Utilization of Waste Heat Recovered from a Petrol-Powered Generator for Textile Drying

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Abstract
The efficiency of a typical petrol-powered generator ranges between 18% to 20%, the remaining heat energy produced in the cylinders of the generators goes through the exhaust to the atmosphere. The increasing use of petrol-powered generators due to the nations epileptic public power supply and the lack of proper waste heat management leads to increased global warming and new environmental challenges. This concerns have led to increased efforts at improving efficiencies of petrol generators. This paper therefore presents an investigation at improving the efficiency of petrol-powered generators by utilizing its waste heat for clothes drying. The generator exhaust channel was modified to incorporate a waste heat recovery (WHR) drying chamber, energy balance and heat transfer equations was used in analyzing the effectiveness of the drying chamber, actual tests were carried out to compare the effectiveness of drying with the WHR dryer and conventional air drying. It was observed that a cloth with a 0.3kg moisture content dried completely within 6 hours using the experimental device amounting to an average of 0.001972kg of moisture removed from the cloth per minute (0.118kg/hr.) while the conventional method gave an average moisture removal rate of 0.002367kg of moisture per minute (0.142kg/hr.) for a similar cloth. Results from experiment showed that drying rate increased with an increase in temperature. Finally, based on obtained results, about one-third (1/3) of input energy of most petrol generators are wasted and emitted to the atmosphere, however, this waste heat can be viably and effectively harvested for clothes drying.

Keywords: Waste heat recovery; Exhaust; Drying; Drying rate; Petrol Generator

Nomenclature

C_p	specific heat capacity (kJ/kg K)
ρ	density of air (kg/m^3)
R_e	Reynolds number
μ	dynamic viscosity (N-s/m^2)
k	Thermal conductivity (W/mK)
T_{h_i}	inlet temperature of hot fluid(^{o}C)
T_{C_i}	inlet temperature of cold fluid(^{o}C)
T_{h_o}	outlet temperature of hot fluid(^{o}C)
T_{C_o}	outlet temperature of cold fluid. (^{o}C)
$\dot{m}$	mass flow rate (kg/s)
N	Number of tubes
Q	actual heat-transfer (kJ)
U	overall heat transfer coefficient (W/m^{2}K)
K	drying rate (%)
t	drying time (h)

T_a ambient-air temperature (°C)
T_{out} outlet gas temperature(°C)
T_{in} inlet gas temperature(°C)
V_a Exhaust air speed (m/s)
T_1 Outlet temperature of exhaust fume from the generator (°C)
T_2 Drying chamber temperature (°C)
T_e exhaust-air temperature (°C)
ΔT the difference in temperature of T_{out} and the inlet gas T_{in}. (°C)
ΔT_m log mean temperature difference(°C)

1.0 INTRODUCTION

Sustainable generation and supply of electricity is a key index of comfortable living in households and an indication economic development in a country. Nigeria's inability to generate and distribute electricity to her populace is concerning, she currently generates slightly over 4,500MW of power as against the 180,000MW which she needs for adequate power supply (Akindele and Adejumobi, 2017). This put actual electrification at 2% of the population and currently only about 45% of the nation's population have access to the national electricity grid, of which 60% experience epileptic power supply (Nanaghan, 2009; Shameer and Christopher, 2013). This low access to public electricity and its epileptic nature compels most Nigerian households and businesses to resort to privately owned petrol/diesel generators for much of their power needs thereby increasing their energy cost. It is estimated that there are over fourteen million generators in the country and a staggering $16million (₦8 billion) annually to fuel them (Nanaghan, 2009; Segun, 2021). According to United States Department of Energy (USDOE, 2004), most of the energy generated from conventional and renewable resources gets lost to the environment, these losses are mainly due to onsite losses (equipment inefficiency and waste heat) and offsite losses (cables and transformers). Bradley and Aliakbar (2018) also claim that only about 20% of the total input energy in petrol generators is utilised, thus 80% of the energy left is wasted through parasitic and friction losses. Of the 80%, about 30% is lost in engine coolant and 45% in the exhaust gases. This high percentage of energy in the exhaust gas makes the concept of recovering energy from the exhaust gas for industrial or domestic uses an attractive proposition. In this system, the exhaust heat from the engine is recovered and put into use for maximized energy utilisation and increased efficiency (Reay and Span, 2013).

A recovery heat exchange system that gives out heat from a process output at high temperatures to another section of the process usually with higher efficiency is termed a waste heat recovery unit (WHRU). The commonly used heat-recovery devices are the recuperators and regenerator types (Jouhara et al., 2018). The Recuperators can either be co-flow, crossflow or counter flow heat exchangers, however, as reported by (Reay and Span, 2013), the crossflow is the most commonly used. They are preferred in convectional-type drying machines, where the transfer of heat and mass results from the contact of hot air on one side with the humid textile on the other side. Also, fresh ambient air and exhaust gases don't mix in this type of heat exchanger because the air and flue gases flow through different channels (Shekh and Saiful, 2013).

Several studies on waste heat recovery systems have been conducted. Waste heat recovery technologies make use of heat exchangers in the waste heat recovery units. Waste heat recovery technologies was studied by Jouhara et al. (2017). They investigated different methods of capturing of waste heats and the associated recovery technologies. They argued that systems with air preheaters were found to be very effective in extracting heat from exhaust systems when used for low to medium temperature utilizations needs.

The techno-economic viability of waste heat utilization for chilled water production and heating applications using biomass in the hospitality industry was carried out by Godawitharana and Rohitha (2014). It was concluded that, running generators only for electricity consumption is not viable due to the high cost of fuel and that its exhaust temperature which is quite high could be extracted in a heat recovery system for the utilization of absorption chillers, that way dependence on grid power for air conditioning would be reduced. Nigerians depends heavily on petrol generators for their energy needs, this no doubt contributes to the worrisome climate change issues because of the exhaust heat fumes been flared to the atmosphere.

Akindele and Adejumobi (2017) studied the livability milieu of residents and domestic electric power generator usage in Ogbomosho, Nigeria. The questionnaire administered to over 500 respondents used a multistage approach, findings revealed that the high level of electric generator usage was due the high level of incidence of power outage.

Chuen-Sen (2008) studied the recovery of heat energy from a diesel engine exhaust system. The study reports that the composition of the flue gas, the exhaust flow rate and its temperature affect the choice of the heat recovery system to be used and the quantity of recoverable heat.

Alkalabi et al. (2021) studied waste heat utilization from a dual powered system which used internal combustion engines for power generation and an organic Rankine cycle for refrigeration, their results show that utilization of the waste heat results in up to 13% reduction in the specific fuel consumption.

In their paper titled “theoretical analysis of waste heat recovery from an internal combustion engine in a hybrid vehicle”, Diego et al. (2006) conducted a theoretical study on using the waste heat from an engine to generate electric power, they used three different configurations of the Rankine cycle and examined the constraints of each of the configurations

In contrast to past works in literature, which focused on mobile waste heat energy source, this article aims to carry out experimental study on waste heat recovery from stationary petrol generators for use in domestic applications i.e. textile drying

2.0 METHODOLOGY

The materials deployed for this study include a 1.5kVA petrol generator, medium density fiber (MDF) board to produce the clothes drying cabinet, copper tubes and aluminum absorber plate for heat recovery unit, digital sensors for measuring temperature, pressure, humidity, and air speed respectively, force draught fan to pressurize fresh ambient air.

2.1 Experimental Setup and procedure

The experimental prototype waste heat recovery facility is schematically illustrated in Figure 1.

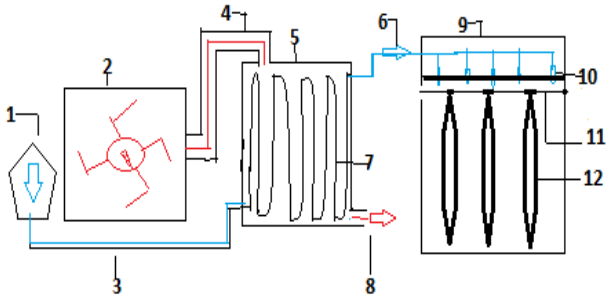


Figure 1: Schematic of the experimental set up

1- force draught fan; 2- petrol generator; 3- air pipe; 4- heat resistant flexible connector; 5- heat exchanger; 6- hot air inlet to drying chamber; 7- copper tube; 8- generator exhaust gases; 9- drying chamber; 10- air vents; 11- clothes hanging line; 12- clothes.

The petrol generator was run at full load (i.e. 6.8 A and 2850 rpm), its hot exhaust gases are channeled into a shell and tube heat exchange box, where the hot gases flow through the shell and pressurized ambient air flows inside copper tubes in a counter-flow arrangement. The air recovers heat from the exhaust gasses before entering the drying chamber through air vents at the top of the chamber where it circulates to carry out the drying. A force draught fan is used to maintain the constant flow of pressurized air. The copper tube is held in place on black-painted aluminum plate, this serves as a heat absorber for retaining heat. It also increases contact surface area of the copper tube to enhance heat exchange between the exhaust gases and air. The experimental rig is operated using specific clothes weight, the time taken for the clothes to get dried is noted and recorded for each test run. The weight of the clothes before and after drying are recorded to obtain the amount of moisture removed from the clothes. Plates showing experimental setup are in appendix A

2.2 Sizing the Heat Exchanger Tube

Table 1 presents the assumptions and initial design conditions used to ascertain the length of heat exchanger copper tube required for the study. The equations 1 to equation 110used in this section was adapted from Cengel (2018).

The quantity of available waste heat from the generator is given as

$$Q_{ex} = V \text{ x } \text{A x } \rho \text{ x } C_p \text{ x } \Delta T \quad (1)$$

Where, Q_{ex} = heat available in the exhaust gas

This is the maximum possible heat transferable heat from the exhaust gases. However, the actual rate of heat transfer to the air inside the heat exchanger was determined using equation 2.

Table 1: Assumptions and initial design conditions

Power rating of petrol generator	1.5kVA
Temperature of exhaust gas leaving the generator (T_{og})	238°C (measured)
Ambient temperature ($T_a = T_{air}i$)	28°C (measured)
Temperature of exhaust gas leaving the heat exchanger (T_{go})	102°C (measured)
Temperature of air from the heat exchanger (T_{airo})	88°C (measured)
Temperature of air from the drying chamber (T_{airoc})	35°C (measured)
Velocity of air (V_a)	1.5m/s (measured)
Velocity of exhaust gases (V_g)	2.0m/s(measured)
Relative humidity (R_h)	52% (measured)
Diameter of exhaust gas tube	2cm
Density of exhaust gas (ρ)	1.2931kg/m^3 (determined)
Specific heat capacity of exhaust gas	1.185kJ/kgK (determined)
Specific heat capacity of air	1.00kJ/kgK (assumed)
Density of air (ρ)	0.8871 kg/m^3 @ 88°C
Diameter of copper tube	2cm

The heat transfer rate in a heat exchanger is given by

$$Q_{air} = \mathrm{UA}\Delta\mathrm{T_m} \quad (2)$$

Where, ΔT_m (the log mean temperature difference) is expressed as in equation 4

$$\Delta\mathrm{T_m} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (3)$$

Neglecting the resistances due to tube wall thickness and due to exhaust scales, the overall heat transfer coefficient (U_o) based on the outer surface is given as:

$$U_0 = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}\ \mathrm{A_O} \quad (4)$$

Based on the inner surface, the overall heat transfer coefficient (U_i) is given as:

$$\mathrm{U_i} = \frac{1}{\frac{1}{h_i} + \frac{r_i}{k}\ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_i}{r_o}\right)\mathrm{x}\frac{1}{h_o}}\ \mathrm{A_o} \quad (5)$$

Where, h = average heat transfer coefficient, h is defined as

$$\mathrm{h} = \frac{k}{d}\,\mathrm{Nu} \quad (6)$$

Nu is defined as the Nussett Number and is given by:

$$\mathrm{Nu} = 0.023(\mathrm{R_e})^{0.8}(\mathrm{P_r})^{0.33} \quad (7)$$

A_o (cross sectional area of the copper tube) is obtained using equation 9

$$A_0 = \pi d_0 l \quad (8)$$

The length of the copper tube, which was determined from equations 3 and 9 is given as.

$$l = \frac{Q_{air}}{U\pi d_0 \Delta T_m} \quad (9)$$

The nature of the air flow (turbulence or laminar) through the heat exchanger must be determined using the Reynolds number equation below

$$R_e = \frac{\rho \, V_{avg} D}{\mu} \quad (10)$$

Where,

V_{avg} = average flow velocity (m/s) of air = 2.1m/s (measured)

ρ = density of air = 0.8871kg/m^3 @ 117^{0}C .

D = length of the drying chamber (diameter of tube in m) = 0.012m

μ = kinematic viscosity of air @ 117^{o}C = 2.1376 x 10^{-5} kg/ms) .

Table 2: Average sizes of fabrics

Fabric type	Length (m)	Width (m)
Shirts	0.75	0.47
Trousers	0.85	0.32
Singlets	0.55	0.40
Napkins	0.35	0.18

2.3 Sizing the Drying Chamber

The sizing of the drying compartment was determined by factoring the number and average sizes of the clothes to be dried. Sample sizes for shirts, trousers, singlets, and napkins used in the experimental study are tabulated in table 2, these average sizes were used in determining the dimensions (height, depth, and width) of the drying chamber.

3.0 RESULTS AND DISCUSSION

Results from temperature measurements taken at intervals showed that the average temperature of the exhaust fumes from the generator exhaust after been in operation for up to 30 mins was 238^{0}C, this temperature falls within the range suitable for waste heat recovery (100^{0}C-400^{0}C) as reported by Jouhara et al. (2017). A 42% loss in weight was recorded during the time indicate.

The average temperature and the relative humidity in the drying chamber is compared to that of the ambient conditions in Figure 2. As seen in the figure, temperature in the drying chamber at the start of the drying process was lower than the ambient temperature. This is due to the much higher moisture content of the wet fabric which tends to bring down the drying chamber temperature thereby increasing the chamber's relative humidity. The drying chambers relative humidity was noticed to progressively increase from the beginning of the drying process till about the mid time of drying, from that point, it began to decreases till end of the drying process, in all of this, the drying chambers relative humidity is always higher than that of the ambient, this may be attributed to the fact that whereas there is continuous loss of moisture from the fabric to sustain the relative humidity, that of the atmosphere is conditioned by the intensity of the sun i.e. decreases during the day (high sun intensity) and increases at sunset (little or no sun intensity)

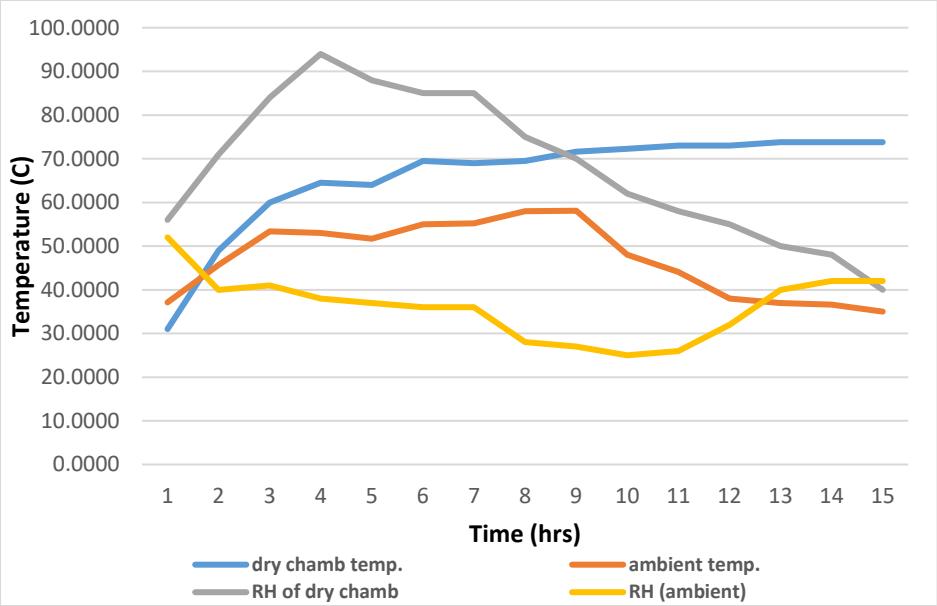


Figure 2: Temperatures and relative humidity of drying chamber and atmosphere

3.1 Effect of Tube Length on Heat Transfer and Effectiveness

Figure 3 shows the variation of tube length with heat transfer rate. As seen from the figure, the increase in length of the tube leads to an increase in the rate of heat transfer across its length and consequently the effectiveness of the heat exchanger is improved. Therefore, increased length of tube increases the effectiveness. the increase noticed in the effectiveness isn't very significant and is even noticed to decrease slightly at some points due to heat losses along the tube length which at some point would void any heat transfer gain by the cold fluid.

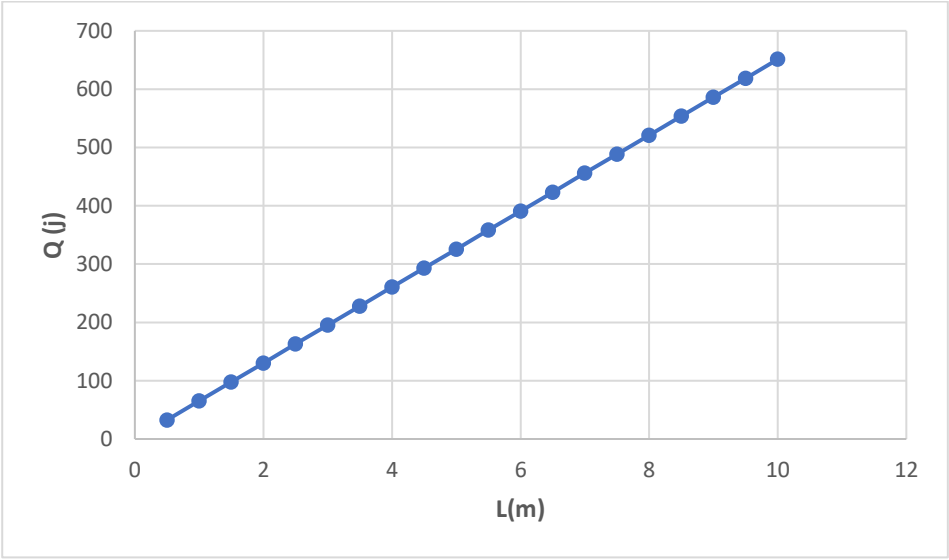


Figure 3: Increase in heat transfer with tube length

3.2 Effect of Tube Diameter with Heat Transfer Coefficient

Variations of tube diameter (outer and inner diameter) with overall heat transfer coefficients is shown in figures 4 and 5. It can be seen that, when the outer diameters increase, the heat coefficient reduces while the reverse is seen for the inner diameter where the heat transfer coefficient increases with increased diameter. The internal diameter of the pipe increases as the air flow rate in the tube increases. The effect of increasing the inner and outer diameters of the tube will lower the effectiveness of the heat exchanger.

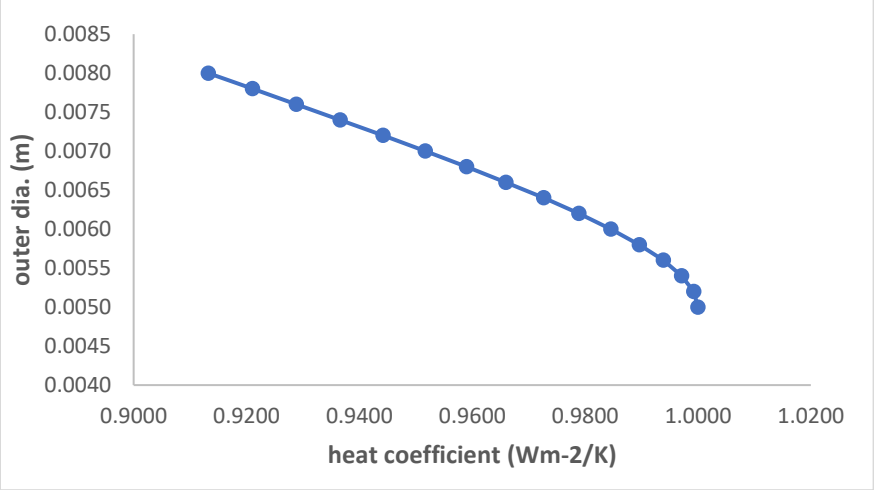


Figure 4: Effect of increasing the tube outer diameter on heat coefficient

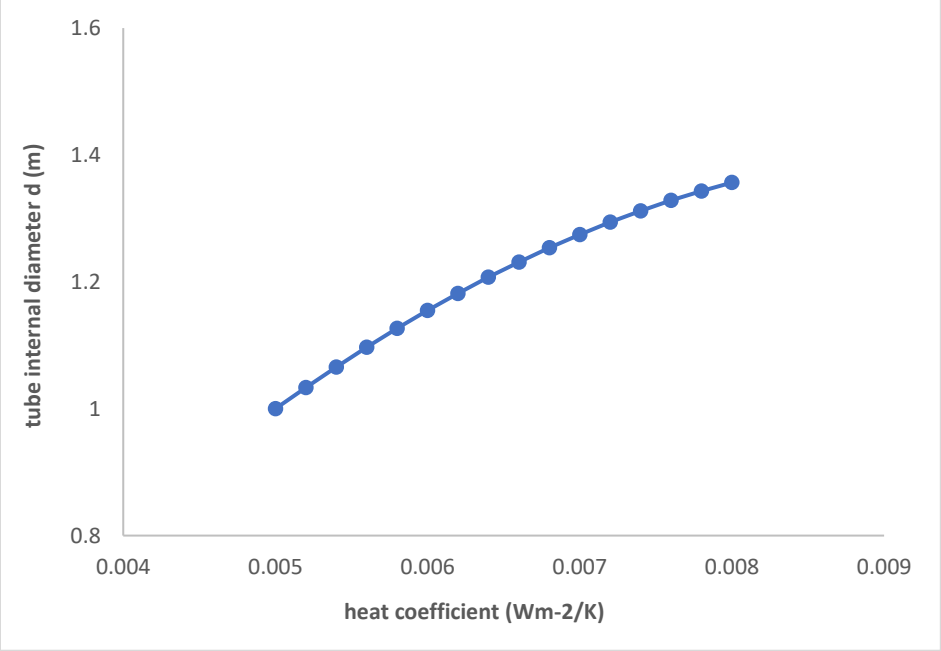


Figure 5: Effect of increasing tube inner diameter on heat coefficient

3.3 Effect of Overall Heat Transfer Coefficients on the Average Heat Transfer Coefficient

The variation of average and overall heat transfer coefficient is shown in figure 6. The increasing the average heat transfer coefficient h leads to a corresponding increase in the overall heat transfer coefficient. Increasing both the overall and average heat transfer coefficient consequently leads to an increase on the heat transfer rate Q between the heating fluid in the generator exhaust and the drying fluid (air).

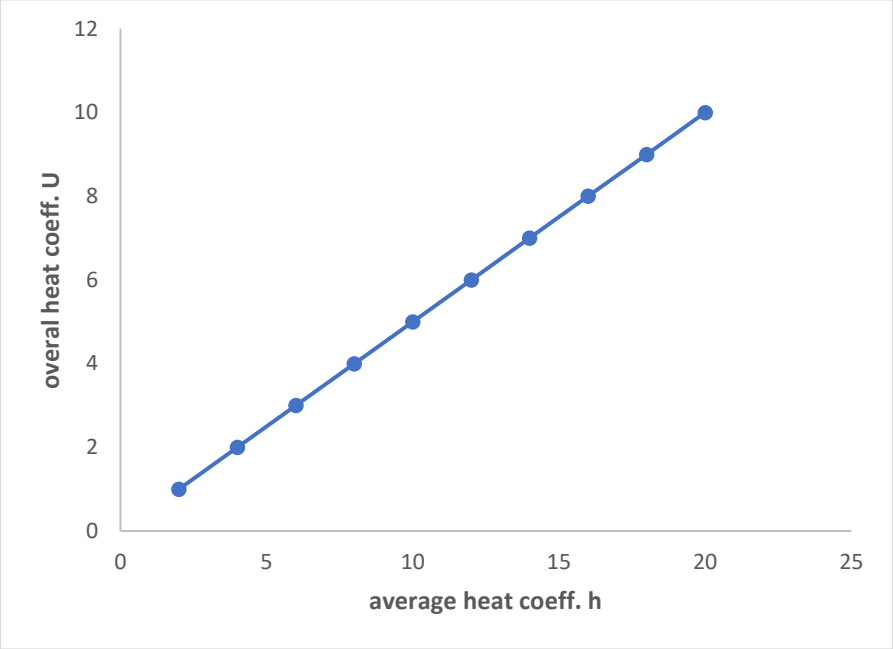


Figure 6: The effect of overall heat transfer coefficient on the average heat transfer coefficients

3.4 Effect of drying chamber temperature on output heat transfer Q

Figure 7 shows the variation of drying chamber temperature on the heat transfer rate. The temperature in the drying chamber is affected by the moist cloths load. When the quantity of moist clothes in the chamber is increased, the relative humidity of the chamber also increases which results in a reduction of the internal temperature in the chamber (T_2) and in turn a decrease in log mean temperature (T_m), as a result of the temperature drop, the rate of heat transfer Q decreases Inference made from this is that the clothes should be well spurned before entering the drying chamber so as to remove as much water content as possible thereby keeping the heat transfer rate as high as possible for effective drying.

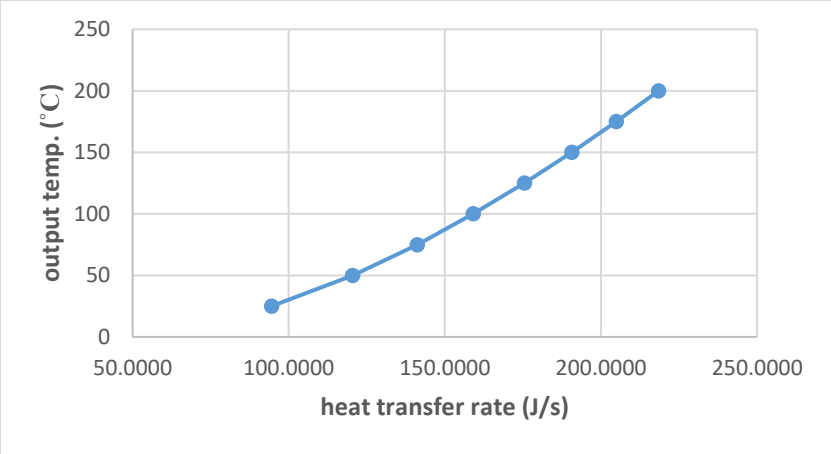


Figure 7: Effect of moist clothes load on logarithmic temperature and heat transfer rate

3.5 Comparing device drying and conventional drying

Figure 8 shows the comparison between the device drying and conventional drying. The rate of moisture loss for the sun-dried fabrics was higher that from the waste heat recovery drying device. The increased rate from the conventional air drying is due to the presence of excess warm air current which impinges on the wet fabric, thereby driving out the moisture at a faster rate, this is not so in the waste heat recovery drying device as its only air flow is that from the heated air which is confined in the geometry of the device. Furthermore, the possibility of intermittent break in the waste heat recovery drying device due to refueling of the generator and the inability of generators to power for long hours. However, the advantage of improving the efficiency of the device by incorporating a waste heat recovery system makes the use of the drying device attractive.

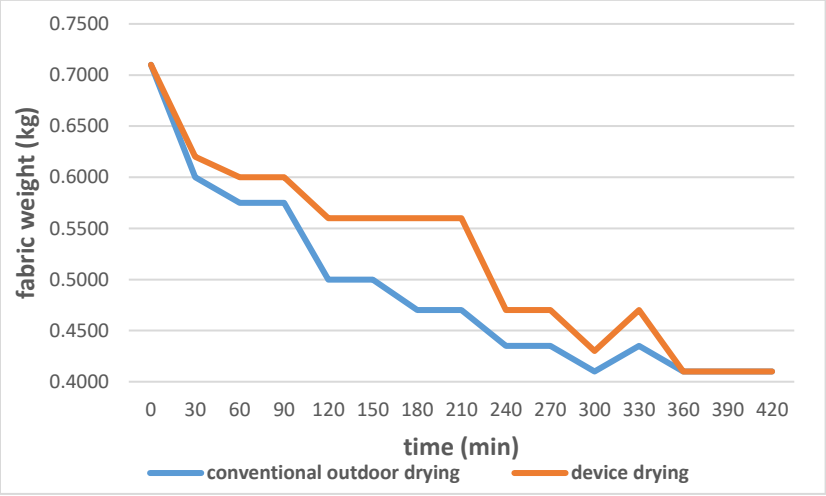


Figure 8: Comparison between conventional sun drying and waste heat drying

3.6 Dry Chamber and Heat Exchanger Effectiveness

The result of the heat exchanger and drying chamber from the test probe are presented in Table 4. The calculated effectiveness of the heat exchanger was 0.553, while the obtained effectiveness of the heat exchanger from test results obtained from the experimental setup was 0.517. The

drying chamber efficiency was 57.8%, the value was obtained from calculated figures from the test probe, these results is in good agreement with the 57% recorded by Aziz et al. (2016)

Table 4: Thermodynamic Parameters of Heat Exchanger and Drying Chamber

Parameter	Value
Average exit temperature from heat exchanger (^oC)	11.4
Average drying chamber (exit temperature) (^oC)	64.6
Exhaust temperature (input temp) from generator(^oC)	238
Assumed Heat Exchanger temperature (^oC)	125
Ambient temperature (^oC)	34
Highest temperature in the drying chamber (^oC)	73.8
Lowest temperature in the drying chamber (^oC)	31
Highest temperature in the Heat Exchanger (^oC)	132.5
Lowest temperature in the Heat Exchanger (^oC)	34.5
Average heat utilized from the heat exchanger (%)	37
Heat exchanger effectiveness (Theoretical) ε_t	0.554
Heat exchanger effectiveness (Actual) ε_a	0.518
Efficiency of drying chamber (%)	57.8

4.0 CONCLUSION

A heat recovery unit to recover waste heat from petrol generator exhaust gases for the purpose of clothes drying was developed. It was observed from experiments that the developed heat recovery unit was able to recover ample heat from the petrol generator exhaust gases to dry clothes. Clothes drying was recorded to be faster and cleaner in the drying chamber of the heat recovery unit than the conventional sun drying. Hence, it is obvious that this unit could be adopted for domestic and commercial use for drying of clothes irrespective of season and weather conditions.

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APPENDIX: Actual Experimental Setup of Waste Heat Recovery Drying Device



