which has been investigated by using GT suite software. Fig. 18 shows that the specific fuel consumption of engine has improved by 3% due to reduction of HC formation into the engine combustion chamber. Fig. 19 shows that the engine equipped with weHS has developed more power than the natural aspirated engine with increasing speed from 1000 to 5000 rpm which is due to the fuel atomization and vaporization and complete combustion while at 6000 rpm engine has experienced less power which due to the fuel energy is lost to the wall heat transfer and the energy gain by the weHS is less than the power consumption by the supercharger and EGR system. This conclusion can be supported by Reitz [22]. The results also indicate that the overall power recovered from the waste heat about 7%.

5.1. Carbon dioxide emission estimation

When fuel is burned, most of the carbon is eventually oxidized to CO_2 and emitted to the atmosphere. The emission of carbon per litre of fuel can be estimated as,

$$C_{emission} = m_c f_{c(oxidized)} \quad (24)$$

with

$$m_c = Q_{fuel} sf_{cc(coeff)} \text{ and } Q_{fuel/l} = V_{fuel} H_{fuel}$$

where, m_c is the mass of carbon content in fuel in kg, C_{fuel} is the carbon content in fuel in kg/J, Q_{fuel} is the fuel combustion energy in J, H_{fuel} heating value of fuel in J/cm^3. Eq. (24) can be rewritten:

$$C_{emission} = V_{fuel} H_{fuel} C_{fuel} f_{c(oxidized))} \quad (25)$$

where, $f_{c(oxidized)}$ is the fraction of carbon oxidized in percentage per litre of fuel.

$$CO_{2(emission)} = C_{emission} \frac{CO_2(m.w)}{C(m.w)} \quad (26)$$

by simplifying Eq. (26) by using Eqs. (24) and (25)

$$CO_{2(emission)} = V_{fuel} H_{fuel} C_{fuel} f_{c(oxidized))} \frac{CO_2(m.w)}{C(m.w)} \quad (27)$$

The amount of fuel (volume) can be estimated directly from the vehicle mobile source.

The Table has presented the estimated CO_2 emission based on the fuel requirement for the engine with weHS and without weHS.

Note: Fuel price RON 95/l before subsidization = RM 3.2

6. Economic and environmental benefit of weHS equipped engine

Emission generations of the engine can be reduced bycontrolling the local operating conditions such as fuel vaporization and engine cylinder combustion heat transfer. This study has presented weHS^c and weHS^{ex} which are greatly contribute on the fuel vaporization and heat transfer. By introducing a weHS^c can increase the injection pressure and temperature of air which helps to vaporise fuel in the engine combustionchamber with a finer droplet size and reduces HC, CO and particulate emissions, but increases cylider wall temperature and NO_x emissions. By introducing MiF-EGR fuzzy logic controling technique the temperature of the engine cylinder chamber will be kept in the range of 120–140 °C so that NOx emission will be reduced significantly. Ecomic and environmental benifit of weHS engine has shown in Table 1.

7. Conclusion and recommendation

Following conclusionshave been made based on the performance of weHS:

(i) Engine combustion chamber's fuel atomization and vaporization and heat transfer needs to control more accurately in order to get more engines BMEP and reduce emission. The designed MiF-EGR and weHS^c could be considered as an innovative solution for engine fuel atomization & vaporization and heat transfer control.

(ii) The crucial role of MiF-EGR on the weHS^c to make the intake temperature in the range of 60–70 °C which could.
   a. Enhance the engine brake power about 7% by harvesting waste heat energy of the engine
   b. Reduce the engine fuel consumption and improve engine emission.

(iii) The engine waste energy has been converted into electrical energy by using the modified TEG which is accounted about 200 W and 20% of alternator output. By improving the heat flux leaking at the hot junction and improving the design of cold junction of TEG, the temperature gradient can be increased and more waste heat energy of engine can be harvested.

(iv) Total of 15% waste heat energy of engine could be recovered by effectively using the weHS.

(v) The engine BMEP and efficiency might be reduced by 15% and 10% respectively if weHS[c]outlet (engine intake) air temperature reaches to 110 °C or more.

(vi) The MiF-EGR valve can be operated by means of electromagnetic field development by installing temperature sensors around the combustion chamber of the engine which could replace the small size electrical motor in the current design of MiF-EGR.

8. Futhure work

The experimental investigation on our innovative product weHS of engine will be conducted once our final protoytpe will be ready. The extensive experimental study on our weHS will be sumbitted for possible publication in the same journal. By employing advanced thermocouple with the engine wall, we will try to monitor the lost of fuel due to the cylinder wall temperature and formation of NO_x.

Acknowledgement

Authors are very much greatful to Research Management Center, Internal Islamic University Malaysia for financing this study. Authors are also grateful to the technician of Automotive Laboratory for helping to develop the product of weHS.

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