Equipment Sizing and Method for the Application of Exhaust Gas Waste Heat to Food Crops Drying Using a Hot Air Tray Dryer

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Abstract

Objectives: Equipment sizing and method of utilizing exhaust gas waste heat for food crops drying are presented. **Methods/findings:** The sizing of the components of the equipment was achieved using known design principles. The system uses an axial flow turbine and a heat exchanger to harness the energy of the exhaust gas of a diesel engine generator. A 250 kVA generator with measured exhaust gas flow rate and temperature of 44.5 m^3/min and 382.7 °C was selected for this work. The purpose of the recovered energy is for the drying of food crops. An arduino platform was used to control the operation of the components of the dryer. The drying chamber consists of three trays whose calculated total volume per batch is 0.0463 m^3. Weight losses across the trays, drying air temperature and humidity in the chamber are monitored by sensors. The test rig of the sized and fabricated equipment is currently undergoing extensive experimentation. Preliminary investigation of the dryer showed that the drying air in the chamber initially at the temperature and humidity of 31°C and 71.2% was heated to 88°C and 22.3% when the dryer operated without load for 45 min at an air speed of 2.0 m/s. **Application:** After cooling and maintaining the drying chamber averagely at 59°C, each tray was loaded with 400 g of the grains of freshly harvested maize. The dryer was allowed to run for 30 min and the results obtained showed weight losses of 127.81 g, 118.36 g, and 116.91 g for the grains in trays 1, 2, and 3, respectively. The application of this energy recovery system to the drying of food products, would help to save a considerable amount of primary fuel which is considered a viable means of cost saving and amelioration of environmental degradation.

Keywords: Waste Heat Recovery, Drying, Global Warming, Environment, Cost Saving, Food Quality.

1. Introduction

There is high demand for fuel as a result of expansion in population, urbanization, and industrialization, which has resulted to increase in fuel costs. For this reason, efforts are being made to make energy use more efficient. Obviously, higher energy conversion or its utilization translates to reduced cost of energy globally. One technique of ensuring high energy efficiency is the use of waste heat recovery systems. Recently, huge efforts are directed to the reduction of the amount of energy that is wasted into the environment. And the ultimate aim is to help in the conservation of exhaustible reserves of fossil fuels, reduction of the carbon footprint and the battling of global climate change as well as the improvement of process economics. According to Ref. [1], waste heat is heat that is produced in a process as a result of fuel combustion or chemical reaction, and then discarded into the environment even though it could still be harnessed for a purpose that is useful and economic. Diminishing petroleum supplies and increasing fuel costs are causing governments and industries to look for ways to increase the power efficiency of engines [2]. An internal combustion (IC) engine heat balance shows that the input energy is approximately split into three equal portions such as the energy put to useful work, energy lost to coolant, and energy lost to the environment with the exhaust gases. Hopefully, with the emerging discoveries on exhaust heat recovery to increase the efficiency of IC engines, world energy demand for the depleting fossil fuel reserves would be reduced and hence the impact of global warming [3–4]. It is widely believed that almost 70% of the energy released from the fuel by an engine is lost, mostly in the form of heat. In Ref. [5], Pradip and Hole reported that approximately 25–30% of the energy generated by engines is dissipated in the form of energy loss through the exhaust gas. In Ref. [6–7], authors stated that the waste heat produced from the thermal combustion process in IC engines which is lost to the environment through an exhaust pipe could get as high as 30–40%. Basically, the direct dumping of exhaust gases into the surroundings, not only wastes energy but also contributes to the damage of the environment. However, several innovative cooling and exhaust heat recovery systems have been introduced to reduce cooling loss and regenerate the power by recovering the waste heat [4]. In recent times, large efforts have been committed towards the recovery of waste thermal energy in vehicles and other waste heat generating industrial machines. Interestingly, it could be surmised from the foregoing that the energy released through the exhaust of IC engines is of the same magnitude as the mechanical power generated by the engine.

Waste heat losses from combustion processes in IC engines are inevitable. Therefore, some facilities can be used to reduce these losses by either improving the equipment efficiency or by the installation of waste heat recovery technologies. The recovered energy could be used in preheating combustion air, space heating, electricity generation, etc. Three essential components are required for waste heat recovery, namely: (i) a source of waste heat that is accessible, (ii) a technique for its recovery, and (iii) an application for the recovered heat. The method required for the heat recovery partly depends on the temperature of the available waste heat and the economics involved. Generally, the higher the temperature of waste heat, the higher the value which makes the heat recovery more cost effective. The temperatures of exhaust gases immediately leaving the engine are as

high as 450–600 °C [2]. Table 1 shows a survey of exhaust gas temperatures from various internal combustion engines of automotive vehicles and stationary engines.

Although technologies of energy recovery are available, there are still large potentials for their application, which according to Ref. [8] have not yet been realized in industries. There are numerous technologies used for waste heat recovery such as heat exchangers, thermoelectric devices, turbo compounds, etc. In this work, the source of waste heat energy to be accessed is the high temperature exhaust gas from a 250 kvA diesel generator. It involves recovery or conversion of energy of the exhaust gas to useful work. Captured and reused waste heat is an emission-free substitute for costly purchased fuels or electricity. By this recovery system, the impact of the pollutants to the environment is significantly reduced. The recovery technology to be employed is an axial flow turbine and a heat exchanger while the energy recovered is to be used for drying agricultural products. Large amounts of agricultural products perish yearly during post-harvest periods because of their relatively high moisture content. Basically, Nigeria is blessed with a landmass of 98.3 million hectares of which 72% is considered suitable for agricultural production [9]. Statistics show that the rate of growth in the area of food production is very low, amounting to 2.5% per annum [10]. According to Ref. [11], the poor growth is attributable to the poor level of food preservation in the country. However, efforts are being made with a view to achieving improvement in food production in a given number of ways such as drying of agricultural products. This encourages farmers to embark on large scale agricultural activities as long as ways of preserving their harvested crops are available. Many agricultural products require long drying times ranging from 5 min to 73 h with optimum drying air temperatures requiring large quantity of energy that results in high overhead drying cost and high prices of dried food products [12–13]. For most food products, it is best to use drying temperatures of 50 °C to 55 °C; and the temperature should never exceed 60 °C [14]. This is because higher temperatures destroy nutrients contained in the food which causes the food to lose its dietary value. When higher temperatures are used, food cooks instead of drying, which results in case hardening where the food dries on outside but moisture trapped inside allowing mold growth [14].

There are many types of dryers operated by different energy sources like solar energy, bioenergy, electrical energy, etc. Most of the energy sources involved in these operations have some associated setbacks in terms of their availability and costs. This makes the end products more expensive. Although sun drying is the cheapest and oldest method of food

TABLE 1. Temperature range for diesel engines

S/N	Engine	Temperature (°C)
1	Single cylinder four stroke diesel engine	456
2	Four cylinder four stroke diesel engine (Tata Indica)	448
3	Six cylinder four stroke diesel engine (Tata Truck)	336
4	Four cylinder four stroke diesel engine (Mahindra Arjun 605 DI)	310
5	Genset (Kirloskar) at power 198hp	383
6	Genset (Cummims) at power 200hp	396

Source: In Ref. [12].

preservation but it exposes the crop to contamination by dust, insect attack and infection by microbes and yet it is not always available. There are different drying methods that are available to ensure continuous high quality food supply. In this work, a hot air tray dryer using exhaust gas waste heat as energy source is considered for the drying of food crops. Tray dryers are widely used in agricultural drying due to their capability to dry products irrespective of time and weather conditions [15]. Among all the drying techniques, the tray dryer is the most extensively used because of its simple and economic design [15]. The food is spread out on trays at an acceptable thickness so that the product can be dried uniformly. Heating may be produced by hot air stream across the trays, conduction from heated trays, or radiation from heated surfaces. In a tray dryer, more products can be loaded as the trays are arranged at different levels. Tray dryers have the capability of drying products at high volume. The key to its successful operation is the uniform airflow distribution over the trays [16].

2. Materials and Methods

2.1. Configuration of the Waste Energy Recovery System

The configuration for the exhaust gas waste energy recovery system from a turbocharged 250 kVA Perkins diesel generator is shown in Figure 1. Here, part of the energy of the exhaust gas is recovered by a turbine and used to drive an electrical generator (dynamo) which in turn is used to charge a battery of 12V7.5Ah. This battery produces the electric power needed to drive the air blower and other devices. The exhaust gas leaving the housing
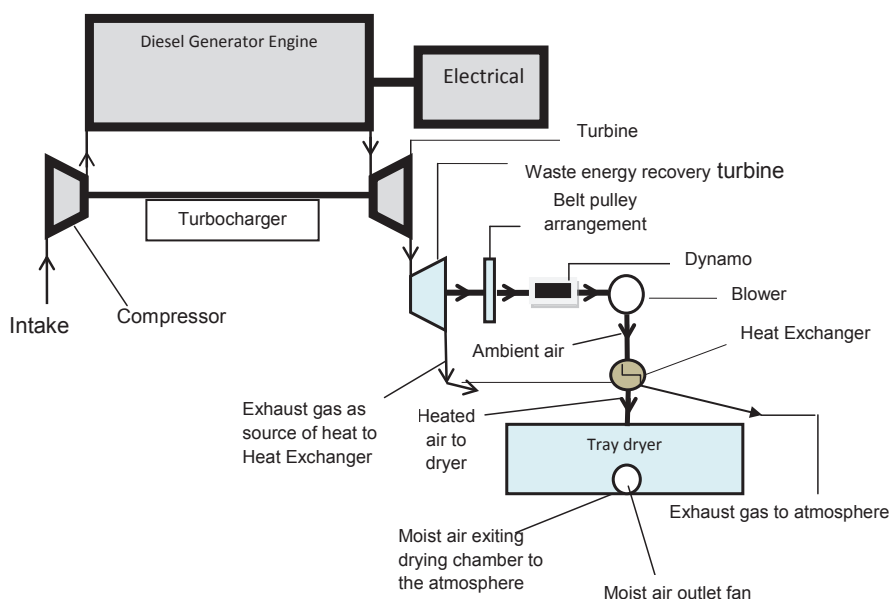

FIGURE 1. The system configuration of the turbine-dryer arrangement for waste energy recovery.

of the energy recovery turbine enters the heat exchanger to raise the temperature of the ambient air (drying air) drawn into the heat exchanger by the air blower. The heated air is then routed to the drying chamber of the tray dryer for the purpose of drying food crops.

The turbine rotor is directly attached to the main shaft connected to the smaller pulley (driving-pulley) as seen in the isometric view of the heat recovery system, Figure 2, while the driven pulley of a larger diameter has a common shaft with the dynamo which generates the electrical energy required in the system. The axial turbine unit, apart from some load bearings, has as its main part as a row of aerodynamically shaped objects (stator) which do not move and a row of aerodynamically shaped objects (rotor) which move and provide the torque to the shaft, and the entire components are placed in a housing as seen in Figure 2. The whole unit is connected to the exhaust gas tail pipe outlet of the generator causing the rotor blades to spin in the exhaust gas streams.

During operation, the exhaust gas travels in the axial direction and enters through the inlet cone, where the fluid slows down to the selected inlet conditions of the blades of the stator. Then the stator blades set the speed and angle at which the exhaust gas must enter the rotor. When the exhaust gas stream strikes the turbine rotor blades by virtue of its kinetic energy, the rotor blades spin providing torque to the shaft which rotates the driving-pulley. The exhaust gas which is still at high temperature exits the turbine housing and enters the heat exchanger where it heats up the ambient air drawn in by the blower (inlet fan). The heated air is then routed through the drying chamber of a tray dryer in
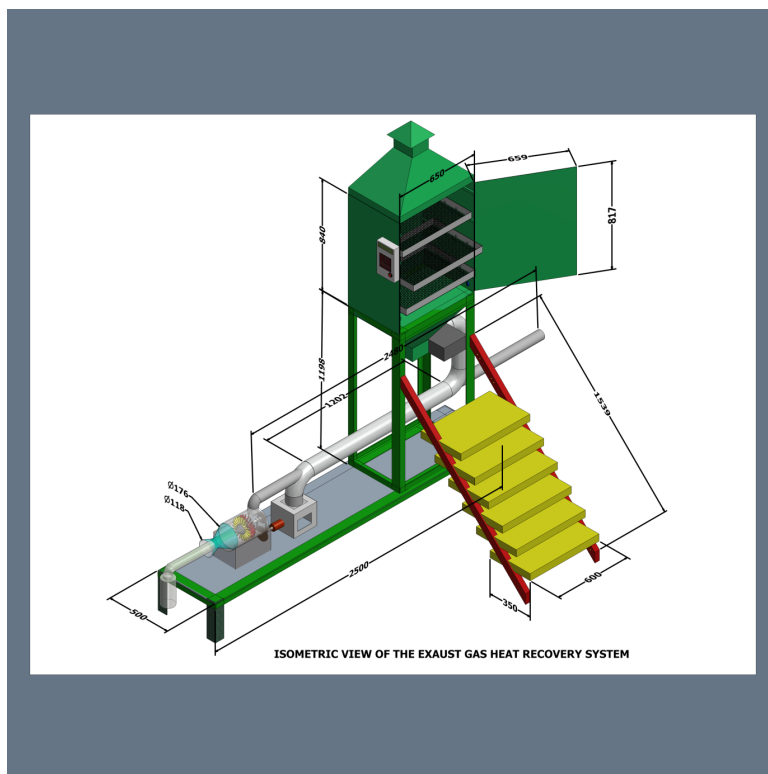

ISOMETRIC VIEW OF THE EXAUST GAS HEAT RECOVERY SYSTEM

FIGURE 2. Isometric view of the exhaust gas heat recovery system.

order to dry the product loaded in the trays. The exploded view of the heat recovery system is shown in Figure 3.

2.2. Selection of the Axial Flow Turbine

The values of the baseline parameters such as the Exhaust gas flow rate, $\dot{V}$ and Inlet temperature of the exhaust gas to the stator, T_{01} are 44.5 m³/min and 655.7 K, respectively. In addition to this, the following preliminary assumptions as seen in Table 2 are used for the selection of the axial flow heat recovery turbine.

Where c_2 is the absolute velocity at inlet of the rotor; V_3 is the relative velocity at exit of the rotor; c_3 is the absolute velocity at exit of the rotor, and V_2 is the relative velocity at inlet of the rotor. Based on the mass flow rate of the fluid and the area upon which it enters the turbine, the initial absolute velocity can be calculated with the flow entering an axial turbine being in an axial direction [17]. The expressions for the annulus areas of the turbine at the various stations, A_i, the mass flow rate, $\dot{m}$ of the exhaust gas, the blade loading coefficient, φ, the blade heights, b_i and the degree of reaction, R_d are given below:

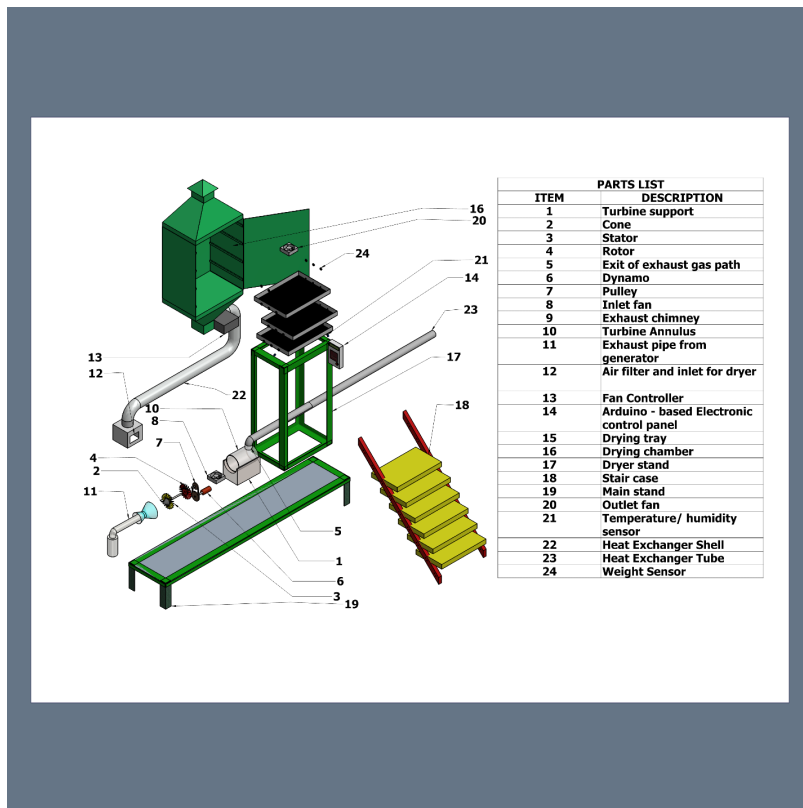$$A_i = \frac{\dot{m}}{\rho_i c_{ai}} \tag{1}$$

PARTS LIST	
ITEM	DESCRIPTION
1	Turbine support
2	Cone
3	Stator
4	Rotor
5	Exit of exhaust gas path
6	Dynamo
7	Pulley
8	Inlet fan
9	Exhaust chimney
10	Turbine Annulus
11	Exhaust pipe from generator
12	Air filter and inlet for dryer
13	Fan Controller
14	Arduino - based Electronic control panel
15	Drying tray
16	Drying chamber
17	Dryer stand
18	Stair case
19	Main stand
20	Outlet fan
21	Temperature/ humidity sensor
22	Heat Exchanger Shell
23	Heat Exchanger Tube
24	Weight Sensor

FIGURE 3. Exploded view of the exhaust gas heat recovery system.

TABLE 2. Assumptions for preliminary selection

Parameter	Value
1. Temperature drop, ΔT_0 (K)	3
2. Mean blade speed, U_m (m/s)	43
3. Nozzle loss coefficient, i_N	0.052
4. Flow coefficient, $\varnothing$	0.80
5. Inlet pressure (kPa)	1.8
6. Pressure ratio, $\dfrac{P_{01}}{P_{03}}$	1.52
7. Rotational speed of shaft, N_b (rpm)	5,100
8. $c_2 = V_3$; $c_3 = V_2$	

$$\dot{m} = V^{\cdot} \times \rho_2 \tag{2}$$

$$\varphi = \frac{2c_p \Delta T_{0s}}{U_m^2} \tag{3}$$

$$b_i = \frac{A_i N_b}{U_m} \tag{4}$$

$$R_d = \frac{h_2 - h_3}{h_1 - h_3} = \frac{T_2 - T_3}{T_1 - T_3} \tag{5}$$

Where the subscript i represents the different stations of the turbine stage, 1, 2, and 3. However, h_1, h_2, and h_3 are the enthalpies of the exhaust gas at inlet to the stator, at entry to the rotor, and at exit from the rotor, respectively. Similarly, T_1, T_2, and T_3 are the static temperatures at the stated positions. ρ_2 is the density of the exhaust gas at inlet of the rotor. The gas angles of the velocity triangles are calculated at the mean radius using the following equations:

$$\tan \beta_3 = \frac{1}{2\varnothing}\left(\frac{\varphi}{2} + 2R_d\right) \tag{6}$$

$$\tan \beta_2 = \frac{1}{2\varnothing}\left(\frac{\varphi}{2} - 2R_d\right) \tag{7}$$

$$\tan \alpha_3 = \tan \beta_3 - \frac{1}{\varnothing} \tag{8}$$

$$\tan \alpha_2 = \tan \beta_2 + \frac{1}{\varnothing} \tag{9}$$

where α is the angle made by the absolute velocity with the axial direction, β is the angle the relative velocity makes with the axial direction. Figure 4 shows the superimposed diagram of the velocity triangles. Δc_w, as seen in Figure 4 is the change in the velocity of whirl while c_{w2} and c_{w3} are whirl velocities.

2.3. Flow Parameters at the Outlet of the Stator

Considering the geometry of the superimposed velocity diagram above, the following equations can be derived:

$$c_{ai} = U_m \varnothing \tag{10}$$

$$c_i = \frac{c_{ai}}{\cos \alpha_i} \tag{11}$$

The temperature equivalent of the exit velocity is:

$$T_{02} - T_2 = \frac{c_2^2}{2c_p} \tag{12}$$

For a single stage turbine, it is assumed that $\alpha_1 = 0$ since c_1 is axial; and this together with the assumptions that $c_1 = c_3$ and $c_{a3} = c_{a2}$, yields,

$$c_{a1} = c_1 = c_3 \tag{13}$$

For choke conditions at the outlet of the stator, we have that,

$$T_2 - T_2^I = \mu_N \left(\frac{c_2^2}{2c_p} \right) \tag{14}$$

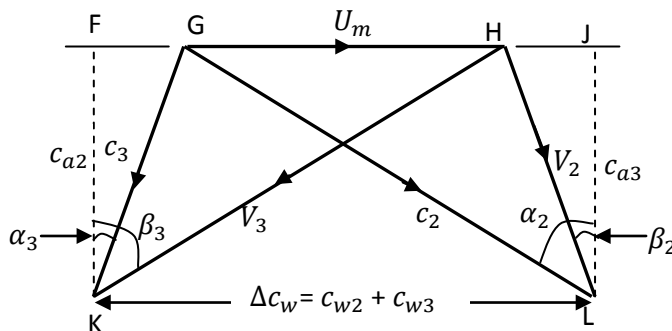P_2 is calculated from the reversible adiabatic relation:

FIGURE 4. Schematic of the superimposed velocity diagram of the single stage reaction turbine.

$$\frac{P_{01}}{P_2} = \left(\frac{T_{01}}{T_2^I}\right)^{\frac{\gamma}{\gamma-1}} \tag{15}$$

Neglecting the frictional effect on the critical pressure ratio, we have that:

$$\frac{P_{01}}{P_c} = \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{\gamma-1}} \tag{16}$$

Calculation of the densities at various stations of the turbine stage is obtained using equation (17):

$$\rho_i = \frac{P_i}{RT_i} \tag{17}$$

The temperatures equivalent of the entry and exit kinetic energies are:

$$T_{01} - T_1 = \frac{c_1^2}{2c_p} \tag{18}$$

and

$$T_3 = T_{03} - \frac{c_3^2}{2c_p} \tag{19}$$

The reversible adiabatic relation at station 3 is given by:

$$P_3 = P_{03}\left(\frac{T_3}{T_{03}}\right)^{\frac{\gamma}{\gamma-1}} \tag{20}$$

The radius ratio of the annulus can be found from equation (20),

$$\frac{r_t}{r_m} = \frac{r_m + \frac{b}{2}}{r_m - \frac{b}{2}} \tag{21}$$

and

$$r_m = \frac{r_t + r_r}{2} \tag{22}$$

where r_r is the root radius of the turbine, r_m is the mean radius, and r_t is the tip radius, respectively. From [18], the power output of the turbine, P_{out} is given by:

$$P_{out} = \dot{m}U_m\Delta C_w \tag{23}$$

where ΔC_w is the change in the velocity of whirl as seen in Figure 4.

The calculated dimensions of the annulus as well as the values of the parameters at various stations of the turbine are presented in Table 3:

TABLE 3. Annulus dimensions and parameters of the turbine at various stations

Station	A(m²)	b(m)	r_t / r_m	$\overset{\cdot}{p}\left(kg/m^3\right)$	$\overset{\cdot}{c_a}\left(m/s\right)$	$\overset{\cdot}{c}\left(m/s\right)$	$\overset{\cdot}{P}$ (bar)	T (K)	$\alpha(°)$	$\beta(°)$
1	0.0188	0.03724	1.602	0.9540	39.081	39.081	1.793	655.04	0	0
2	0.0216	0.0427	1.722	0.9464	34.400	70.510	1.775	653.53	28.33	60.8
3	0.0324	0.0641	2.323	0.6302	34.400	39.081	1.179	652.04	60.8	28.33

2.4. Description and Sizing of the Hot Air Tray Dryer and the Heat Exchanger

The tray dryer consists of seven major components, namely: the drying chamber, the trays, the chimney, the heat source, the temperature and humidity sensors, the strain gauge weight sensors, and the moist air outlet fan.

The dryer operates on the principle of batch drying. The dryer consists of a drying chamber where the products to be dried are fed before drying commences. Heat is supplied to the chamber by the air which is heated up by the exhaust gas passing through the heat exchanger. The heat energy supplied to the dryer is part of the heat recovered from the exhaust gas energy of the diesel generator. The blower powered by the electrical energy from the dynamo is connected to the heat exchanger into which it draws in ambient air to be heated up by the exhaust gas passing through the heat exchanger. The air heated by the heat exchanger expands and travels upwards into the drying chamber. The food crops to be dried are spread on wire gauze trays inside the drying chamber. As the warm air moves and circulates across the surface of the food crops, it picks up moisture. The air that has picked up moisture on its way through the drying chamber leaves the chamber through the chimney provided at the top of the dryer. The effectiveness of the evacuation of moist air through the chimney is enhanced by the use of a moist air outlet fan placed near the top of the dryer. The cooled exhaust gas passing through the heat exchanger is then released into the atmosphere through a pipe whose outlet is at a reasonable height and distance away from the dryer to avoid the contamination of the surrounding air to be used for drying. However, black carbon (a fine particle of $PM_{2.5}$ emitted from diesel engines) and other larger airborne particles which settle faster due to gravity [19] may pose a danger of contamination to the drying air. Therefore, the drying air to be heated is drawn into the heat exchanger by the aid of a blower whose box is covered with a mesh air filter of 2 μm size in order to prevent these particles from being sucked into the drying chamber. To preserve the quality of the food crops during drying, the temperature of the drying chamber is controlled so that it does not exceed 60 °C. The temperature of the drying chamber is monitored using sensors. However, above this temperature, the temperature sensors give a signal to the control system which cuts off the flow of heated air into the chamber by switching off the air

blower until the chamber temperature falls to 50 °C (which as stated above is within the range of acceptable drying temperatures) before the operation of the blower is activated once again. The whole control processes in terms of temperature and velocity regulation of the drying air within the drying chamber are made possible by the use of an arduino platform. The 3-D view of the dryer is shown in Figure 2, while the schematic diagram of the dryer is shown in Figure 5. The dryer has been fabricated using the following materials: (i) Equal length angle steel for constructing the dryer frame; (ii) Mild steel sheet for covering the external part of the drying chamber; and (iii) Fiber glass used as a lagging material for the thermal insulation of the dryer. The trays of the dryer were constructed using galvanized steel wire gauze with aluminum profile base. The aim of using galvanized steel wire gauze is to avoid the problem of rust of the tray material and contamination of products to be dried.

2.5. The Dimensions of the Trays and the Drying Chamber

The drying chamber is composed of three (3) trays with the total volume, of products to be dried per batch equal to 0.04628 m³. The length, l_t, width, w_t and height, H_t of the trays are 0.66 m, 0.468 m, and 0.05 m, respectively. The head and bottom spaces of the dryer from the trays, H_{hs} and H_{bs} are 0.12 m and 0.17 m while the tray thickness, T_t and the space between the trays, S_t are 0.005 m and 0.09 m, respectively. The drying chamber height, H_C, 0.71 m is obtained using the expression:

$$H_c = 3 \times H_t + 3 \times T_t + 2 \times S_t + H_{hs} + H_{bs} \qquad (24)$$

2.6. Water Removal from Products to be Dried and Heat Requirement for Drying

Weight loss from wet to dried product is calculated from Ref. [20], where the mass of dry product is given by:

$$M_d = \frac{m_w \left(100 - \varnothing_o \right)}{100 - \varnothing_f} \qquad (25)$$

where m_w is the total mass of wet product per batch, $\varnothing_o$ is the percentage maximum moisture content of freshly harvested product, and $\varnothing_f$ is final moisture content after drying. Thus, the mass of water to be removed, M_w is

$$M_w = m_w - M_d \qquad (26)$$

According to Ref. [20], the quantity of heat required to remove water from the product is given by:
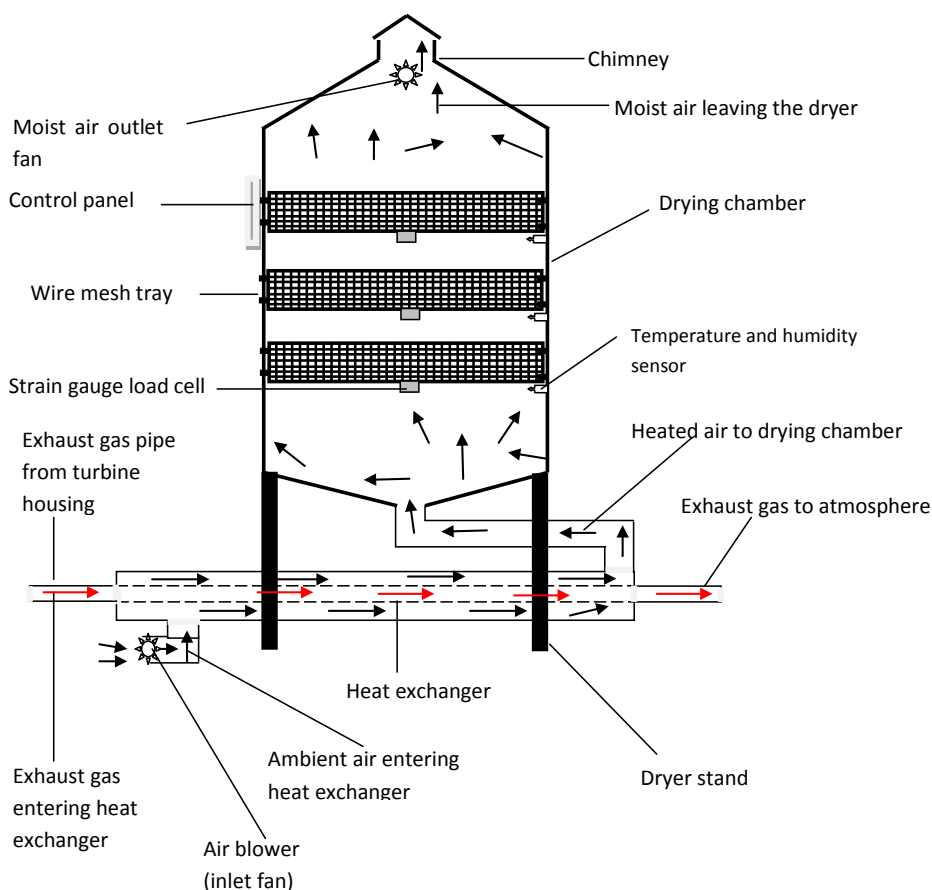
FIGURE 5. Schematic diagram of the hot air tray dryer.

$$Q = m_w \times c_c \left(T_{dp} - T_{ic} \right) + M_w L \qquad (27)$$

where L is the latent heat of vaporization of water (2256 kJ/kg), T_{dp} is the temperature of the drying product, and T_{ic} is the initial temperature of the drying chamber. The rate of heat transfer, $\dot{Q}$ between the exhaust gas and the ambient air in the heat exchanger is given by Ref. [21] to be:

$$\dot{Q} = \frac{Q}{\text{time}\left(s \right)} \qquad (28)$$

The temperature of the exhaust gas at the exit of the heat exchanger, t_{g2} can be obtained from the heat transfer rate $\left(\dot{Q} \right)$ expression:

$$\dot{Q} = \dot{m}_g c_{pg} \left(t_{g1} - t_{g2} \right) \qquad (29)$$

where c_{pg} is specific heat of exhaust gas at constant pressure, $\dot{m}_g$ is mass flow rate of exhaust gas, and t_{g1} is the temperature of the exhaust gas at the inlet of the heat exchanger.

The logarithmic mean temperature difference (LMTD), θ_m is used to determine the area
and length of the heat exchanger to be used, and is given by Ref. [22], as:

$$\theta_m = \frac{\theta_1 - \theta_2}{\text{In}\left(\dfrac{\theta_1}{\theta_2}\right)} = \frac{\left(t_{g1} - t_{a1}\right) - \left(t_{g2} - t_{a2}\right)}{\text{In}\left[\left(t_{g1} - t_{a1}\right) / \left(t_{g2} - t_{a2}\right)\right]} \tag{30}$$

While the overall heat transfer coefficient, U according to [22] is given by:

$$\frac{1}{U} = \frac{r_o}{r_i} \times \frac{1}{h_i} + \frac{1}{h_o} \tag{31}$$

where h_i is inside heat transfer coefficient, h_o outside heat transfer coefficient, r_i and r_o
are inside and outside radii of the tube. For a parallel flow heat exchanger, the length, L_{he}
and area, A_{he} can be estimated using the expressions by [22]:

$$A_{he} = \pi d_o L_{he} \tag{32}$$

and

$$L_{he} = \frac{\dot{Q}}{U \times \left(\pi d_o \theta_m\right)} \tag{33}$$

2.7. Thermal Insulation of the Drying Chamber

The drying chamber is thermally insulated to avoid excessive heat losses through its walls.
The thickness of the heat insulating (lagging) material was determined using an analogy
of heat transfer through a composite wall. The chamber wall is made of mild steel plates
at the internal and external parts while the heat lagging material (fiber glass) is placed
in-between the mild steel plates of the chamber walls as shown in Figure 6.

The thermal conductivities of the three layers of the wall are as follows: k_1, k_2, and k_3,
while their thicknesses are as follows: x_1, x_2, and x_3, respectively. The temperatures of
the inner and outer surfaces of the drying chamber walls are t_1 and t_4 while the interface
temperatures are t_2 and t_3, respectively. However, for continuity of flow to be achieved,
the rate of heat transfer $\dot{q}$ through the layers of the wall must be the same. In addition, the
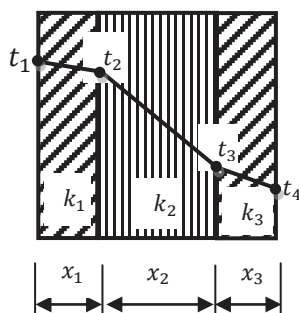

FIGURE 6. Lagging of the drying chamber walls.

internal and external parts of the chamber are of the same material and thickness. Hence, $k_1 = k_3$ and $x_1 = x_3$. Thus, by Fourier's equation of heat transfer, we have:

$$\dot{q} = \frac{k_1 A(t_1 - t_2)}{x_1} = \frac{k_2 A(t_2 - t_3)}{x_2} = \frac{k_1 A(t_3 - t_4)}{x_1} \tag{34}$$

2.8. The Fan/Blower Selection

Blowers are sized based on the volume of air delivery and static pressure. In this work, the blower is powered by the electricity generated by the dynamo which is driven by the waste energy recovery turbine. The work done by the heat source per unit time, P_{ht} is equivalent to the work done on the ambient air, P_a [21]. Thus:

$$P_{ht} = P_a = \dot{m}_a c \Delta T \ . \tag{35}$$

where $\dot{m}_a$ is mass flow rate of air, c is specific heat capacity of air, and ΔT is the difference in the values of the maximum allowable temperature of the product to be dried and the ambient temperature. From Ref. [21], discharge of the heated air, D is given by:

$$D = \dot{m}_a v_a \tag{36}$$

where v_a is the specific volume of air. The velocity of air, c_a required through the drying chamber [21], is calculated thus:

$$c_a = \frac{D}{A_2} \tag{37}$$

where A_2 is the area of the drying chamber's floor. From Ref. [21], the static pressure loss due to sudden enlargement is given by:

$$L_{se} = \frac{U_1^2}{2g\left(1 - \dfrac{A_1}{A_2}\right)} \tag{38}$$

where U_1 is the linear velocity of air in the heat exchanger; g is the acceleration due to gravity; and A_1 is the area of the heat exchanger. The horsepower of the blower, B_{hp} [21] is obtained using the expression below:

$$B_{hp} = \frac{D \times p_{st}}{6320 \times \xi_b} \tag{39}$$

where ($6320 \times \xi_b$) is called the conversion factor and ξ_b is the blower efficiency, D is the airflow rate (in CFM), and p_{st} is the static pressure in inches of water.

3. Discussion

Heat recovery from the high heat content exhaust gases exiting the tail pipe of diesel powered electric generators has barely received any meaningful attention by researchers. Yet, this enormous heat is directly dumped into the environment even though it could still be harnessed and transferred to a productive end-use. The reason behind the unavailability of research works done in this area could perhaps be due to the inability of researchers to identify any meaningful and beneficial application of the energy recoverable from these systems using available techniques. However, it is pertinent to note that heat recovery from stationary diesel generators can be used effectively to power dryers for the preservation of agricultural products. This is achievable by an appropriate arrangement of the components of the energy recovery turbine, the heat exchanger and the tray dryer as demonstrated in this work. In a conventional tray dryer, the blower required for its operation is usually attached to the dryer where it blows air across the heating source and supplies the resulting hot air to the trays holding the materials to be dried. However, the blower used in this work has no direct connection with the dryer. It is rather connected to the heat exchanger into which it drives air to be heated by the exhaust gas passing through the heat exchanger, and the heated air is then routed to the drying chamber of the dryer through a duct for drying of products. In addition, unlike in most heat recovery systems, it is interesting that in this work, there are two stages of energy recovery. The first stage is the energy recovery turbine while the second stage involves the use of a heat exchanger to further recover heat from the exhaust gas exiting the turbine housing. The test rig of this whole equipment has been fabricated and is currently undergoing experimental investigation.

   Preliminary test of the dryer showed high drying rate of maize grains. During the test, the tray dryer was initially allowed to run without load for a period of 45 min; and the air temperature and humidity of 88°C and 22.3% were recorded in the drying chamber for an air speed of 2.0 m/s. The initial temperature and humidity of the chamber were 31°C and 71.2%, respectively. Subsequently, the three trays of the dryer were loaded with grains of freshly harvested maize after the chamber had been cooled and averagely maintained at 59°C. The trays contained 400 g each of the grains and the dryer was allowed to run for a period of 30 min. The results obtained showed weight losses of 127.81 g, 118.36 g, and 116.91 g for the grains in trays 1, 2, and 3, respectively. Detailed results of experiments on the drying of freshly harvested maize grains and other crops such as yam and cassava chunks, etc. will be made available (with results analyses) in due course. Performance evaluation of the tray dryer will be carried out; and the specific energy consumption, effective moisture diffusivity, and activation energy for the thin-layer drying of the crops will be determined. The proximate analyses of the dried products will also be obtained in order to determine their chemical and nutritional values.

4. Conclusion

The sizing of the equipment and method of utilizing exhaust gas waste heat for food crops drying have been considered. The sizing of the components of the equipment was achieved

using known design principles. The concept is geared towards recovery and utilization of waste heat of a diesel engine thereby helping to reduce the quantity of pollutants released into the environment. The recovered energy may be put to useful work, such as drying of products, instead of resorting to burning of fresh fuels to do the same work. It is intended that the application of the recovered waste energy (from the exhaust gas of Internal Combustion engines) to drying, would reduce the energy demand for the depleting fossil fuel reserves which saves a considerable amount of primary fuel; help farmers to minimize the amount of agricultural product losses usually incurred during postharvest, thus improving the value and profit margin of the farmers, and hence the impact of global warming.

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