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A Design and Fabrication of Heat Exchanger for Recovering Exhaust Gas Energy from Small Diesel Engine Fueled with Preheated Bio-oils

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Abstract

The exhaust gas source from diesel engines with high temperature was an energy one, in which was utilized to not only improve the operation efficiency of diesel engines, but also reduce the environmental reduction. However, this energy source from small diesel engines has almost not been recovered, utilized in comparison with ever-depleting fossil fuel. A pure vegetable oil heating system by directly utilizing exhaust gas heat, and using heated pure vegetable oil as fuel in that diesel engine was presented in this work. This system was installed in experimental diesel engine to determine the exhaust gas temperature and check the working features of engine at 10%, 25%, 50%, 75%, 100% of engine load, and 1500 rpm, 2000 rpm of engine speed. The paper results showed that, after installing the heat exchanger in the engine exhaust pipe, the technical indicators of experimental diesel engine were guaranteed, the difference between simulation and experimental results was not significant, the average of utilized exhaust gas energy was about 52%. Moreover, the resistance in the exhaust pipe at 75% of load was 191 Pa and 468 Pa at 100% of load compared to 1472 Pa of requirement. Backpressure in the exhaust pipe line was simulated by ANSYS FLUENT. Some benefits such as saving energy, reducing the environmental pollution and the dependence on fossil fuel, contributing the efficient energy use were obtained as utilizing the exhaust gas energy of the engine for heating pure vegetable oil, and using heated pure vegetable oil as the alternative fuel.

Keywords: exhaust gas energy, design, diesel engine, heat exchanger, bio-oils

INTRODUCTION

In among of main power in the transportation, construction, fishery and agriculture machinery, engine has played an important part and consumed more than 60% of fossil fuel, thus it was able to result in exhausting the fossil fuel. Recent propensity about using the energy sources aiming at reducing the fossil fuel consumption as well as pollution was considered as urgent task. Up to now, the major consumer of fossil fuel was the internal combustion engines (ICE), however only about 30-40% energy of combustion in the engine chamber was transformed into useful mechanical work [1-4]. The rest heat source expelled to the environment or lost through exhaust gases and cooling water/oil were approximately 25–35%, hence it was necessary to utilize and recover the waste heat to increase the heat efficiency of internal combustion engines. The waste heat recovery and utilization not only saved energy but also reduced the toxic pollution. Engine manufacturers have implemented and improved the latest techniques to increase thermal efficiency by enhancing the fuel-air mixing, using turbo-charger, and variable valve timing or advance combustion chamber [5-7].

WH was heat generated by the fuel combustion or chemical reaction. In the time of engine run, four sources of WH such as exhaust gas, cooling oil/water/liquid, lube oil, and turbocharger were dissipated to the atmosphere from the engine. WH depended on not only the temperature of the waste heat gases, but also mass flow rate of exhaust gas of engines. Exhaust gas temperature of diesel engines after leaving the engine were as high as 450 - 600°C. Consequently, the higher the exhaust gas temperature was, the higher the heat value was, however, the temperature of exhaust gases were limited by the laws of thermodynamics. Total energy from diesel engines was shown in the Figure 1.

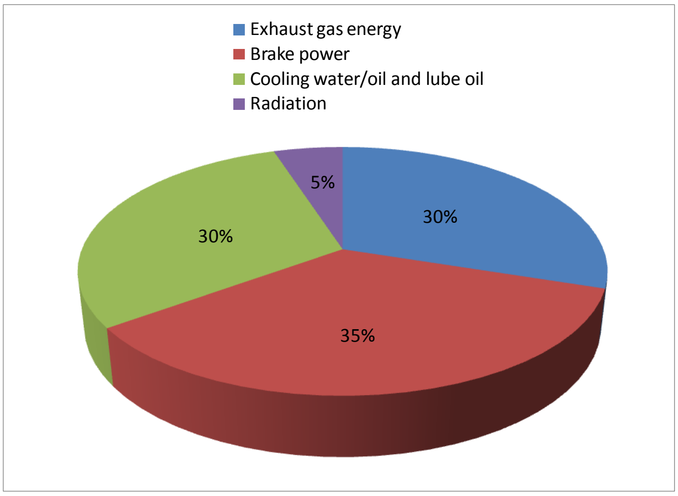

Figure 1. Total energy in diesel engines

For diesel engines in the ships that were bigger than 3000 hp of the power, the utilizing of exhaust gas energy was much more interested due to the space of engine room was large and exhaust gas heat energy was high. The exhaust gas energy of these engines could be recovered through the system of auxiliary – exhaust gas boiler or turbo - compressor aimed to heat up heavy fuel or turbochargers. However, small power engines, especially the diesel engines with less than 100 hp of power, it was not occasional to arrange the devices for utilizing the exhaust gas energy [8,9]. In automobile diesel engines, significant amount of heat energy of exhaust gas (about 35% of the thermal energy) was released into the environment. The amount of such losses were recoverable at least partly or greatly and depend on the engine load and engine speed. Among various advanced methods, exhaust energy recovery for automobile diesel engines were proved to improve fuel consumption, reduce CO_2 and other harmful emissions.

WH recovery from diesel engines brought many big benefits not only high power engines, but also smaller engine. Benefits from WH recovery might be divided into direct or indirect benefits.

Direct benefits: Recovery of WH from diesel engine might affect on the efficiency of combustion process due to it increases the obtained total energy, hence WH recovery was considered a solution with lower costs, cutting emissions, and especially, increasing the EEDI (Energy Efficiency Design Index) for the ship.

Indirect benefits: Some indirect benefits from WH recovery were included the pollution reduction, the device size reduction because of reducing the total fuel consumption, reduction in energy consumption for auxiliary devices such as boiler, compressor.

Many researchers have recognized that WH utilization and recovery from engine exhaust gas was considered as the potential in order to decrease fuel consumption without increasing emissions. In the systems of WH, the energy might be recovered by many different technologies. Many researchers about recovering the WH both on diesel engines [1] and gasoline engines [2] have been carried out. The results of combination of ETC (Electric turbo compound) and TEG (Thermo-electric generator) led 3% to 5% of fuel saving and 1% to 4% of CO_2 reduction [3] as this combination system was installed in was noticed. A combination between a TEG and an ORC (Organic Rankine cycle) system aiming at recovering the WR energy was investigated by Zhang et al. [4]. Alberto. A.B et al. [5] showed the way of recovering the WH from exhaust gases of ICE by using Organic Rankine Cycle (ORC) system. In this case, the engine power increased in comparison with 3.4% of the flow rate of fuel energy on average. However, the average flow rate of fuel energy increased up to 5.1% with 8.2% of top improvements if combined two ORCs. Moreover, the WH recovery by using Stirling engine was studied by Daloglu. A et al [6]. , this study showed that, Stirling cycle was useful, efficient to recover the WH of ICE without intake or exhaust gas. Similar results about Stirling engine advantages operated by WH such as high efficiency, favorable and quiet operation, non-emissions, low maintenance and vibrations, usability with many different fuels was presented [7].

It was also found that, pure vegetable oil was the fuel rapidly growing in use, and it should have good fluidity, low viscosity and good atomization which can only possible by preheating. There were many researchers used heating method to heat up pure vegetable oil aiming at direct using in diesel engines, Acharya S.K et al. [8] used kusum oil for small diesel engine and shown that viscosity was close to diesel's by preheating to100–130ºC. P. P. Sonune et al. [9] used mahua oil and concluded that at the temperature above 100ºC the viscosity reached to ASTM limits. Besides, R Rattan et al. [10] used mustard oil that was preheated up to 130°C to get the same density as that of diesel. R. Raghu et al. [11] used rice bran oil (RBME) and denoted that its viscosity was closer to diesel's as heated to 158°C. Soma C et al. [12] used waste vegetable oil which was preheated to 100°C and its viscosity became close to that of diesel. Ramadhas et al. [13] also denoted that jatropha and neem oil was preheated to overcome higher viscosity and lower volatility associated with biodiesel. A.T. Hoang et al. [14] compared the engine performance and thermal efficiency as using biodiesel, diesel oil and preheated vegetable oil.

The results from above researches showed that, the solutions of TEG, ORC, ETC, compressor-turbocharger, Stirling engines, exhaust boiler, and absorbent refrigerator for recovering the WH from diesel engines were popular. However, the WH utilization from small diesel engines for heating up pure vegetable oil in order to improve their properties equally to the properties of diesel fuel was almost not mentioned, although pure vegetable oil were renewable and able to rival to fossil fuels [15]. Thus, this paper presented the methods of utilizing the exhaust gas energy from small diesel engines to heat up pure vegetable oil, and using heated pure vegetable oil as the alternative fuel not only increased the efficiency of using energy, but also reduced the pollution emission.

MATERIALS AND METHOD

A. Materials

Chemically speaking, pure bio-oil included vegetable oils and animal fats after removing the water, ash, and free acid. In this study, straight coconut oil (SCO) was heated by exhaust gas energy through the heat exchanger and was used as fuel. The ASTM D1298 standard procedures were used to measure density, the ASTM D 445 standard was used to measure kinematic viscosity, and Du Nouy ring method with a tension meter based on the ASTM D971 standard was used to measure surface tension of the SCO that was one of pure vegetable oil. The physicochemical properties of SCO were presented in the Table 1.

Table 1. Physicochemical properties of SCO at room temperature

Properties	Methods	Unit	Result
Lower heating value, LHV	ASTM D 240	kJ/kg	35.85
Cetan number, CN	ASTM D 976	-	40
Density, D	ASTM D 1298	g/cm^3	0.9103
Kinematic viscosity, KV	ASTM D 445	cSt	35.3
Surface tension, ST	ASTM D 971	N/m	0.0322

It is observed that, HHV of SCO is 5 – 8% smaller than that of diesel fuel. However, the KV of the SCO is 7 – 10 times higher than that of diesel fuel. Therefore, heating SCO up to the suitable temperature in order to the KV of SCO close to diesel fuel'KV was necessary. The experimental test result about the relationship between the KV of SCO, ST of SCO and temperature shown that, heated SCO up 110ºC (HSCO_110) satisfied the requirements of diesel fuel based on Vietnamese standard 5689:2013. At this temperature, the KV and ST of SCO was the similar to those of diesel oil (DO).

B. Method

In this work, SCO was heated by using exhaust gas energy from a small D243 diesel engine that the technical parameters were briefly featured as Table 2.

Table 2. Technical parameters of engine D243

Description	Unit	Parameter
Power, N_e	HP	80
Revolution, n	rpm	2200
Bore, D	mm	110
Stroke, S	mm	125
Compression ratio, ε	-	16.7:1

The SCO heating system by exhaust gas energy was designed based on the principles of ensuring that the system met all the basic functions such as fuel storage, supplying, filter, safety. The theoretical basis for calculating the fuel heating system was based on the equations of energy balance and heat transfer to guide the design and manufacture the SCO heating system. The SCO heating system by utilizing exhaust gas energy was shown in Figure 2. First, SCO was heated by electricity energy up to 80ºC to help the cold-start of engine easily. After engine ran stably, the exhaust gas energy was flowed through the heat exchanger aiming the radiation heat transfer to SCO. A four-stroke four-cylinder diesel engine with 16.7:1 of compression ratio was used to conduct the experience as diesel engine ran at 1500 rpm and 2000 rpm at internal features with 10%, 25%, 50%, 75% and 100% of engine load. The maximum of engine power was 80 hp. The adopted methodology was presented following as:

1. A simple smooth tube heat exchanger as the preliminary design was installed in the exhaust manifold to get the heat energy from high temperature exhaust gas and it was connected to fuel injector through fuel filter. T_1 and T_2 were considered as EG temperatures inlet and outlet of heat exchanger. Meanwhile, t_1 and t_2 were considered as SCO temperatures before and after pre-heating by EG heat. The EG energy was calculated as equation:

$$Q_{EG} = G_{SCO}\left(\frac{G_a}{G_o g_c} G_o + 1\right) C_{p-EG}(T_1 - T_2)$$

$$G_a = \frac{\pi.D^2}{4} S.\eta_{\text{int}}.\rho_{\text{int}}$$

G_o - the required amount of dry air to burn completedly 1kg of SCO fuel, kg/kg fuel;

$$G_o = \left(\frac{\%C}{12} + \frac{\%H}{4} + \frac{\%S}{32} - \frac{\%O}{32}\right).\frac{32}{0,232}$$

$$g_c = \frac{G_{SCO}}{60.i.z.n}$$

G_a - The actual amount of air to burn the amount of fuel injected into the cylinder chamber of engine, kg/s

G_{SCO} – Mass flow of SCO fuel consumption of engine for 1 hour, kg/s;

C_{p-EG} – Isobaric mass heat capacity of EG, kJ/kg.K

T_1– Inlet exhaust gas temperature,ºC

T_2 – Outlet exhaust gas temperature, ºC

D – Diameter of engine cylinder, m;

S - Piston stroke, m;

η_{int} - Intake coefficient of the engine;

ρ_{int} – Density of intake air, kg/m^3.

g_c - The amount of fuel injected into the cylinder in a cycle, kg

i – Number of cylinders;

z – Stroke coefficient;

n – Engine speed, rpm;

The heat exchanger dimension was determined based on the energy balance equation as:

$$Q_T = F.k.\Delta t = (\pi.m.l.h.d_1)\left(\frac{1}{\frac{1}{\pi d_1 \alpha_1} + \frac{1}{2\pi\lambda}\ln\frac{d_1}{d_2} + \frac{1}{\pi d_2 \alpha_2}}\right)\left(\frac{\Delta t_1 - \Delta t_2}{\ln\frac{\Delta t_1}{\Delta t_2}}\right) = \eta_T Q_{EG} = Q_{SCO}$$

$$Q_{SCO} = G_{SCO} C_{P-SCO}(t_2 - t_1) = \left(n\frac{\pi d_2^2}{4}\rho_{SCO}\omega_{SCO}\right) C_{P-SCO}(t_2 - t_1)$$

Q_T –The used heat energy of EG for pre-heating SCO, W

Q_{SCO} – The required heat energy of SCO fuel, W;

C_{p-SCO} – Isobaric mass heat capacity of SCO fuel, kJ/kg.K

t_1 – Inlet SCO fuel temperature, ºC

t_2 – Outlet SCO fuel temperature, ºC

l – Length of SCO fuel pipe line, m;

d_1, d_2 – the outer and inner diameter of the tube, m;

η_T – EG recovery efficency;

ρ_{SCO} – Average density of SCO, kg/m^3;

ω_{SCO} – Flow speed of SCO in pipe line, m/s

m – Number of turns required;

h – Number of pipe line rows;

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α_1 – Convection coefficient from EG to SCO pipe line outer surface, W/m^{2}K

α_2 – Convection coefficient from SCO pipe line inner surface to SCO fuel, W/m^{2}K

λ - Conduction coefficient of material of from SCO pipe line, W/mK

δ - Thickness of SCO pipe line, m

k - Overall heat transfer coefficient, W/m^{2}K

Δt – Difference of logarithmic mean temperature, K

In this case, $\Delta t = \{[(T_1-t_1) - (T_2-t_2)]/\ln((T_1-t_1) - (T_2-t_2))\}$

2. By-pass valve was regulated in combination and used to control the EG flow rate. As the SCO temperature crossed 115°C, EG was navigated to discharge the ambient without passing through the heat exchanger. In this heat transfer process, SCO flow and EG was arranged parallel in same directions.

RESULTS AND DISCUSSION

A. Exhaust gas heat energy and heat exchanger

The relationship between exhaust gas temperatures of diesel engine D243 such as T_1 and T_2 and engine loads when engine was operated at internal feature at 1500 rpm and 2000 rpm and fueled with SCO was plotted in Figure 3. At the different temperatures, the isobaric mass heat capacity of EG was also various. The relationship between isobaric mass heat capacity of EG and exhaust gas temperature was considered as a linear function in this work following:

$$C_{p\text{-}EG} = 0.313T_{EG} + 1027, \text{ kJ/kg.K}$$

Thus, the relationship between EG heat energy and engine load through intermediate EG temperature variable was presented in Figure 2. The dimension of smooth tube heat exchanger for recovering the EG heat energy from engine was given in Table 3. The EG heat exchanger was installed on the test-bed of diesel engine D243 to evaluate the capacity and efficiency of EG heat energy and shown in Figure 3.

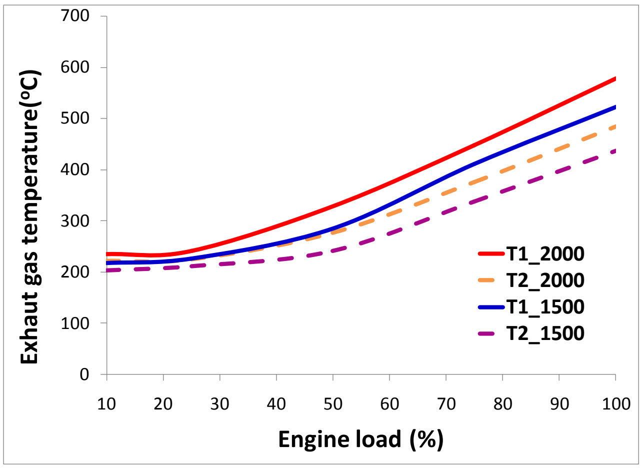

Figure 2. Relationship between engine load and EG temperature at 1500 rpm and 2000 rpm

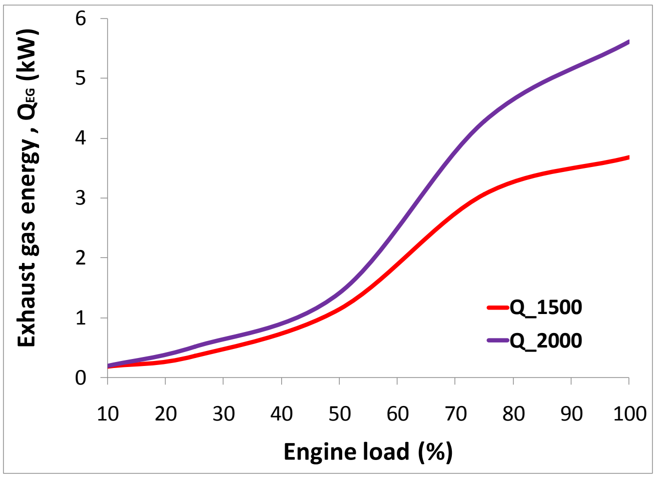

Figure 3. Relationship between engine load and exhaust gas energy at 1500 rpm and 2000 rpm

Table 3. Basic parameters of as-designed and as-fabricated heat exchanger

Parameters	Sign	Unit	Result
Diameter of SCO smooth tubes	d_2/d_1	mm/mm	12/8
Heat coefficient of SCO	α_2	W/m^2.K	25.3
Heat coefficient of exhaust gas	α_1	W/m^2.K	78.2
Thermal conductivity coefficient of tubes material	λ	W/m.K	35.5
Heat transfer coefficient	k	W/m^2.K	19.08
Heat transfer surface square	F	m^2	0.04

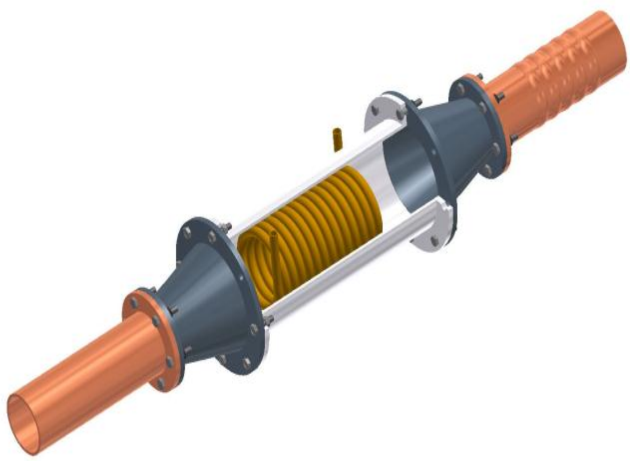

Figure 4. Structure of heat exchanger

Figure 5. The SCO heating system by utilizing exhaust gas energy

From Figure 2, Figure 3, it can be seen that, at low load (<50% of load), the EG temperature and energy were also low. At 75% and 100% of load with 1500 rpm of engine speed, EG temperature and energy were increased up to 300-400% compared with low loads. Exhaust gas temperature out of from heat exchanger was 338ºC and 437ºC, respectively. Besides, at 75% and 100% of load with 2000 rpm of engine speed, exhaust gas temperature out of from heat exchanger was 376ºC and 484ºC, respectively. Thus, in this work, the remaining EG energy after pre-heating SCO was high. With 50 liters of the initial volume of SCO tank, the EG energy from engine at 75% and 100% of load must be redundant. Based on the fuel and energy, the strategies of renewable fuel and energy needed to be considered. The device of utilizing exhaust gas energy was calculated and designed in case of the diesel engine working at 90% of engine load and 1500 rpm of engine speed and shown in Figure 4. The redundancy heat energy of exhaust gas would be adjusted to expel into the environment by the by/pass valve. The SCO heating system by utilizing, recovering the exhaust gas energy used for engine start was fabricated and shown in Figure 5.

B. Back pressure in the exhaust pipe line

After calculating and designing the heat exchanger and due to the heat exchanger is placed in the exhaust pipe of diesel engine, therefore the determining of exhaust back pressure is necessary to ensure that the utilizing of the exhaust gas energy by heater to heat up SCO does not effect on the normal exhaust process of diesel engine. The parameters of the resistance in the exhaust pipe while the engine working at 75% and 100% of load are given in the Table 4.

Table 4. Parameter of the resistance in the exhaust pipe

Parameters	**Unit**	**75% Ne**	**100% Ne**
Reynolds of exhaust gas, Re_{eg}		1982	2126
Density of exhaust gas, D_{eg}	kg/m^3	0.491	0.415
Friction coefficient of parallel pipes cluster, ξ		0.015	0.0297
Friction resistance of heater to exhaust gas current, Δp_{fr}	Pa	152	421
Local drag coefficient of parallel pipes cluster, ξ_{ld}		2.5	3.8
Local resistance of heater to exhaust gas current, Δp_{lr}	Pa	39.7	47
Total resistance of heater to exhaust gas current, Δp_{tr}	Pa	191	468
Standard resistance, Δp_{sr}	Pa	1472 ÷ 3924	

Table 4 showed that, total resistance of heater to exhaust gas current satisfies the standard resistance at 75% and 100% of engine load. Therefore, the heater could be used in order to heat up SCO. The SCO heater after calculating and designing was installed on the test-bed. Then, the HSCO was used as fuel in the diesel engines D243. After designing the heat exchanger for recovering exhaust gas energy, it was simulated by ANSYS FLUENT. The simulating results were shown in Figure 6, Figure 7 and Figure 8.

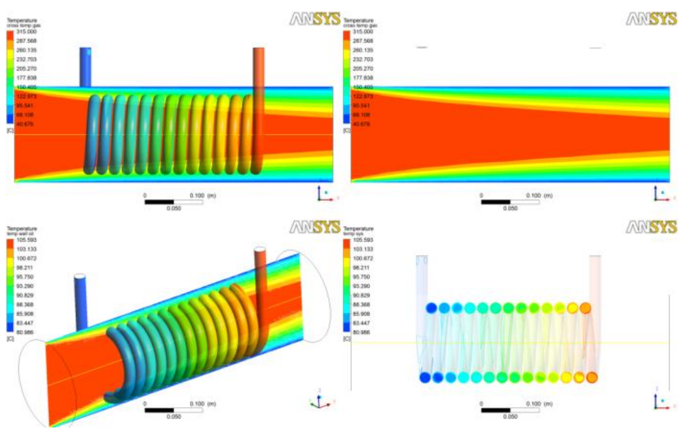

Figure 6a. Inlet and outlet exhaust gas temperature and SCO at 10% of load and 1500 rpm

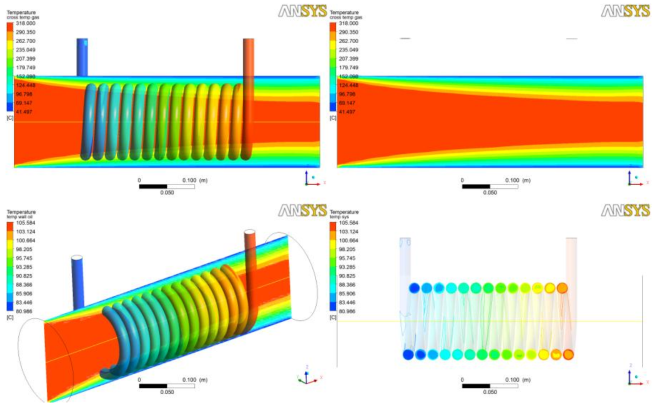

Figure 6b. Inlet and outlet exhaust gas temperature and SCO at 10% of load and 2000 rpm

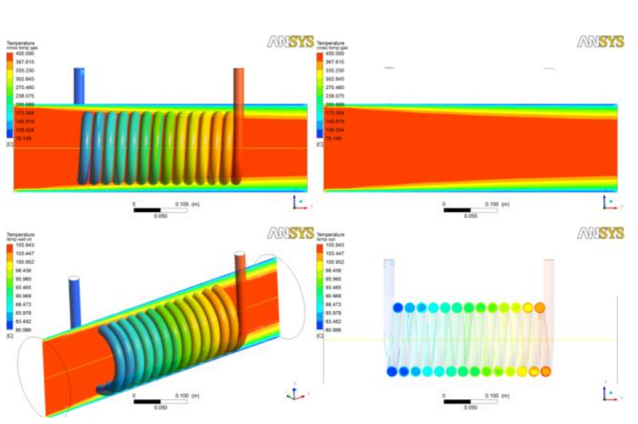

Figure 7a. Inlet and outlet exhaust gas temperature and SCO at 75% of load and 1500 rpm

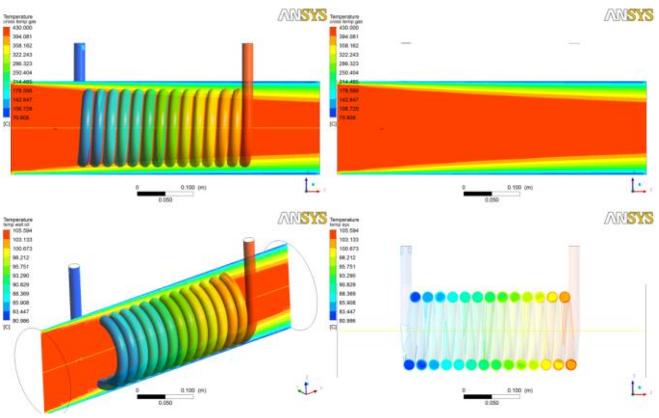

Figure 7b. Inlet and outlet exhaust gas temperature and SCO at 75% of load and 2000 rpm

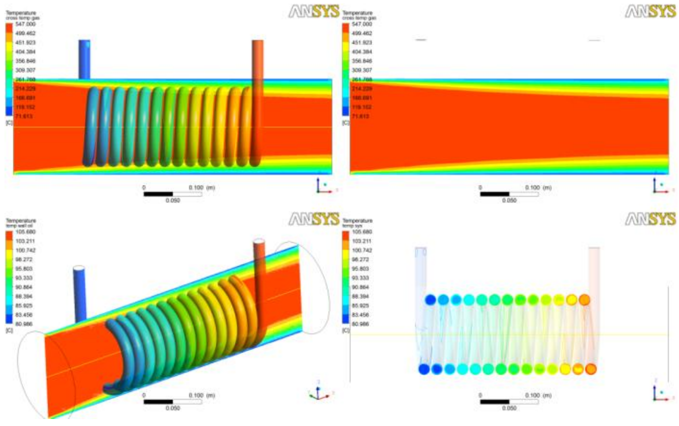

Figure 8a. Inlet and outlet exhaust gas temperature and SCO at 100% of load and 1500 rpm

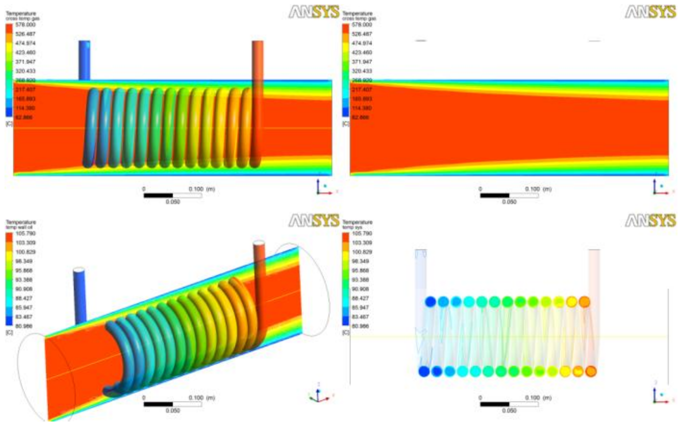

Figure 8b. Inlet and outlet exhaust gas temperature and SCO at 100% of load and 2000 rpm

From Figure 6, Figure 7 and Figure 8, it could be seen that, at 10% of load, the exhaust gas temperature and exhaust gas energy were low, the ability of utilization was still low. SCO fuel temperature out of heater almost constant and exhaust gas temperature was higher than 221ºC and therefore, it was also higher than dew point temperature hence did not cause corrosion of equipment. At 75% and 100% of load with 1500 rpm of engine speed, temperature and higher temperature exhaust gases should be able to take advantage of higher temperatures. Exhaust gas temperature out of from heater was 376ºC and 474ºC, respectively. Besides, at 75% and 100% of load with 2000 rpm of engine speed, exhaust gas temperature out of from heater was 390ºC and 510ºC, respectively. At the low load, the heat energy utilized was lower, so SCO temperature was nearly unchanged. However, at 75% and 100% of load, the temperature of SCO was 103ºC and 105ºC respectively.

The results of calculation and simulation showed that the calculation and design of the heat exchanger for heating up pure vegetable oil did not affect the normal working in intake-exhaust stroke of the engine even as the engine was running at small speed and low load. In addition, the exhaust gas temperature of the engine out of heat exchanger was always higher than the dew-point temperature, so the corrosion of the heat exchanger mounted on the exhaust pipe by acid would not occur. The average error between the experimental and simulation results was 9%, the exhaust gas energy could be optimized when the engine was operated at speed higher than 1500 rpm and over 50% of the load. At low loads, the exhaust gas energy was low due to the low exhaust gas temperature and mass flow, the SCO fuel temperature out of the heat exchanger was almost unchanged. Therefore, to heat SCO and maintain SCO's temperature at the right temperature, electrical energy or traditional diesel fuel should be used in this case. Thus, the SCO heating system by exhaust gas heat could only be fitted with diesel engines using dual fuel systems or integrated electric power.

CONCLUSIONS

Utilization of exhaust gas energy of diesel engine to improve the disadvantages of pure vegetable oil and use heated pure vegetable oil as fuel was necessary to improve the heat efficiency of diesel engine and reduce the environmental pollution. However, up to now, almost exhaust gas energy of diesel engine in the ocean ship with large engine was utilized primarily by heat exchanger such as exhaust boiler, compressor-turbocharger. Meanwhile, in Vietnam, small diesel engines (power <100 HP) on the inland fleet of ship or vessel were very large and the pure vegetable oil source was abundant. Thus, the complex between exhaust gas energy recovery and using pure vegetable oil as fuel for small diesel engines would bring big benefits, it would solve the matters related to energy saving, diversification of fuel source, pollution reduction, economic improvement. The results of this paper will orientate to calculate, design, fabricate and install the heat exchanger to recover the exhaust gas heat of diesel engines in small ship/vessel, generator or agricultural machinery, and convert the small diesel engines into operating with pure vegetable oil. In next research, the effect of temperature on the heat exchanger strength will carry out and the power, emission characteristic of diesel engine fueled with pure vegetable oil will be also mentioned aiming at proving the applicability the bio-oil heating system by exhaust gas energy.

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