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DESIGN PROCEDURE OF MOBILE WASTE HEAT RECOVERY SYSTEM FOR PRODUCTION OF HOT WATER IN AGRICULTURAL APPLICATIONS

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Abstract

Pesticides are widely used in the control of agricultural pests by spraying machines. While a significant part of the fuel energy used by the tractor in pesticide applications is used for the cooling of the engine, the other important part is discharged from the exhaust as heat energy. In this study, the possibilities of controlling agricultural pests by producing superheated water from waste heats at high temperature and flow rates with an integrated production method from cooling and exhaust systems were investigated. The use of this high-value lost energy in the control of agricultural pests is important for reasons such as economic and environmental factors. The water in the engine cooling system circulates at about 100^oC. The waste heat of the exhaust gases can reach up to 600^oC. It is possible to produce superheated water at 150^oC by using both waste heat. For this, some equipment is added to the engine cooling system and the waste energy of the water circulating in the system is transferred to the water used in spraying. This water with a temperature of 100^oC is passed through the heat exchanger placed in the exhaust system and its temperature is increased to 150^oC. In the evaluation made regardless of the time-dose relationship, pests can be struggled with 7-51 Lha^{-1} diesel fuel and 558-875 Lh^{-1} capacity hot water. Considering the time-dose relationship, it is predicted that the fuel consumption will decrease and the temperature and superheated water capacity will increase.

Key words: ecological agriculture, heat transfer, plant protection, sustainability, thermodynamics

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1. Introduction

1.1. Importance of the subject

Increased concern about the widespread use of pesticides in crop production has created an urgent need to develop new environmentally friendly crop protection strategies. The use of soil fumigation in particular is based on highly toxic substances that not only eliminate soil-borne pests and diseases, but also kill most of the saprophyte and beneficial micro flora (Chen et al., 1991; Gamliel, 2000). Due to the regulatory constraints of fumigant use and public resistance, production systems without fumigants need to be developed (Funk et al., 2002). Weeds have the highest potential among all yield loss in crop production. The evolution of herbicide resistant weed populations has become common in weed control regions of the World (Coleman et al., 2019; Heap, 1997). Limitations to herbicide use due to resistance and environmental factors, the lack of new herbicide development and removal of existing chemicals as regulators will create less herbicide options due to the depending on the use of highly effective alternative weed control technologies now and within the foreseeable future in agricultural pest systems (Coleman et al., 2019). Relying on herbicides in weed control is due to their effectiveness as well as the lack of suitable alternatives (Walsh and Powles, 2014). In many countries, the risk of environmental pollution

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and drinking water reservoirs has led to several restrictions on the use of herbicides for weed control (Augustin, 2003; Hansson, 2002; Kristoffersen et al., 2004; Lefevre et al., 2001), this increases the need for alternative control methods.

Using hot water in agricultural pest is not a new invention. This method, the first patent of which was obtained in the 1880s, is an old method, but it is not an outdated method. Hot water or steam machines working with different energy sources are used in the control soil disinfection, weeds, disease and pests. Karcher in Germany, Oeliatec in France, Waipuna System in New Zealand, Aqua Heat in the USA, Multevo in the UK, Heatweed Technologies in Norway, Mantis in Germany, Empas in the Netherlands and companies in other European countries sell machines that produce hot water. These machines work with different energy sources such as LPG and diesel oil. Due to their high costs, the areas of use are limited to the struggle against weeds for landscaping within the municipality boundaries. The high cost of hot water production is due to the need for a unit is that uses fuels such as LPG, diesel oil externally to produce hot water in addition to the fuel consumed by the tractor for the progress of the production, and the costs such as maintenance and repair. Researches are being carried out at European and American universities regarding the application of hot water, which is used at a limited level due to technological inadequacy, for agricultural treatment.

Most plant pathogens and pests are mesophilic and do not tolerate temperatures above 31-32ºC. All soil-borne organisms are normally directly or indirectly heat-inactivated (Katan, 1987; Stapleton and DeVay, 1982). It is claimed that the disinfection in Japan with hot water is highly effective and can last up to three years (Tateya, 2001). Kurfess and Klieisinger (2000), investigated the destruction of weeds in apple orchards at a speed of 6 kmh^{-1} using water at 85-95ºC under German conditions. According to the results obtained, they stated that quite good results were obtained against weeds without harming apple trees. Hot water application, as recommended in the Australian National Phylloxera Management Protocol, has been reported to be effective for disinfection of various phylloxera strains (Clarke et al., 2017). Deadly temperatures for various types of nematodes have been determined by Wageningen University. Kristoffersen et al. (2008), show that hot water application produces the lowest grass cover even with the least treatment frequency.

Water is one of nature's best energy carriers and is usually easily available. This makes the use of hot water a natural choice. Gravity allows the water to reach the lower plant and roots quickly. This increases its effectiveness.

1.2. Original value of the subject

In the studies on the use of engine waste heat, studies have been carried out on the utilization of exhaust gas energy. By placing a single layer heat exchanger in the exhaust system, the possibilities of using it for heating and cleaning purposes by producing hot water at low temperatures such as 30-40ºC and flow have been investigated. No study has been found on the use of waste energy lost in the engine cooling system. In this system, in order to benefit from the waste heat energy of the engine cooling system, a four-way valve is attached to the system and the temperature of the water used during spraying is passed through the engine surfaces and its temperature is increased. Meanwhile, the safe operation of the system is ensured by means of some protective units against temperature and pressure increase. The water with increasing temperature is directed to the heat exchanger in the exhaust system. In order to benefit effectively from the exhaust gas energy, a nested three-layer heat exchanger has been designed. Thus, while achieving high performance from energy, space is saved. With the integration of both systems, the temperature of the spray water at ambient temperature can be raised to 150ºC and superheated water can be produced at a flow rate of 0.205-0243 Ls^{-1} (Eq. 63).

On average, the closest distance of the agricultural lands to the settlements of the producers is 15 minutes. The engine of the tractor reaches its regime temperature in a short time. Hot water can be produced in a very short time by providing energy transfer from the cooling and exhaust waste heat of the fuel used by the tractor while moving towards the field and spraying in the field to the cold water circulating in the system.

This integrated system, which is used to raise water to very high temperatures for use in agricultural struggle, has not been used in any other application, especially in agricultural production. The importance and specificity of this study is that it is an economically preferable environmentalist control system against to pesticides by increasing the water temperature levels with only engine waste heat without using any other energy source.

This system, which produces hot water, can be used in the cultivation of all kinds of agricultural products. It can turn it into an economical and environmentally friendly alternative to pesticides, especially in the cultivation of products grown with high return organic and good agricultural practices. In addition, with the widespread use of organic agriculture in production, the sale of the same product at high prices, growing healthy products, reducing the health sector burden by reducing pesticide-related diseases, creating employment in the sector with the expansion of the system, providing new research opportunities, increasing the export of products without pesticide residue, reducing application costs or it has benefits such as keeping it in balance.

1.3. Objective

Can superheated water with a capacity of 500 Lh^{-1} be produced by adding some equipment to the cooling and exhaust systems of a tractor engine with a power of 75 kW? Can the temperature be raised to 150ºC and above? What are the waste heat potential savings of the fuel used? What are the installation possibilities? Could there be operational difficulties to consider? Can it be effective against weeds and soil pathogens in field applications? What is the effectiveness degree?

2. Material and methods

Two heat exchangers with the same structure are produced for the operation of the system. First, a single heat exchanger is installed in the exhaust pipe to determine the highest temperature level that can be reached with the amount of hot water. Then, without affecting the engine performance, two heat exchangers are attached to the exhaust system and the highest temperature levels can be determined with the highest flow rate. The effective energy dose in the agricultural pest control with hot water can be reduced by using coarse droplets compared to fine droplets, heating the water temperature as high as possible and adding a surfactant to the hot water (Rask and Kristoffersen, 2007). In addition, it is stated that effective results are obtained with narrow hot water application (Hansson and Ascard, 2002). In spray applications of different nozzles, changes are made in water flows and evaluations are made according to the performance values between the results.

Some equipment is added to the tractor engine cooling system to preheat the water intended for use in agricultural pest. With heat exchanger to be installed in the exhaust system, hot water is heated and sent to the spray pipes. 500 L capacity sprayer tank is designed as a water source. It is intended to install a check valve after the valve to prevent back escape by attaching a valve to the sprayer tank outlet. It is suitable to install siliphos filter at the tank outlet to clean the water and prevent calcification. Some of the cooling water, whose temperature increases gradually with the operation of the engine, is directed through the pipe to the 100 L capacity accumulation tank. An expansion tank is placed in the system to prevent pressure drop. Thus, continuous flow is provided in the spray. The remaining hot water is mixed with the cold water coming from the tank via the cooling system via a four-way valve (Eq. 1).

Temperature of the water obtained from the cooling system is calculated as given by Eq. (1):

$$T_{mixture} = \frac{m_{incoming\ engine}\ T_{incoming\ engine} + m_{incoming\ tank}\ T_{incoming\ tank}}{m_{engine} + m_{\ tank}} \tag{1}$$

$$T_{mixture} = \frac{(1.941 \times 100) + (0.139 \times 20)}{2.080}$$

$T_{mixture}$ = 94.654 ~ 95ºC

In today's engines, thermostats start to open at 80ºC and fully open at 110 ºC (Megep, 2011b). The drop in water temperature in the radiator is ΔT_{water} 7-8 ºC, the water circulation in the cooling system is 7-12 times per minute and 5 ºC in high-speed engines. Accordingly, the engine input and output temperature differences in the system occur as in Eq. (2).

$$\Delta T_{water} = T_{incoming\ engine} - T_{mixture} \tag{2}$$

ΔT_{water} = 100-95 = 5ºC

In order to ensure the safe operation of the tractor engine without being affected by temperature and pressure changes, the temperature of the mixing water is adjusted to 95ºC by means of a four-way valve, pressure and temperature sensors and by-pass arrangement, and the engine is operated at regime temperature. In the engine with increasing temperature, the coolant is connected to the main inlet end of the four-way valve. The water connection in the tank used for spraying is connected to the other main inlet end of the four-way valve. Both waters are mixed in a four-way valve by keeping the temperature constant at 95ºC. The mixing water is sent to the engine cooling system inlet and takes the heat from the engine surfaces and raises its temperature to 100ºC. Thus, the cold water supplied to the cooling system from outside is taken back at 100ºC.

The other end of the four-way valve is connected to the radiator inlet end and the transfer of the remaining water in the system is provided. In order to ensure safe operation of the engine by preventing the high temperature and pressure that may occur, an additional bay-pass pipe is connected to the radiator before the four-way valve. Temperature and pressure sensors and their control device are used to automatically control the bypass path. In case of excessive increase in these values, safe operation is ensured by passing the water from the bay-pass channel to the radiator by controlling the solenoid valve of the system. Hot water obtained from the system is directed to the accumulation tank (Fig. 1). By installing a pump and regulator at the accumulation tank outlet, the desired 500 kPa operating pressure and 0.243 Ls^{-1} flow are obtained and spraying is provided. The pressurized hot water is directed to the copper pipe inlet end inside the heat exchanger placed on the exhaust pipe.

A check valve is placed at the inlet end to prevent water escaping from the accumulation tank. The high heat of the exhaust gases passing by licking the outer surface of the spiral copper pipes in the heat exchanger is transferred to the counter flow water at about 100ºC. Thus, in addition to the heat that the water previously received from the engine surface, the temperature level is raised up to 150ºC at the heat exchanger outlet. According to the steam tables, hot water at 500 kPa pressure is heated without changing the phase in closed system. Accordingly, the water passing through the heat exchanger can be heated without any phase change. An expansion tank is added to prevent pressure drop in the system. The water brought to high temperature is directed to the pipes for spraying. 500 kPa pressure sensor on the spray pipe main line controls the solenoid valve and the water flows from the spray nozzles (Fig. 1). The three-layer coil heat exchanger is prepared to transfer the exhaust gas waste energy to the hot water coming from the engine cooler.

In order to utilize engine exhaust gas waste heat, 10 mm diameter, 1.00 mm wall thickness inner structure 0.75 x 2 = 1.5 m long copper pipe coil helical copper structure 150 mm diameter 0.85 x 2 = 1.6 m long black draw pipe placed inside. Two heat exchangers produced in the same structure are attached to the system single and double. Accordingly, the upper and lower limit parameters that can be obtained in hot water production are determined. The smoke area of the tractor exhaust pipe with a diameter of 63 mm is 3115.67 mm^2.

The area of exhaust gas passing through the heat exchanger is as in Eq. (3) (Fig. 2).

$$0.785\times(R_1^2+(R_2^2-R_1^2)+(R_3^2-R_2^2)+(R_4^2-R_3^2)) \quad (3)$$

Heat exchanger area is:

$$0.785\times(841+3800+7120+10179)=17222.9 \text{ mm}^2$$

The heat exchanger will not have any negative effects on the engine performance related to the exhaust outlet (Eq. 3). By installing an adjustable air intake nozzle on the exhaust pipe, higher temperatures can be achieved by ensuring that the raw fuel discharged from the engine continues to burn in the exhaust pipe.

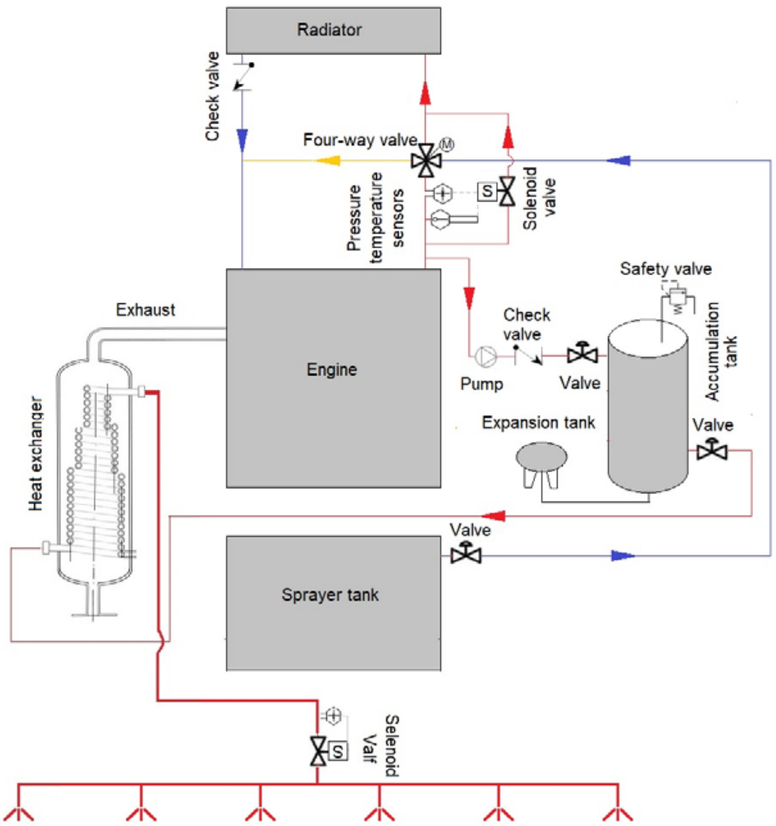


Fig. 1. The operating scheme of the system

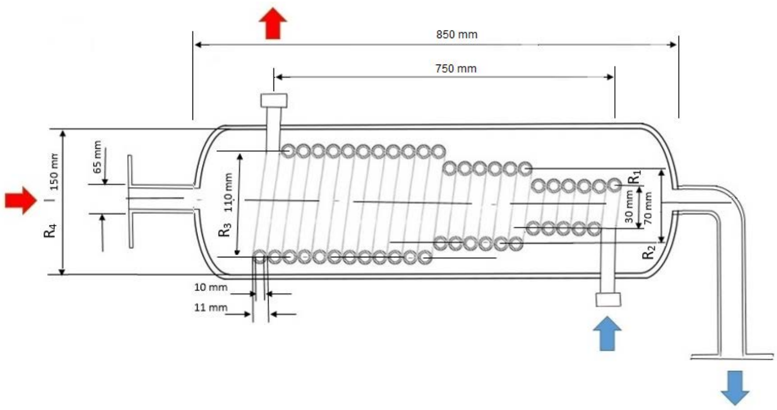


Fig. 2. Heat exchanger used in the experimental setup

3. Results and discussion

Approximately 22% to 40% of the total heat supplied to the engine as fuel is converted into useful mechanical work. Residual heat is released into the environment through exhaust gases and engine cooling systems. Serious and concrete effort is required to conserve this energy with exhaust heat recovery techniques (Balcı, 2011; Jadhao et al., 2011; Megep, 2011a; Morgan et al., 2016; Wilson et al., 2017; Zorin and Vaida, 2011). The temperatures of the waste heat range from 200ºC surface temperatures to 600 ºC gas temperatures. Therefore, the size of the energy currently waste is important and there is a great opportunity to use this waste heat for productive purposes. Vehicle waste heat is generally two times higher than the mechanical output of the engine. An engine running at 30% thermal efficiency releases the remaining 70% of fuel energy as coolant, exhaust gases and engine compartment waste heat. During a typical driving cycle, the efficiency of the engine is lower than the maximum efficiency, and since this operating efficiency decreases (for example 30% - 15%), the size of the waste heat increases, thus creating a greater energy potential for cooling (Pandiyarajan et al., 2011; Sayin et al., 2006).

The exhaust gas flow from the engine must be calculated to estimate possible hot water production and heat exchanger selection. To know the actual amount of exhaust gases, the actual air requirement and the theoretical air requirement to get more air is determined. Theoretical amount of exhaust gas is determined with extreme air knowledge.

Energy balance for the engine (Eq. 4):

$$\sum \dot{I}nput = \sum Output \tag{4}$$

According to Kolchin and Demidov (1984), fuels used in diesel engines are fuels consisting of hydrocarbon mixtures. The fuel used in a diesel engine has the following average basic components. The mass content of diesel fuel is 87% C, 12.6% H and 0.004% O (the amount of sulfur (S) and water vapor (W) is taken as zero). In the calculations of the calorific value of the fuel *Hu* is the lower calorific value of the fuel used and is determined by the Mendeleev formula given below (Eq. 5).

Heat Analysis for fuel:

$$H_u = [33.91C + 125.6H - 10.89(S - O) - 2.51(9H + W)]10^3 \tag{5}$$

$$H_u = \begin{bmatrix} (33.91)(0.870) + (125.6)(0.126) - 10.89(0.004 - 0) \\ -2.51(9(0.126 + 0 \end{bmatrix} 10^3$$

$H_u =$ 42.440 kJ kg^{-1}

The average power of tractors made of intensive agriculture in Turkey is reported to be 74-75 kW (ÇŞB, 2019; Korucu et al., 2015). Arrangements have been made by Kolchin and Demidov (1984), on the working processes and features of an overfilled turbo diesel engine. This study is based on these fixed data. Calculations were continued with the features of Tümosan 4DT-39l-105C four-stroke (τ = 4) four-cylinder (i = 4) combustion chamber (Turbo) Tractor engine with volumetric diesel engine. It has been neglected because the intercooler has little effect on the results.

While only air (in diesel engines) is used as work gases in cylinders in suction and compression processes, combustion product compositions are used in combustion, expansion and exhaust processes.

Amount of air:

Complete combustion of a mass or unit volume fuel unit requires a certain amount of air, called the theoretical air requirement, and is determined by the final fuel composition. Theoretically, the amount of air required for the combustion of 1 kg of diesel fuels (Eq. 6):

$$l_o = \frac{1}{0.23}(C\frac{8}{3} + 8H - O) \tag{6}$$

$$l_o = \frac{1}{0.23}(0.870\frac{8}{3} + (8 \times 0.126) - (0.004))$$

$l_o = 14.452$ kg air (kg fuel)$^{-1}$

Theoretically, the amount of air required to combustion 1 kmol of fuel for diesel fuels (Eq. 7):

$$L_o = \frac{1}{0.208}(\frac{C}{12} + \frac{H}{4} - \frac{O}{32}) \tag{7}$$

$$L_o = \frac{1}{0.208}(\frac{0.870}{12} + \frac{0.126}{4} - \frac{0.004}{32})$$

$L_o = 0.500$ kmol air (kg fuel)$^{-1}$

Reducing the Air excess factor (λ) to the permissible limits reduces cylinder size and therefore increases engine power per liter. However, it worsens the heat stresses of the engine, which is especially true for piston group parts and adds to the smoky exhaust. The best modern turbocharged diesel engines with jet injection run smoothly without overheating at λ = 1.6 to 1.8 as material. Accordingly, for a turbocharged diesel engine λ= 1.7. The amount of fresh air taken into the cylinder (Eq. 8):

$$M_1 = \lambda L_o = 1.7 \times 0.500 \tag{8}$$

$M_1 = 0.85$ kmol mixture(kg fuel)$^{-1}$

Amounts of ingredients found in combustion products (Eq. 9):

$$M_2 = M_{CO_2} + M_{H_2O} + M_{O_2} + M_{N_2} \tag{9}$$

The mean elemental composition of diesel fuels in fraction of total mass (kg) are C =0.870, H=0.126, O= 0.004 (Kolchin and Demidov, 1984)

$$M_{CO_2} = \frac{C}{12} = \frac{0.87}{12} = 0.0725 \text{ kmol CO}_2\text{(kg fuel)}^{-1}$$

$$M_{H_2O} = \frac{H}{2} = \frac{0.126}{2} = 0.063 \text{ kmol H}_2\text{O(kg fuel)}^{-1}$$

$$M_{O_2} = 0.208(\lambda - 1)L_0 = 0.208 \times (1.7 - 1) \times 0.500 = 0.0728$$
kmol H_2O (kg fuel)$^{-1}$

$$M_{N_2} = 0.792\lambda L_0 = 0.792 \times 1.7 \times 0.500 = 0.6732$$
kmol H_2O(kg fuel)$^{-1}$

$$M_2 = \frac{C}{12} + \frac{H}{2} + 0.208(\lambda - 1)L_o + 0.792\lambda L_o$$

$$M_2 = 0.8815 \text{ kmol mixture (kg fuel)}^{-1}$$

Atmospheric pressure for turbocharged diesel engines (Eq. 10):

When an engine is operating with no supercharging, atmospheric air enters the cylinder (P_o=0.1 MPa; T_o=293 K). When tractor engines are supercharged the ambient pressure and temperature are assumed to equal the compressor outlet pressure P_c and temperature T_c.

$$P_c = P_o \times 1.7 = 0.17 \text{ MPa} \tag{10}$$

Ambient temperature for turbocharged diesel engines (Eq. 11):

$$T_c = T_o \left(\frac{P_c}{P_o}\right)^{\frac{n_c - 1}{n_c}} \tag{11}$$

$$T_c = 293 \times \left(\frac{0.17}{0.1}\right)^{\frac{1.65-1}{1.65}}$$

$$T_c = 361 \text{ K}$$

(n_c= Polytropic index of air compression in the compressor (supercharger) with a cooler is taken 1.65)

The high compression ratio of a turbocharged diesel engine increases the residual gas temperature and pressure (T_r and P_r). When the engine temperature rises and increases the T_r and P_r values, it is evaluated as T_r = 800 K, P_r = 0.95, P_c = 0.95 x 0.17 = 0.162 MPa while overcharging. A high compression ratio of an unsupercharged diesel engine reduces the residual gas temperature and pressure, while an elevated engine speed somewhat increases the values of T_r and P_r. When supercharging, the engine temperature rises and increases the values of T_r and P_r. Therefore, we may assume that with supercharging T_r= 800 K, P_r= 0.95xP_c, P_r= 0.95x0.17=0.162 MPa. The natural preheating of the fresh air intake in the turbocharged diesel engine grows less due to a decrease in the temperature difference between the engine components and the overcharged air. For this reason, ΔT = 10ºC in this study it was handled as.

Fresh air intake density (Eq. 12):

$$\rho_c = \frac{P_c 10^6}{R_a T_c} \tag{12}$$

R_a= The gas specific constant of air (Eq. 13):

$$R_a = \frac{R}{\mu_a} \tag{13}$$

$$R_a = \frac{8315}{28.96}$$

$$R_a = 287 \text{ J (kg}^{o}\text{C)}^{-1}$$

$$\rho_c = \frac{0.17 \times 10^6}{287 \times 361}$$

$$\rho_c = 1.641 \text{ kgm}^{-3}$$

Engine intake process:

Engine inlet pressure losses, the pressure losses ΔP_a due to the resistance in the suction system and the decreasing velocity in the cylinder, can be determined with a certain assumption by the Bernoulli equation (Eq. 14).

$$\Delta P_a = \frac{(\beta^2 + \xi_{in})(\omega_{in})\rho_c 10^{-6}}{2} \tag{14}$$

$$\Delta P_a = \frac{2.7 \times 70 \times 1.641 \times 10^{-6}}{2}$$

$$\Delta P_a = 0.011 \text{ MPa}$$

It is taken by assuming that the diesel engine inlet manifold resistances in accordance with the engine speed are small in both turbo and non-turbo engines. Accordingly, it takes $(\beta^2 + \xi_{in})$=2.7 and (ω_{in}) = 70 ms^{-1}.

Pressure at the end of the intake process (Eq. 15):

$$P_a = P_c - \Delta P_a \tag{15}$$
$$P_a = 0.17 - 0.011$$
$$P_a = 0.159 \text{ MPa}$$

Residual gas coefficient (Eq. 16):

$$\gamma_r = \frac{T_c + \Delta T}{T_r} \frac{P_r}{\varepsilon P_a - P_r} \tag{16}$$

$$\gamma_r = \frac{361 + 10}{800} \frac{0.162}{17 \times 0.159 - 0.162}$$

$\gamma_r = 0.030$

Temperature at the end of intake (Eq. 17):

$$T_a = \frac{T_c + \Delta T + \gamma_r T_r}{1 + \gamma_r} \tag{17}$$

$$T_a = \frac{361 + 10 + 0.030 \times 800}{1 + 0.030}$$

$$T_a = 384 \text{ K}$$

Volumetric efficiency (Eq. 18):

$$\eta_v = \frac{T_c(\varepsilon P_a - P_r)}{(T_c + \Delta T)(\varepsilon - 1)P_c} \tag{18}$$

$$\eta_v = \frac{361 \times (17 \times 0.159 - 0.162)}{(361 + 10)(17 - 1) \times 0.17}$$

$$\eta_v = 0.909$$

Compression process in engine:

For the compression process is fairly fast (0.015-0.005 s in design condition), the overall heat exchange between the working medium and the cylinder walls during the compression process remains negligible and the value of n_1, may be evaluated by the mean specific heat ratio k_1. When the diesel engine is operated under design conditions, we can run the compression polytropic index n_1 roughly equal to k_1. In the average compression process, the coefficients of adiabatic and polytropic indices for ε = 17 and T_a = 384 K values in turbo charged diesel engines are close to each other according to the graph (Kolchin and Demidov, 1984; k_1=1.3615 and n_1= 1.362).

Pressure at the end of compression (Eq. 19):

$$P_{co} = P_a \varepsilon^{n_1} \tag{19}$$

$$P_{co} = 0.159 \times 17^{1.362}$$

$$P_{co} = 7.538 \text{ MPa}$$

Temperature at the end of compression (Eq. 20):

$$T_{co} = T_a \varepsilon^{n_1 - 1} \tag{20}$$

$$T_{co} = 384 \times 17^{1.362-1}$$

$$T_{co} = 1071 \text{ K}$$

At the end of compression:

The ratio of the amount of heat given to an environment to its temperature change is called the average heat capacity (specific heat) of medium. The value of the heat capacity depends on the temperature, pressure, physical properties of the environment and the nature of the transaction. Constant volume (mc_v) and constant pressure (mc_p) kJ (kmol °C)$^{-1}$ average molar heat capacities are usually used to calculate the operating processes of engines. The relationship between these values (Eq. 21):

$$mc_p - mc_v = 8.315 \tag{21}$$

Formulas, tables or graphs are used to determine the average molar heat capacity of various gases with respect to temperature.

In the pressure ranges that occur in automobile and tractor engines, the effect of pressure on average molar heat capacities is neglected.

(a) average molar specific heat for air (Kolchin and Demidov, 1984; Eqs. 22-23):

$$(mc_v)_{t_0}^{t_c} = 20.6 + 2.638 \times 10^{-3} t_c \tag{22}$$

$$t_c = T_{co} - 273 \tag{23}$$

t_c = 798°C

$$(mc_v)_{t_0}^{t_c} = 20.6 + 2.638 \times 10^{-3} \times 798$$

$$(mc_v)_{t_0}^{t_c} = 22.705 \text{ kJ(kmol °C)}^{-1}$$

(b) Average molar specific heat for the residual gas (Kolchin and Demidov, 1984; Eq. 24), an intermediate value for ε = 17 and t_c = 798 °C in a turbocharged engine.

$$(mc''_v)_{t_0}^{t_c} = 24.386 \text{ kJ (kmol °C)}^{-1} \tag{24}$$

(c) Average molar specific heat of the working environment (Eq. 25):

$$(mc'_v)_{t_0}^{t_c} = \left(\frac{1}{1+\gamma_r}\right)\left[(mc_v)_{t_0}^{t_c} + \gamma_r (mc''_v)_{t_0}^{t_c}\right] \tag{25}$$

$$(mc'_v)_{t_0}^{t_c} = \left(\frac{1}{1+0.03}\right)\left[22.705 + 0.03 \times 24.386\right]$$

$$(mc'_v)_{t_0}^{t_c} = 22.754 \text{ kJ(kmol °C)}^{-1}$$

Combustion process in the engine.

Molecular change coefficient of fresh mix (Eq. 26):

$$\mu_o = \frac{M_2}{M_1} \tag{26}$$

$$\mu_o = \frac{0.8815}{0.85}$$

$$\mu_o = 1.037$$

Molecular change coefficient of work mix (Eq. 27):

$$\mu = \frac{\mu_o + \gamma_r}{1 + \gamma_r} \tag{27}$$

$$\mu = \frac{1.037 + 0.03}{1 + 0.03}$$

$$\mu = 1.036$$

The heat of combustion of the work mixture (Eq. 28):

$$H_{w.m} = \frac{H_u}{M_1(1 + \gamma_r)} \tag{28}$$

$$H_{w.m} = \frac{42440}{0.85 \times (1 + 0.03)}$$

$$H_{w.m} = 48480 \text{ (k* kmol}^{-1}\text{)}$$

Average molar specific heat of combustion products (Kolchin and Demidov, 1984; Eq. 29):

$$(mc_v^{''})_{t_0}^{t_c} = (\frac{1}{M2})(M_{CO_2}((mc_{vCO_2}^{''})_{t_0}^{t_z} + M_{H_2O}((mc_{vH_2O}^{''})_{t_0}^{t_z}$$
$$+ M_{O_2}((mc_{vO_2}^{''})_{t_0}^{t_z} + M_{N_2}((mc_{vN_2}^{''})_{t_0}^{t_z} \tag{29}$$

$$(mc_v^{''})_{t_0}^{t_c} = (\frac{1}{0.8815}) \times (0.0725 \times (39.123 + 0.003349 t_z) + 0.063 \times$$
$$(26.67 + 0.004438 t_z) + 0.0728 \times (23.723 + 0.0155 t_z) +$$
$$0.6732 \times (21.951 + 0.001457 t_z)$$

$$(mc_v^{''})_{t_0}^{t_c} = 23.847 + 0.00183 t_z$$

$$(mc_p^{''})_{t_0}^{t_c} = (mc_v^{''})_{t_0}^{t_c} + 8.315$$

$$(mc_v^{''})_{t_0}^{t_c} = 23.847 + 0.00183 t_z + 8.315$$

$$(mc_v^{''})_{t_0}^{t_c} = 32.162 + 0.00183 t_z$$

Temperature at the end of the combustion processcan be calculated with Eqs. (30-31):

$$\xi_z H_{w.m} + \left[(mc_v^{'})_{t_0}^{t_c} + 8.315\alpha \right] t_c + 2270(\alpha - \mu) = \mu (mc_p^{''})_{t_0}^{t_c} t_z \tag{30}$$

The pressure increase in the diesel engine mainly depends on the amount of fuel consumption per cycle. In order to reduce the gas-induced stresses of the crank gear parts, it is recommended to have a maximum combustion pressure of not more than 11-12 MPa. Accordingly, it is recommended to take α = 1.5 for an overloaded diesel engine ($\alpha = P_z/P_{co}$ =1.5 pressure increase; 8.315x273=2270). Jet injection diesel engines performed well in the modern open combustion chamber. For an overloaded diesel engine that creates better combustion conditions due to an increase in engine heat release rate, the heat use coefficient can be taken as ξ_z = 0.86. The combustion equation contains two unknowns. At the temperature (t_z) formed at the end of combustion, constant volume $(mc_v^{'})_{t_0}^{t_c}$ or at constant pressure $(mc_p^{''})_{t_0}^{t_c}$ are the specific heats of the combustion products. Defining $(mc_v^{'})_{t_0}^{t_c}$ or $(mc_p^{''})_{t_0}^{t_c}$ according to Kolchin and Demidov (1984) data can be solved for t_z. When $(mc_v^{'})_{t_0}^{t_c}$ or $(mc_p^{''})_{t_0}^{t_c}$ is determined by approximate formulas Kolchin and Demidov (1984), the combustion equations take the form of a quadratic equation after changing their numerical values for all known parameters and subsequent conversions.

$$At_z^2 + Bt_z - C = 0 \tag{31}$$

where A, B and C are numerical values of known quantities and t_z:

$$t_z = \frac{(-B + \sqrt{B^2 + 4AC}}{2A}$$

Eq. (30) according to:

$$0.86 \times 48480 + [22.754 + 8.315 \times 1.5] \times 798 +$$
$$2270 \times (1.5 - 1.036) = 1.036 \times (32.162 + 0.00183 t_z) t_z$$

or Eq. (31) according to:

$$0.001896\, t_z^2 + 33.320\, t_z - 70860 = 0$$

$$t_z = \frac{(-33.32 + \sqrt{33.32^2 + 4 \times 0.001896 \times 70860}}{2 \times 0.001896}$$

$$t_z = 1919\ ^{o}\text{C}$$
$$T_z = t_z + 273$$
$$T_z = 2192 \text{ K}$$

Maximum combustion pressure (Eq. 32):

$$P_z = \alpha P_{co} \tag{32}$$
$$P_z = 1.5 \times 7.538$$
$$P_z = 11.307 \text{ MPa}$$

Expansion process in the engine:

Pre-expansion and rate (Eq. 33):

$$\varsigma = \frac{\mu T_z}{\alpha T_c} \tag{33}$$

$$\varsigma = \frac{1.036 \times 2192}{1.5 \times 1071}$$

$$\varsigma = 1.41$$

Post-expansion rate is given by Eq. (34):

$$\delta = \frac{\varepsilon}{\varsigma} \tag{34}$$

$$\delta = \frac{17}{1.41}$$

$$\delta = 12.06$$

Average expansion adiabatic and polytropic indices for diesel engines can be taken by considering a rather large cylinder size slightly smaller than the adiabatic index of expansion determined (for Kolchin and Demidov, 1984). For δ= 12.06, T_z = 2192K and λ = 1.7 in a turbocharged diesel engine; k_2 = 1.2792 and n_2 = 1.267.

Pressure at the end of expansion process (Eq. 35):

$$P_b = \frac{P_z}{\delta^{n_2}} \tag{35}$$

$$P_b = \frac{11.307}{12.06^{1.267}}$$

$$P_b = 0.482 \text{ MPa}$$

Temperature at the end of expansion process (Eq. 36):

$$T_b = \frac{T_z}{\delta^{n_2-1}} \tag{36}$$

$$T_b = \frac{2192}{12.06^{1.267-1}}$$

$$T_b = 1129 \text{ K}$$

Work cycle parameters.

Theoretical mean indicated pressure (Eq. 37):

$$P_i^{'} = \frac{P_{co}}{\varepsilon - 1}\left[\alpha(\varsigma - 1) + \frac{\alpha\varsigma}{n_2 - 1}(1 - \frac{1}{\delta^{n_2} - 1}) - \frac{1}{n_1 - 1}(1 - \frac{1}{\varepsilon^{n_1} - 1})\right] \tag{37}$$

$$P_i^{'} = \frac{7.538}{17-1}\left[\begin{array}{l} 1.5\times(1.41-1) + \frac{1.5\times 1.41}{1.267-1}\ (1 - \frac{1}{12.06^{1.267-1}}) \\ - \frac{1}{1.362-1}\ (1 - \frac{1}{17^{1.362}-1}) \end{array}\right]$$

$$P_i^{'} = 1.266 \text{ MPa}$$

Eq. (38) indicates the verage pressure. A decrease in the theoretical mean is evaluated by the value of the design process diagram rounding coefficient (φ_r) as a result of the actual process being separated from the design process. Diagram rounding coefficient φ_r = 0.95.

$$P_i = \varphi_r P_i^{'} \tag{38}$$

$$P_i = 0.95 \times 1.266_i^{'}$$

$$P_i = 1.203 \text{ MPa}$$

Indication efficiency (Eq. 39):

$$\eta_i = \frac{P_i l_o \lambda}{H_u \rho_c \eta_v} \tag{39}$$

$$\eta_i = \frac{1.203 \times 14.452 \times 1.7}{42.44 \times 1.641 \times 0.909}$$

$$\eta_i = 0.467$$

Appropriate engine speeds (Eq. 40) at which the power (Eq. 47) required for hot water application work to be performed with the tractor are obtained, the average pressure of mechanical losses (Eq. 41), average effective pressure and mechanical efficiency (Eqs. 42-43), heat efficiency and effective specific fuel consumption (Eqs. 44-45), cylinder sizes (Eq. 46) must be determined. The tractor is in a state of rest in the field in idle speed in agricultural spraying. While the tractor is spraying, it is not operated for a long time at high engine speeds. The system produces hot water when operating is the engine at idle and high speeds. However, in these periods, the tractor does not produce work in the field. The operating parameters for determining the economical fuel consumption were determined for the engine revolutions at which the tractor mostly operates while performing work, in the range of 1500-1900 min^{-1} (Eqs. 48-51).

Piston average speed is expressed by Eq. (40):

$$v_{p.m} = \frac{Sn}{30000} = \frac{115 \times 2300}{30000} = 8.817 \text{ ms}^{-1} \tag{40}$$

The mean pressure of mechanical losses (Eq. 41):

$$p_m = 0.089 + 0.0118 v_{p.m} = 0.089 + 0.0118 \times 8.817 = 0.193 \text{ MPa} \tag{41}$$

Average effective pressure and mechanical efficiency (Eqs. 42-43):

$$p_e = P_i - P_m = 1.203 - 0.193 = 1.01 \text{ MPa} \tag{42}$$

$$\eta_m = \frac{P_e}{P_i} = \frac{1.01}{1.203} = 0.840 \tag{43}$$

Heat efficiency and effective specific fuel consumption (Eqs. 44-45):

$$\eta_e = \eta_i \eta_m = 0.467 \times 0.840 = 0.392 \tag{44}$$

$$g_e = \frac{3600}{H_u \eta_e} = \frac{3600}{42.44 \times 0.392} = 216.392 \text{ g (kW)} \tag{45}$$

Engine capacity is given by Eq. (46):

$$V_l = \frac{\pi B^2 S \tau}{4 \times 10^6} \quad (46)$$

$$V_l = \frac{3.14 \times 104^2 \times 115 \times 4}{4 \times 10^6} = 3.906 \text{ L}$$

Engine power nominal (Eq. 47):

$$N_e = \frac{P_e V_l n}{30\tau} \quad (47)$$

$$N_e = \frac{1.01 \times 3.906 \times 2300}{30 \times 4} = 75.614 \text{ kW}$$

Fuel consumption (Eq. 48):

$$G_f = N_e g_e \quad (48)$$

$$G_f = 75.614 \times 0.216 = 16.333 \text{ kgh}^{-1}$$

Total amount of heat supplied to the engine as fuel (Eq. 49):

$$Q_o = \frac{H_u G_f}{3.6} = \frac{42440 \times 16.333}{3.6} = 192547.92 \text{ Js}^{-1} \quad (49)$$

Engine power (1500-1900 min^{-1}) (Eq. 50):

$$N_{e1500-1900} = N_e \frac{n_x}{n_e}\left[0.87 + 1.13\frac{n_x}{n_e} - \left(\frac{n_x}{n_e}\right)^2\right] \quad (50)$$

$$N_{e1500-1900} = 75.614 \times \frac{1500}{2300}\left[0.87 + 1.13\frac{1500}{2300} - \left(\frac{1500}{2300}\right)^2\right]$$

$$N_{e1500-1900} = 58.270 \text{ - } 70.026 \text{ kW}$$

Effective specific fuel consumption (1500-1900 min^{-1}) (Eq. 51):

$$g_{e1500-1900} = g_e\left[1.55 - 1.55\frac{n_x}{n_e} + \left(\frac{n_x}{n_e}\right)^2\right] \quad (51)$$

$$g_{e_{1500-1900}} = 216 \times \left[1.55 - 1.55\frac{1500}{2300} + \left(\frac{1500}{2300}\right)^2\right]$$

$$g_{e_{1500-1900}} = 208.324 g(kWh)^{-1} - 205.629 \text{ g(kWh)}^{-1}$$

Hourly fuel consumption (1500-1900 min^{-1}) (Eq. 52):

$$G_{f1500-1900} = 10^{-3} g_{e1500-1900} N_{e1500-1900} \quad (52)$$

$$G_{f_{1500-1900}} = 10^{-3} \times 208.324 \times 58.270$$

$$G_{f_{1500-1900}} = 12.139 - 14.447 \text{ kgh}^{-1}$$

m_w is the amount of water required to achieve ΔT temperature rise per unit time. The engine runs at different speeds in the field conditions. It is necessary to know the data specified in Eqs. 53-61 in order to determine the amount of superheated water produced while the engine is working in field conditions. With the engine operating at 1500-1900 min^{-1} cycle, the system can produce 558-875 Lh^{-1} (Eq. 62, Eq. 65) hot water.

Total amount of heat supplied to the engine as fuel (1500-1900 min^{-1}) (Eq. 53):

$$Q_{o1500-1900} = \frac{H_u G_{f1500-1900}}{3.6} \quad (53)$$

$$Q_{o1500-1900} = \frac{42440 \times 12.139}{3.6}$$

$$Q_{o1500-1900} = 143105.322 - 170314.078 \text{ Js}^{-1}$$

Heat equivalent to effective work per second (1500-1900 min^{-1}) (Eq. 54):

$$Q_{e1500-1900} = 1000 N_{e1500-1900} \quad (54)$$

$$Q_{e1500-1900} = 1000 \times 58.270$$

$$Q_{e1500-1900} = 58270 - 70026 \text{ Js}^{-1}$$

Heat transferred to cooling (1500-1900 min^{-1}) (Eq. 55):

$$Q_{c1500-1900} = C\tau B^{1+2m} n^m \left(\frac{1}{\lambda}\right) \quad (55)$$

$$Q_{c1500-1900} = 0.53 \times 4 \times 10.4^{1-2\times0.68} \times 1500^{0.68} \times \left(\frac{1}{1.7}\right)$$

$$Q_{c1500-1900} = 41252.929 - 53145.093 \text{ Js}^{-1}$$

The temperature of residual gases (Eq. 56):

$$T_r = \frac{T_b}{\sqrt[3]{\frac{P_b}{P_r}}} \quad (56)$$

$$T_r = \frac{1129}{\sqrt[3]{\frac{0.482}{0.162}}} = 786 \text{ K}$$

Exhaust gas temperature (Eq. 57):

$$t_r = T_r - 273 \quad (57)$$

$$t_r = 786 - 273 = 513\,^{o}\text{C}$$

Exhaust gas heat (1500-1900 min^{-1}) (Eq. 58):

$$Q_{r1500-1900} = \frac{G_{f1500-1900}}{3.6}\left[M_2 (mc_p^{"})_{t_0}^{t_r} t_r - M_1 (mc_p)_{t_0}^{t_c} t_c\right] \quad (58)$$

$$Q_{r1500-1900} = 40841.362 - 48606.571 \text{ Js}^{-1}$$

$(mc_p^{"})_{t_0}^{t_r} = (mc_v^{"})_{t_0}^{t_r} + 8.315 = 23.290 + 8.315 = 31.605$ kJ (kmol°C)$^{-1}$ (59)

$(mc_v^{"})_{t_0}^{t_r} = 23.290$ (Kolchin and Demidov, 1984), λ=1.7, t_r=513°C it was obtained by finding intermediate value for.

$(mc_p)_{t_0}^{t_c} = (mc_v)_{t_0}^{t_c} + 8.315 = 20.829 + 8.315 = 29.194$ kJ (kmol°C)$^{-1}$ (60)

$(mc_p)_{t_0}^{t_c} = 20.829$ (Kolchin and Demidov, (1984) λ=1.7, t_c=88 °C it was obtained by finding intermediate value for (air column).

Total waste heat (1500-1900 min^{-1}) (Eq. 61):

$$Q_{tot1500-1900} = Q_{c1500-1900} + Q_{r1500-1900} \quad (61)$$
$$Q_{tot1500-1900} = 41252.929 + 40841.362$$
$$Q_{tot1500-1900} = 82094.291 - 101751.664 \text{ kW}$$

The amount of water raised to high temperature (1500-1900 min^{-1}) (Eq. 62):

$$m_{1500-1900} = \frac{Q_{tot_{1500-1900}}}{\rho_w c_{pw} \Delta T} \quad (62)$$

$$m_{1500-1900} = \frac{82 \times 3600}{972 \times 4.187 \times (150-20)}$$

$$m_{1500-1900} = 558 - 653 \text{ Lh}^{-1}$$

where: ρ_w is the specific mass of the water and c_{pw} is the specific heat (for the average temperature 80 °C).

The amount of water circulating in the radiator (Eq. 63):

$$Q_c = m_c c_{pc} \Delta T_m = m_c c_{pc} (T_{h_o} - T_{h_i}) \quad (63)$$

$$m_{c1500-1900} = \frac{Q_{c1500-1900}}{c_{pc}(T_{h_o} - T_{h_i})}$$

$$m_{c1500-1900} = \frac{41252.929}{4201.4(100-95)} = 1.964 \text{ kgs}^{-1}$$

$$m_{c1500-1900} = m_{1500-1900} \rho_l$$

$$m_{1500-1900} = \frac{m_c}{\rho_l}$$

$$m_{1500-1900} = \frac{1.964}{0.965}$$
$$m_{1500-1900} = 2.035 - 2.320 \text{ Ls}^{-1}$$

Flow rate of the water to which heat transfer is made in the heat exchanger (Eq. 64):

$$Q_r = m_w c_{pf} (T_o - T_i) \quad (64)$$

$$\Delta T = T_o - T_i = 150 - 100 = 50\ ^{o}\text{C}$$

$$m_w = \frac{Q_r}{c_{pf} \Delta T}$$

The specific heat of water at 125 °C of average temperature is c_{pf} = 4253.275 J (kg°C)$^{-1}$ and density m_w = 938.865 kgm^{-3} (Table 2).

$$m_{w1500-1900} = \frac{40841.362}{4253.275 \times 50} = 0.192047 \text{ kg s}^{-1}$$

$$m_{w1500-1900} = \frac{0.192047}{938.865} = 0.000205 \text{ m}^3 \text{ s}^{-1}$$

$$m_{w1500-1900} = 0.205 - 0.243 \text{ L s}^{-1}$$

The water flow needed in the heating system is the parameter that must be known in sizing the pipe system that forms the heating system (Eqs. 62-64). Cooling system between engine speed 1500-1900 min^{-1} can produce 2.035-2.320 Ls^{-1} (Eq. 63) hot water. Since there is no direct data for exhaust gas in the thermodynamic tables, the exhaust gas is modeled as a real mixture of certain gas mixtures as a result of combustion. Enthalpy of this mixture was calculated based on enthalpy additives proportional to the mole ratio of each combustion product.

Combustion products are determined by the actual combustion equation written for exergy analysis. $C_{17}H_{34}$ formula usually represents used as diesel engine fuel (Megep, 2011a). Exhaust gas ideal heat transfer capacity was made under ideal conditions. Exhaust gas heat transfer was calculated by obtaining intermediate values (Bergman and Lavine, 2017) for extreme air factor λ = 1.7 waste gas temperature T_r = 786 K (Eq. 65).

Combustion reaction for unit fuel:

$$C_{17}H_{34} + 43.35(O_2 + 3.76N_2) \rightarrow$$
$$17CO_2 + 17H_2O + 163N_2 + 17.85O_2$$

The thermophysical properties of exhaust gas (Table 1) are determined according to the Eq. (64) for each parameter.

Energy balance for the engine is given by Eq. (65):

$$\sum \frac{M_i N_i X_i}{M_i N_i} \quad (65)$$

The average temperature specific heat of water at 125°C is given in Table 2.

In order to determine the heat exchanger dimensions, the flow rate of the exhaust gases must be known.

Flow rate of the exhaust gases (1500-1900 min^{-1}) (Eq. 66):

$$m_{fuel1500-1900} = \frac{G_{f1500-1900}}{3600} \tag{66}$$

$$m_{fuel1500-1900} = \frac{12.139}{3600}$$

$$m_{fuel1500-1900} = 0.00337 - 0.00401 \text{ kg s}^{-1}$$

$$l_0 = 14.452 \text{ kg air (kg fuel)}^{-1}$$

Along with the air excess coefficient (Eq. 67):

$$l_{0_\lambda} = 14.452 \times 1.7 = 24.568 \text{ kg air (kg fuel)}^{-1}$$

$$l_{0_\lambda} = \frac{m_{air}}{m_{fuel}} \tag{67}$$

m_{air}= 0.09852 kgs^{-1}

$$m_{rmass1500-1900} = m_{fuel} + m_{air} \tag{68}$$

$$m_{rmass1500-1900} = 0.10189 - 0.10253 \text{ kg s}^{-1}$$

Exhaust gas outlet temperature (1500-1900 min^{-1}) (Eq. 69):

$$Q_{r1500-1900} = m_{rmass} c_{pr} (T_{egi} - T_{ego}) \tag{69}$$

$$Q_{r1500-1900} = 0.10189 \times 1177 \times (513 - T_{ego})$$

$$Q_{r1500-1900} = 172 - 110\ ^{o}\text{C}$$

The inlet and outlet temperatures of the exhaust gas and water to the heat exchanger were calculated as the logarithmic average temperature difference (Fig. 3; Eqs. 66-70).

$$\Delta T_{lm1500-1900} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{70}$$

$$\Delta T_{lm1500-1900} = \frac{363 - 10}{\ln\left(\frac{363}{10}\right)}$$

$$\Delta T_{lm1500-1900} = 180 - 98\ ^{o}\text{C}$$

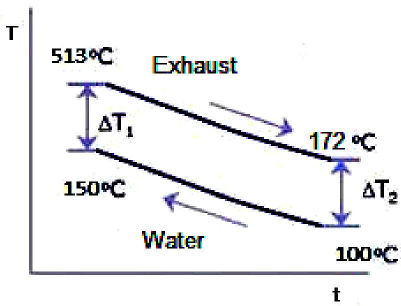
Heat exchanger diameter is D 150 mm. In order to obtain the effect of the pipe diameter in finding the length of the spiral copper pipe placed in the heat exchanger and used to increase the temperature of the water, the Reynolds (Eqs.71-72) and Nusselt number (Eq. 73) with the pipe friction coefficient (Eq. 74) should be determined.

Table 1. Thermophysical properties of exhaust gases

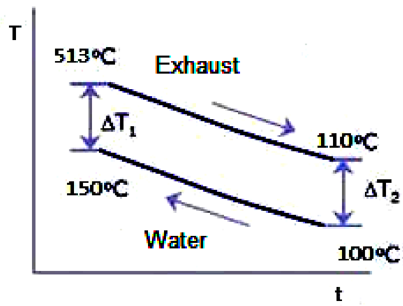
Conditions	***CO₂***	***H₂O***	***N₂***	***O₂***	***Exhaust***
M kg	44.0098	18.1520	28.0134	15.9994	
N kmol	17	17	163	17.85	
Specific heat c_{pr} J(kg K)$^{-1}$	x_i 1159	2144.3	1122	1049.6	1177
Density ρ_r kg m^{-3}	0.6997	0.2865	0.4454	0.5088	0.4724
Thermal conductivity k_{Tc} W(mK)$^{-1}$ conductivityconductivity $k\ [W\ (m\ K)^{-1}]$	0.053634	0.068298	0.054148	0.062151	0.055209
Dynamic viscosity μ Nm^{-2}	34486x10^{-9}	2889x10^{-8}	3476x10^{-8}	4150x10^{-8}	3475x10^{-8}
Kinematic viscosity υ m^{2}s^{-1} [m^{2}s^{111}]^1]$\upsilon\ (m^2\ s^{-1})\upsilon\ (m^2\ s^{-1})$**viscosity** $\upsilon\ (m^2\ s^{-1})$	49246x10^{-9}	10407x10^{-8}	8021x10^{-8}	8388x10^{-8}	7771x10^{-8}
Prandtl number P_r	0.7460	0.9118	0.7219	0.7011	0.7339

Table 2. Thermophysical properties of water at 125 ºC

c_{pf} [J(kg ºC)$^{-1}$]	m_w [kgm^{-3}]	k [W(mK)$^{-1}$]	μ [Nsm^{-2}]	υ [m^{2}s^{-1}]	P_r
4253.275	938.865	0.692	2.2315 x 10^{-4}	0.2376 x 10^{-6}	0.9936



(a)



(b)

Fig. 3. Temperature conversions in the heat exchanger for (a) 1500 min^{-1} and (b) 1900 min^{-1}

Reynolds number of the fluid gas in the pipe (1500-1900 min^{-1}; Eqs. 71-72):

$$\mathrm{Re}_{HE} = \frac{\rho v D}{\mu} \tag{71}$$

$$m_{rmass} = \rho A v \tag{72}$$

$$\mathrm{Re}_{HE1500-1900} = \frac{4m_{rmass}}{\pi D\mu}$$

$$\mathrm{Re}_{HE1500-1900} = \frac{4\times 0.10189}{3.14\times 0.15\times 0.00003475}$$

$$\mathrm{Re}_{HE1500-1900} = 24901 - 25057$$

Since Re >10000, the flow is turbulent.

Since black pipe will be used in heat exchanger production, the surfaces are rough.

Nusselt number according to the Gnielinski formula (1500-1900 min^{-1}) is given by Eq. (73):

$$Nu_{HE1500-1900} = \frac{\left(\frac{f}{8}\right)(\mathrm{Re}_{HE}-1000)\,\mathrm{Pr}}{1+12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(\mathrm{Pr}^{\frac{2}{3}}-1\right)} \tag{73}$$

The friction factor is derived from the formula (1500-1900 min^{-1}) (Eq. 74):

$$f = (0.79\ln \mathrm{Re}_{HE} - 1.64)^{-2} \tag{74}$$

$$f_{1500-1900} = 0.79\times \ln(24901) - 1.64)^{-2}$$

$$f_{1500-1900} = 0.0248 - 0.0247$$

$$Nu_{HE1500-1900} = \frac{\left(\frac{0.0248}{8}\right)\times(24901-1000)\times 0.7339}{1+12.7\times\left(\frac{0.0248}{8}\right)^{\frac{1}{2}}\left(0.7339^{\frac{2}{3}}-1\right)}$$

$$Nu_{HE500-1900} = 81 - 63$$

In order to find the length of the helical (spiral) copper pipe laid in the heat exchanger to increase the temperature of the water, the friction coefficient of the copper pipe must be determined. On the other hand, it is necessary to know the heat transfer coefficient in order to determine the amount of energy in the exhaust gas passing through the copper pipe to the water flowing (Eq. 75). For single-phase flow in a spiral pipe, a secondary flow patterns occur that changes fluid behavior and hence the friction factor in the pipe. To determine the pipe friction coefficient Reynolds number (Eq. 76) and the Dean number is needed to determine the friction factor inside the spiral tube (Eq. 77). Accordingly, the copper pipe friction coefficient is determined in order to find the length of the spiral (spiral) copper pipe placed in the heat exchanger and used to increase the temperature of the water (Eqs. 78-79).

Exhaust gas heat transfer coefficient (1500-1900 min^{-1}) (Eq. 75):

$$h_{d_D 1500-1900} = \frac{Nu_{HE1500-1900}\,k_{Tc}}{D} \tag{75}$$

$$h_{d_D 1500-1900} = \frac{81\times 0.055209}{0.15}$$

$$h_{d_D 1500-1900} = 29.813 - 23.188\ \mathrm{W(m^2K)^{-1}}$$

Reynolds number of the spiral tube is calculated based on Eq. (76):

$$\mathrm{Re}_{D_{sp}} = 2100\left[1+12\sqrt{\frac{D_{D_{sp}}}{2R_{D_{sp}}}}\right] \tag{76}$$

$$\mathrm{Re}_{D_{sp}} = 2100\times\left[1+12\sqrt{\frac{0.01}{2\times 0.055}}\right] = 9698$$

Friction loss for curvature ratios (*R/a*) has been reported as Dean Number. Values that characterize a circular bent pipe; pipe radius (*a*) and curvature radius (*R*) (Kakaç, 1987) (Eq. 77).

$$De_{D_{sp}} = \mathrm{Re}_{D_{sp}}\sqrt{\frac{D_{D_{sp}}}{2R_{D_{sp}}}} \tag{77}$$

$$De_{D_{sp}} = 9698\times\sqrt{\frac{0.01}{2\times 0.055}} = 2924$$

Friction factor for flow in a smooth spiral tube is (Eq. 78):

De> for 2000;

$$f_{c_D smooth} = \frac{7.0144}{\mathrm{Re}_{D_{sp}}}\sqrt{De_{D_{sp}}} \tag{78}$$

$$f_{c_D smooth} = \frac{7.0144}{9698}\sqrt{2924} = 0.0391$$

The following correction is proposed to explain the effect of the roughness. For this correction, the friction factor for both rough and smooth straight pipe must be known.

Ratio of smooth pipe and rough pipe friction factors for helix forming cooper pipe (Eq. 79):

$$f_r = \frac{f_{c_D rough}}{f_{c_D smooth}} \tag{79}$$

Relative roughness for heat exchanger (Eq. 80):

$$k_{srltv} = \frac{k_s}{D} = \frac{0.0000015}{0.01} = 0.00015 \tag{80}$$

Equations (78-79) correction is proposed to explain the effect of roughness. For this correction, the friction factor should be known for both rough and smooth straight pipe. Next, the ratio of friction factors for the smooth pipe and rough pipe in the diameter of the pipe forming the helix is determined.

Copper pipe roughness coefficient was evaluated as k_s 0.0015x10^{-3} (Eq. 80). To determine the thickness of the thermal boundary layer and the hydrodynamic boundary layer, the Prandtl number 1.41 (Eq. 81) is determined according to the data in Table 3.

$$\mathrm{Pr}_{D_{sp}} = 4.763\times10^{-12}\times125^{6} - 3.305\times10^{-9}\times125^{5} + 9.165\times10^{-7}\times125^{4} - 0.00013111\times125^{3} + 0.0105426\times125^{2} - 0.4958255\times125 + 13.6167 \tag{81}$$

$$\mathrm{Pr}_{D_{sp}} = 1.41$$

The Nusselt number (Eq. 82), heat transmission coefficient (Table 4; Eq. 83), and heat conduction coefficient (Eq. 84) of the copper pipe are determined to find the energy transferred to the water flowing through the copper pipe.

Nusselt number for copper pipe (1500-1900 min^{-1}) (Eq. 82):

$$Nu_{D_{sp}1500-1900} = \frac{\left(\frac{f_{c_D rough}}{8}\right)(\mathrm{Re}_{D_{sp}} - 1000)\,\mathrm{Pr}}{1 + 12.7\left(\frac{f_{c_D rough}}{8}\right)^{\frac{1}{2}}\left(\mathrm{Pr}^{\frac{2}{3}} - 1\right)} \tag{82}$$

$$Nu_{D_{sp}1500-1900} = \frac{\left(\frac{0.0012903}{8}\right)\times(9698-1000)\times1.41}{1 + 12.7\times\left(\frac{0.0012903}{8}\right)^{\frac{1}{2}}\times\left(1.41^{\frac{2}{3}} - 1\right)} = 1.899$$

Copper pipe heat transmission coefficient is

determined according to Table 4 (Eq. 83):

$$k_{ht} = 0.565028 + 0.002636\times125 + (-0.000125\times125^{1.5}) + (-1.515\times10^{-6}\times125^{2}) + (-0.0009412\times125^{0.5}) \tag{83}$$

$$k_{ht} = 0.685640$$

The thermal conductivity coefficient of copper pipe with thickness δ_1=1 mm is λ_1=400 W(mK)$^{-1}$. To find the pipe length, the mixed transmission coefficient must be determined. Transmission coefficient is U (W(m^{2}K)$^{-1}$ Eqs. 85-87. When the Eq. 87 processes are repeated for the structures located in the middle and inner parts of the spiral pipe, which are in three groups within the heat exchanger, the total heat conduction results are obtained in Eq. 88. Total heat transfer coefficient in a pipe heat exchanger with no fins is Eqs. (84-86). The length of the copper pipe used in the construction of the heat exchanger is obtained by Eq. 89. The helix length of the outer, middle and inner parts of the three-layer structure of the heat exchanger produced in the spiral structure is found by Eq. 90. The length of the heat exchanger is obtained using Eq. 91. Adding 50% tolerance to the results achieved, the heat exchanger length becomes approximately 0.75 m.

Heat conduction coefficient to water flowing through spiral copper pipe (1500-1900 min^{-1}) (Eq. 84):

$$h_{i_D 1500-1900} = \frac{Nu_{D_{sp}} k_{ht}}{D_{D_{sp}}} \tag{84}$$

$$h_{i_D 1500-1900} = \frac{1.899\times0.685640}{0.01} = 130.203 \ \mathrm{W(m^2K)^{-1}}$$

Transmission coefficient (Eq. 85):

$$UA = \frac{1}{R} \tag{85}$$

Total heat transfer coefficient in a pipe heat exchanger with no fins (Eqs. 86-87):

$$U_D A = \frac{1}{\left(\frac{1}{h_{d_D} A_i} + \frac{\ln\frac{r_d}{r_i}}{2\pi k L} + \frac{1}{h_{i_D} A_d}\right)} \tag{86}$$

$$A = 2\pi r_i L$$

$$U_D = \frac{1}{\left(\frac{1}{h_{d_D}} + \frac{r_i}{k}\times\ln\frac{r_d}{r_i} + \frac{r_i}{r_d}\times\frac{1}{h_{i_D}}\right)} \tag{87}$$

Table 3. Coefficient of variation of Prandtl number by temperature

$\mathrm{Pr}_{D_{sp}}$ =$aT^6+bT^5+cT^4+dT^3+eT^2+fT+13.6167$					
a	b	c	d	e	f
4.763x10^{-12}	-3.305x10^{-09}	9.165x10^{-07}	-0.00013111	0.0105426	-0.4958255

Table 4. Heat transfer coefficient of transferred to water flowing through copper pipe k_{ht} [W/(m°K)] for 0 < T ≤ 300 (Popiel and Wojtkowiak, 1998)

	a	*b*	*c*	*d*	*e*
$k_{cp}=a+bT+cT^{1.5}+dT^2+eT^{0.5}$	0.565028	0.002636	-0.000125	-1.515x10^{-6}	-0.0009412

Calculations are for the copper pipe helix structure in the outer (1500-1900 min^{-1}):

$$U_{D1500-1900} = \frac{1}{\left(\frac{1}{29.813} + \frac{0.05}{400} \times \ln\frac{0.055}{0.05} + \frac{0.05}{0.055} \times \frac{1}{130.203}\right)}$$

$$U_{D1500-1900} = 24.631 - 19.937 \text{ W(m}^2\text{K)}^{-1}$$

The results are reached.

When the calculations are repeated for the copper pipe helix structure in the middle (1500-1900 min^{-1}):

$$U_{O1500-1900} = 25.773 - 20.661 \text{ W(m}^2\text{K)}^{-1}$$

The results are reached.

When the calculations are repeated for the inner copper pipe helix structure within the spiral copper pipe in three groups in the heat exchanger (1500-1900 min^{-1}):

$$U_{I1500-1900} = 27.521 - 21.787 \text{ W(m}^2\text{K)}^{-1}$$

The results are reached.

The total heat conduction coefficient of the copper pipe helix structure consisting of three nested structures (1500-1900 min^{-1}) (Eq. 88):

$$U_{T1500-1900} = U_{D1500-1900} + U_{O1500-1900} + U_{I1500-1900} \tag{88}$$

$$U_{T1500-1900} = 24.631 + 25.773 + 27.521$$

$$U_{T1500-1900} = 77.925 - 62.385 \text{ W(m}^2\text{K)}^{-1}$$

Length of copper pipe (1500-1900 min^{-1}) (Eq. 89):

$$Q_{r1500-1900} = U_T A \Delta T_{lm} \tag{89}$$

$$L_{1500-1900} = \frac{Q_{r1500-1900}}{U_T(\pi D_{he_{avrg}})\Delta T_{lm}}$$

$$L_{1500-1900} = \frac{40841.362}{77.925 \times (3.14 \times 0.105) \times 180}$$

$$L_{1500-1900} = 10 - 25 \text{ m}$$

The helix length of the spiral pipe on the outside (Eq. 90):

$$L_{sp_D} = \sqrt{(\pi D_{avrg_{D_{sp}}})^2 + t^2} \tag{90}$$

$$L_{sp_D} = \sqrt{(3.14 \times 0.105)^2 + 0.010^2} = 0.3299 \text{ m}$$

The helix length of the spiral pipe in the middle:

$$L_{sp_O} = \sqrt{(\pi D_{avrg_{O_{sp}}})^2 + t^2}$$

$$L_{sp_O} = \sqrt{(3.14 \times 0.065)^2 + 0.010^2} = 0.204 \text{ m}$$

The helix length of the spiral pipe inside:

$$L_{sp_I} = \sqrt{(\pi D_{avrg_{I_{sp}}})^2 + t^2}$$

$$L_{sp_I} = \sqrt{(3.14 \times 0.025)^2 + 0.010^2} = 0.079 \text{ m}$$

Heat exchanger length (1500-1900 min^{-1}) (Eq. 91):

$$L_{exc1500-1900} = \frac{Lt}{L_{sp_D} + L_{sp_o} + L_{sp_I}} \tag{91}$$

$$L_{exc1500-1900} = \frac{10 \times 0.010}{0.3299 + 0.204 + 0.079}$$

$$L_{exc1500-1900} \cong 0.2 - 0.5 \text{ m}$$

When 50% tolerance is added, the heat exchanger length becomes about 0.75 m.

Various studies show that high temperatures are more effective than low temperatures that cause plant damage (Ascard, 1995; Daniell et al., 1969; Levitt, 1980). Weed control effect is higher at the same energy dose level, usually at a higher water temperature. Temperatures above 60 ^oC, which are effective in killing plants, dissolve the waxy coating on weed leaves and disrupt the cellular structures of the plant (Hansson and Mattsson, 2003).

Understanding that cell killing rate is related to time and temperature has led to several different methods for converting temperature data over time into a common unit that will allow comparison of different heating systems. This is important for applications of hyperthermia because temperatures during heating are typically non-uniform and temporarily unstable.

Some authors have determined the heat of activation of cells using Arrhenius analysis. After the break point in the Arrhenius plot, the rate of cell killing doubles for each degree of temperature increase. This means that of each degree increase in temperature, the time required to achieve the same thermal isoeffect is halved (Roizin and Pirro, 1991). Sapareto and Dewey (1984) developed a method for converting one time-temperature combination to another. This method is called "thermal isoeffective dose". Here, the time-temperature values are converted to an equivalent minute at 43^{o}C, which is determined as the index temperature. There was no particular reason for choosing 43 ^{o}C. The equation (Eq. 92) needed to do this conversion.

$$CEM\,43\,^oC = tR^{(43-T)} \tag{92}$$

Some researchers have noted that with each degree increase in temperatures above 43 ^{o}C, the time required to achieve the same biological effect in estimating tissue lesion margin decreases approximately twice (Sapareto and Dewey, 1984). The equivalent time for each second of exposure to temperatures higher than 43 ºC is calculated by the isoeffekt equation (Eq. 93).

$$t_{\exp} = t_{critical} \times 2^{(T_{\exp}-43)} \quad (93)$$

In this application, the exposure time is reduced by an exponential time of 2 for each temperature increase. İncreased temperature causes the heating time to decrease rapidly (Dewey, 1994; Dewhirst et al., 2003). In the study on human muscle damage, the predicted heating time at 41 °C was 1800 minutes or 1.23 days, while the heating time approaching the muscle damage threshold at 42 °C decreased to 230 minutes (Field and Morris, 1983). The iso-effect dose method gives good results for predicting damage types in the range of time-temperature combinations. Twelve different temperatures between 41.8 and 46°C applied to the tail of the baby rat were examined the times required to reach 50% probability of losing 10 or more vertebral bodies. They reported that while 500 minutes at 41.8°C were required to achieve the same results, 85minutes at 43°C and 12 minutes at 46°C were sufficient (Morris et al., 1985). Similar studies have been investigated for gut, skin, ear necrosis, foot loss, testicular weight and a wide variety of tumors. The isoeffect dose gave a consistent value over a wide range of temperature-time combinations. This data parameter has biological validity used to determine thresholds of thermal damage in humans or tissues (Kapp and Cox, 1995; Seegenschmiedt et al., 1988).

The results of the research were evaluated with a simulation. In this simulation, it is designed according to the comparison of the proportional correlation of the energy of the fuel consumed for the advancement of the tractor during pesticide applications with the results obtained from the system producing hot water.

Cauwer et al. (2014), reported that the application carried out at 98ºC in their weed control with hot water is 2 and 6 times more effective than 78ºC and 88ºC. Hansson and Ascard (2002), reported that 3400 MJha^{-2} energies is required to reduce 90% weed control in *Sinapis alba* L. control using 116ºC hot water. They calculated this to be around 126 kg or 150 L of diesel fuel ha^{-1}. They reported that this application gave similar dose-response curves with foliar applied herbicides with systemic activity and contact herbicides. 18.7% of the 192.548 kW fuel energy consumed by a 75 kW tractor is consumed in the power take off (PTO) (Fluck, 1992). According to equation 4, the increase in fuel consumption is proportional to the increase in waste heat per unit time. In both pesticide and hot water applications, PTO power is used for spray. The determining factor between the two is the speed of the tractor and the amount of spraying (time-dose). According to the studies of Cauwer et al. (2014) and Hansson and Ascard (2002), when an interpolation is made, the energy required for weed control decreases 6 times from 3400 MJha^{-1} to 570 MJha^{-1} with an increase of 20 ºC in temperature. According to Fourier's law, the greater the temperature difference between the sprayed leaf surface and the application water temperature, the more lethal the hot water. According to Fourier's law, it is predicted that the energy required for weed control efficiency will decrease much more by raising the temperature of the water up to 150 ºC. The consumed energy and fuel consumption of the recommended 3 Lha^{-1} application dose of glyphosate, which is widely used in weed control, can be used as an example comparable to pesticide applications. The application energy combined with the energy consumed in glyphosate production results in a requirement of 604 to 712 MJha^{-1} depending on the number of applications, combination, herbicide rate (Coleman at al., 2019). The diesel fuel equivalent of this is 16.948 Lha^{-1}-19.976 Lha^{-1}. When the unit price of diesel fuel is approximately 0.72 \$, the cost is 12.2-14.4 \$ha^{-1}. When the purchase price of glyphosate is 1.5 \$L^{-1}, the average cost is 16.7-18.9 \$ha^{-1}. The approximate energy equivalent of this is 566 kW-589 kW. For weed control with hot water, the temperature is increased to 150ºC. It is determined that by increasing the temperature to 150ºC for weed control with hot water, the efficiency will increase 10 times according to the interpolation method made according to Cauwer et al. (2014). When the interpolation is applied to the application water data reported by Cauwer et al. (2014) and Hansson and Mattsson (2002), it is predicted that an application water in the range of 230-1930 Lha^{-1} will be used in a weed control application to be made with hot water. Accordingly, the diesel fuel consumption of Tümosan 4DT-39l-105C engine is 7-51 L (1500-1900 min^{-1}) in order to increase 230-1930 Lha^{-1} application water to 150ºC. When the unit price of diesel fuel is approximately 0.72 \$, it is understood that only 5-37 \$ha^{-1} cost will be spent for weed control and no other costs are required. The approximate energy equivalent of this is 60 kW-503 kW. The energy required for this is met by the diesel fuel consumption that the tractor has spent while moving in the field for weed control. There is also no need to consume fuel. This result is the results obtained according to the proportionality theory, without considering the isoeffective dose (time-dose) relationship. Interpolation approach gives the most negative result possible. However, these results show that the results obtained with superheated water will be realized at an economic level close to herbicide applications. In fact, high temperature will shorten the application time much more due to the exponential effect between time and dose to achieve the same effect. This result is accepted as an important indicator that this system, which produces hot water for agricultural purposes, can become an economically commercial product.

4. Conclusions

The heat energy transferred from the engine to the cooling at an average speed of 1500 min^{-1} of the tractor working in field conditions is 41,252 kW and the amount of heat released from the exhaust is equivalent to 40,841 kW. At 1900 min^{-1} engine speed, the heat energy transferred to the cooling system becomes 53,145 kW and the amount of heat energy thrown out from the exhaust becomes 48,606 kW. The system, designed to obtain hot water from engine cooling water and exhaust waste energy by adding some equipment to the 75 kW tractor engine, has the capacity to produce 558-875 Lh^{-1} superheated water at 150ºC. With the addition of a second heat exchanger to the system, there is a potential to increase the temperature of the hot water at the same capacity.

According to traditional spraying methods, 82094.291 Js^{-1}-101751.664 Js^{-1} of the 192547.92 Js^{-1} fuel energy consumed by the tractor engine is saved and brought into production through the equipment installed in the cooling and exhaust systems. With the help of equipment installed in the cooling and exhaust system of the tractor engine, at least 42% to 53% less fuel is consumed compared to other hot water generating systems. The possibilities and savings of installing a waste energy-using system on a tractor engine can be a good investment.

When compared with the commonly used glyphosate application, the energy consumed is 560-589 $kWha^{-1}$, while it is predicted that it will be 60-503 $kWha^{-1}$ in hot water application according to the interpolation approach, which includes negative results with no time-dose relationship. This is an important indicator that will make this system that produces hot water economically useful. The system has the potential to be used effectively with its low cost and small size. In reality, the energy required will be further reduced due to the time-dose.

There is no difficulty in the installation of the system in terms of equipment to be placed in the cooling system. The parts can be fixed to their appropriate places and the desired hot water can be obtained with the fasteners. The accumulation tank can be used by fixing it to the side of the tractor or to the sprayer tank. Space can be saved by placing the expansion tank on top of the accumulation tank. The most important operational difficulty is that the gases passing through the heat exchanger create accumulation over time and cause a decrease in efficiency or blockage. For this, it is recommended to place a particle filter in the exhaust system before the heat exchanger. The fact that the heat exchanger is detachable is another recommendable approach, as it facilitates its cleaning. In order to make the system work more efficiently, all parts of the system are completely isolated. Thus, it is prevented that the employees are exposed to any danger due to high temperature and the machine parts are damaged.

When the time-dose relationship is not considered, a maximum of 7-51 L diesel fuel is needed per hectare for weed control. The cost of this application is 5-37 $ per hectare.

This study can provide research possibilities of first and second soil cultivation as well as other agricultural processes such as seed bed preparation, spacing, and hoe to regarding the usability.

It can also provide opportunities for the development of machines with different characteristics in the agricultural field and economic solutions that can meet different production needs in other industrial areas. This system, which can create high efficiency with its low cost and size, has the potential to be a commercial product alternative to chemical control.

Nomenclature

$m_{incoming\ engine}$ = Amount of hot water circulating in the engine
$m_{incoming\ tank}$ = Amount of hot water coming from the tank
$T_{incoming\ engine}$ = Temperature of the water circulating in the engine
$T_{incoming\ tank}$ = Temperature of the hot water coming from the tank
$T_{mixture}$ = Temperature of the water obtained from the cooling system
ΔT_{water} = Temperature difference of the water circulating in the cooling system
R_a = Gas constant $J(kgK)^{-1}$
R = 8315 $J(kmole^oC)^{-1}$. The universal gas constant
μ_a = 28.96 kg/kmole. The mass of 1 kmole of air
β = Decreasing velocity load coefficient in the cylinder cross-sectional area in question
ξ_{in} = İntake system resistance coefficient of the narrowest cross-sectional area of the system
ω_{in} = Average charging speed in the narrowest cross-sectional area of the intake system
ε = high compression ratio (17)
t_0 = Temperature equal to 0 ºC
t_z = Temperature of the mixture at the end of combustion
ξ_z = low rate of combustion heat that increases the internal energy of the gas used to perform work
S = stroke
N = Piston rev speed
B = Cylinder diameter
τ = the number of cylinders
n_e = Nominal piston rev speed
$(mc_v), (mc_p)$ = Average molar specific heat capacities at constant volume and constant pressure (for air)
$(mc_v''), (mc_p'')$ = Average molar specific heat for residual gas at constant volume and constant pressure
t_c = Ambient temperature for turbocharged diesel engines
t_0 = Atmosphere temperature
ρ_w = The specific mass of the water (for 80 ºC)
c_{pw} = The specific heat of the water (for 80 ºC)
ρ_l = The specific mass of the water (for 90ºC=0.965kgL^{-1})
c_{pc} = Average specific heat of water (for 90ºC=4201.4 J $(kg^oC)^{-1}$
T_{ko} = The temperature of the water leaving the radiator
T_{ki} = The temperature of the water entering the radiator

m_w = Flow rate of water passing through the heat exchanger
c_{pf} = The specific heat of the water (for 125ºC)
T_o = The temperature of the water output the heat exchanger
T_i = The temperature of the water entering the heat exchanger
M_i = Mass of exhaust gases components in kg
N_i = Mass of exhaust gases components in kmol
X_i = Each of the other thermophysical properties of the exhaust gases components
l_0 = Theoretical air requirement for fuel combustion
m_{air} = Flow rate of air
m_{fuel} = Rate of the fuel
m_{rmass} = Flow rate of the exhaust gases
T_{egi} = The temperature of the exhaust gas entering the heat exchanger
T_{ego} = The temperature of the exhaust gas leaving the heat exchanger
ρ = Fluid density
v = Average fluid velocity in the pipe section area
μ = Fluid viscosity
Nu_{HE} = Nusselt number of the fluid gas in heat exchanger
f = Friction factor for heat exchanger
Re_{HE} = Reynolds number of the fluid gas in the pipe
De_{sp} = Outer spiral Dean number
$D_{D_{sp}}$ = Outer spiral copper pipe inner diameter (10 mm)
$\mathrm{Re}_{D_{sp}}$ = Reynolds number for outer spiral copper pipe
$R_{D_{sp}}$ = Helical radius of outer spiral pipe (from axis to pipe centerline 55 mm)
f_r = Ratio of smooth pipe and rough pipe friction factors for helix forming cooper pipe
$f_{c_D smooth}$ = Friction factor for smooth spiral copper tube
$f_{c_D rough}$ = Friction factor for rough spiral copper pipe
k_{srltv} = Relative roughness coefficient
k_s = Copper pipe roughness coefficient
R = resistance components
U_D = Heat transfer coefficient for the outer pipe helix structure of the heat exchanger
t = Step, Time interval
R = Time required to compensate for a change in temperature (0.5 for 43 ºC <)
T = Average temperature during the time interval t

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