DEVELOPMENT OF A CEREAL GRAIN DRYING SYSTEM USING INTERNAL COMBUSTION ENGINE WASTE HEAT

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ABSTRACT

The overall objective of this study was to minimize the environmental impact of agricultural operations by developing a system which uses engine waste heat for grain draining. Waste heat is drawn from the engine exhaust and the cooling system and used for heating drying air. The system heats ambient air to the desired drying temperature of 55^0 C. This waste is used to supplement methane-based fuel which is the main energy source for cereal drying. Grain is normally harvested at a moisture content of 32% wet basis but can be higher or lower depending on factors such as the stage of maturity, season, weather pattern as well as drying facilities. For maize, the traditional approach is to leave the crop in the field until the moisture content has fallen to around 20%. Changes in weather patterns, however, have resulted in farmers' inability to sufficiently dry their cereals leading to frequent cases infection with aflatoxins and consequent disposal of affected maize. In this study and analysis of a 223.71kW diesel engine with 81.33% of load, 599.9℃ exhaust temperature and 244 brake horsepower were used. The proposed heat exchanger was found to have effectiveness of 92% and 93% for energy from the exhaust and coolant, respectively. Energy harnessed from the coolant and exhaust was used to heat ambient air to 55^0C which was then directed into the cereal dryer for maize gains drying. The dryer has a batch drying capacity of 40,000 kg for one drying operation in an hour and some of the exhaust air is redirected into the dryer as it was still considered to have sufficient heat to dry the maize in the chamber.

Keywords: Waste heat recovery, internal combustion engine (ICE), cereal dryer, coolant, exhaust. Waste heat recovery.

1. INTRODUCTION

Today, food production and processing has become an industrialized function with mechanization and application of modern technology being necessary for efficiency and cost reduction. For food to last longer and remain safe and preserve quality, affordable but effective drying is necessary (Rivier, Collignan, Meot, Madoumier, & Sebastian, 2018). Moisture content of cereal grains is a very important physiological factor in successful storage with minimal quality loss. This is because high moisture content leads to fungal and insect infestation , grain respiration and germination (Pekmez, 2017). Drying and storage are a part of food production system consisting of two subsystems, crop production and post-harvest operation. The reduction in post-harvest losses depends on the proper threshing, cleaning, drying and storage of the crops. Global food security will be enhanced by increasing food production and reduction of post-harvest and storage loss of grains and other crop products(Bala, 2017).

Loss of food through post-harvest handling is not a new problem to humanity and significant quantities of food is lost after harvesting especially in developing countries(FAO, 1983). Maize for example, is the most important cereal crop for sub-Saharan Africa. Maize is eaten in various forms including maize meal and processing of other industrial products like starch, oil, and animal feeds. It is high yielding, readily digested, easy to process and comparably cheaper than most other cereals available in the market. It can also be processed into different forms, most popularly in the form of maize meal which makes up the staple food for a huge number of people in Africa. Effective management of post-harvest factors such as moisture content, temperature, sanitation of storage facilities and aeration go a long way towards preserving grain and keeping it at safe for human consumption. When there is need to dry maize to the ideal moisture content for storage, further drying by use of artificial means becomes necessary. Drying is carried out by heating ambient air using large furnaces that are powered by methane-based fuels such as furnace oil or diesel. This, without doubt tends to be a very expensive venture for the farmer and can be evaded if more affordable and convenient methods which are at the farmers' disposal are employed in cereal drying(Bala, 2017).

One of the most common type of farm machinery used is engine driven tractors which are powered. Other common applications of diesel engines is use as prime mover of farm machinery like combined harvesters, dyers, and others(Kabeyi, 2020b). Diesel engines are widely used globally because of their high-power density and efficiency

compared to petrol or spark ignition or petrol and gas engines (Kabeyi & Oludolapo, 2020b; Mitzlaff, 1988; Wilson, Singh, Singh, & Subramanian, 2017). Internal combustion engines dissipate significant amount of energy in terms of heat through exhaust gases and the coolant. This waste heat is a potential source of energy and can be s harnessed by various devices via heat exchangers currently. Several thermodynamic cycles have been researched as potential methods for heat recovery and the analysis and comparison of each method has resulted in the organic Rankine cycle (ORC) as the most ideal candidate in terms of efficiency. The most important developmental field for the ORC is waste heat recovery, where a low boiling point organic fluid is used as working medium (Kabeyi & Oludolapo, 2021; Oludolapo & Kabeyi, 2020). The ORC can be applied to heat and power plants or to industrial manufacturing processes such as die casting(L. Rettig, Lamare, Li, Mahadea, McCallion, Chernushevich, 2011). The ORC can, however, be expensive to install despite its reputable efficiency which makes other waste heat recovery systems ideal for waste heat recovery for a farmer like the Rankine cycle (Kabeyi, 2019, 2020a). Other alternatives that can be sought include the use of heat exchangers which are way more affordable and easier to maintain. The application of waste heat recovery in drying cereal would mean that as the internal combustion engine performs another task such as shelling corn, as the waste heat is harnessed and channelled towards drying the cereal. This, without doubt, leads to higher efficiency is realised on the farm in addition to savings on transportation, labour, and drying costs which farmers have incurred for years in the past(Oludolapo & Kabeyi, 2020).

In this research, a design model is proposed to use engine waste heat from the exhaust and cooling systems to reduce load and demand on the conventional heat source like natural gas, where the dryer is engine driven. This will lead to cost reduction, reduced environmental impact by using waste heat to prevent further consumption of conventional fuels and hence improve sustainability in food production.

1.1. Research Problem and Justification

Cereals drying allows it to have long period of storage time, hence continues food supply over a long period of time throughout the year, permits early crop harvest hence reduces field losses and better-quality produce which also enable off season food supply(Tamil Nadu agricultural University, 2017).Wet grains would heat up, develop fungi and even germinate in storage causing huge losses(Shareweb, 2019). For safe storage, grains have to be dried t0 10 -15% moisture content(Reykdal, 2018).

1.2. Drying Theory

Drying refers to the process of controlled heat addition to remove water from a moist material(Reykdal, 2018). Drying is technically a convection process that removes moisture from a material. The dryness of an agricultural product is a function of its moisture content. This content can reduce, or increase based on environmental condition in terms of vapour pressure difference between the product and the atmosphere(Tamil Nadu agricultural University, 2017). Moisture content is the quantity of water in a substance per unit mass expressed as a percentage. Drying will continue as long as surrounding air can absorb more water hence the need control humidity and air movement(Shareweb, 2019). . Movement of moisture takes place due to

i. Capillary flow – Liquid movement due to surface forces
ii. Liquid diffusion – Liquid movement due to difference in moisture concentration
iii. Surface diffusion - Liquid movement due to moisture diffusion of the pore spaces
iv. Vapour diffusion – vapour movement due to moisture concentration difference
v. Thermal diffusion - vapour movement due to temperature difference
vi. Hydro dynamic flow – water and vapour movement due to total pressure difference

The drying process can be classified into thin layer drying and deep bed drying based on thickness of the grain or material being dried. Dryers can be classified into many types based on mode of drying and contact with drying medium. They can be classified as continuous flow dyers (mixing and non-mixing), baffle dryer, which is continuous mixing or non-mixing dyer, recirculatory baffle dryer, pneumatic dryers, fluidized bed dryers and rotary dyer. Commonly used drying air temperature 54°C while air flow velocity is 60-70 m^3/min/tonne(Tamil Nadu agricultural University, 2017).

Maize is best stored with moisture content of 13.5% percent while surrounding air temperature should range between 25ºC and 30ºC and relative humidity of 70%. Corresponding moisture content for beans is 13.5%, sorghum 13.5%, rice with hask (paddy) is 15%, rice without husk or milled rice is 13.0%(Shareweb, 2019). Barley is harvested with moisture content of up to 30% and is dried to between 12% and 14% moisture content for long and short storage respectively (Reykdal, 2018). The maximum drying temperature depends on application, maize for milling has maximum drying temperature of 60ºC, grains for sowing 40 ºC, rice for food 45 ºC and beans for food 35ºC. while 35-

38°C is recommended for malting barley but 40ºC is optimum for seed barley. The maximum for feed barley is 80 - 100ºC. Low temperatures are recommended for too wet or moist grain.(Reykdal, 2018; Shareweb, 2019).

2.0.CONCEPTUAL FRAMEWORK FOR A GRAIN DRYING SYSTEM

The conceptual framework of this design depicts both the flow of harnessed waste heat energy as well as maize in a 2-dimensional representation. Figure 1 below demonstrates the conceptual framework. Figure 1 demonstrates the proposed model of a cereal's dryer.

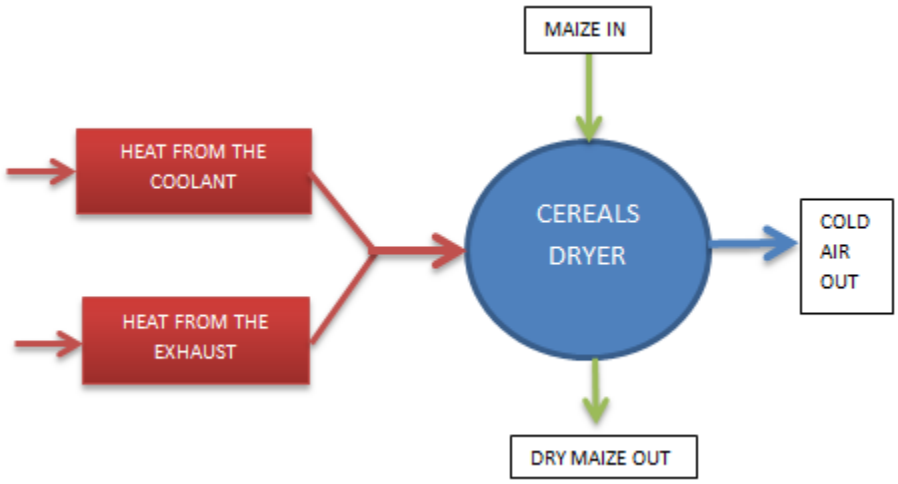

Figure 1: Conceptual framework

From figure 1, it is noted that energy can be recovered from the cooling system and the exhaust and used to dry maize cereals. Maize is moved across the dryer where it is heated to remove moisture and reduce it to acceptable level.

Internal combustion engines have been a primary source of power for automobiles for the past century. They convert a little over 30% of energy in a mechanical form. Energy which is not converted to mechanical form is dissipated as heat via different streams such as combustion gas, coolant, and lubricant. Exhaust gases are the most thermodynamically attractive among the heat loss streams because their temperature ranges between 500 ^{0}C and 700 ^{0}C(Bianchi, 2015). These gases result in entropy rise and severe environmental pollution hence a need to put the waste heat to useful work.

Excellent engine performance is dependent on a certain safe and satisfactory operating temperature range. The risks of high engine temperatures are reduced oil viscosity, leading to engine parts such as pistons not moving freely and, consequently, loss of power; wear and tear, burnt gaskets, and eventually, metal-to-metal contact. Increased oil consumption can be caused by loss of oil lubricity. However, fuel vaporization is required for complete fuel combustion, and at low engine temperatures, incomplete combustion can happen, leading to excess fuel requirements for proper engine performance, due to improper vaporization. Moisture from combustion can also mix with the unburnt hydrocarbon fuel forming acidic mixtures which can cause acidic corrosion to the engine metal. This can lead to engine damage(Bari & Hossain, 2013; Kabeyi & Oludolapo, 2020a).

It is estimated that 25% to 35% of the heat supplied in the fuel is removed by the cooling medium, and that lost by lubricating oil and by radiation is 3% - 5% bringing the total average heat loss to 34%(Kabeyi & Oludolapo, 2020b, 2020c; Oludolapo & Kabeyi, 2020). This is a substantial loss in engine power that might affect engine performance(Jack & Ojapah, 2013). The properties of interest in a water coolant are the density, specific heat at constant pressure and viscosity. The high specific heat of water offers advantages in avoiding thermal overloads due to excessive component temperatures. The ease with which water flows is due to its low viscosity and its ability to take and release heat from an engine makes it an ideal coolant. Water used for engine cooling is mostly a mixture of drinking water and antifreeze with some additives such as ethylene-glycol. Additives act as inhibitors to protect against rust and/or corrosion(Mitzlaff, 1988; L. L. Rettig et al., 2011). Table 1 below shows a comparison of water and water/ethylene -glycol mixture as engine coolant.

Table 1: Physical Properties of Water compared to Water/ethylene-glycol mixture (Jack & Ojapah, 2013)

Property	Water	Ethylene-Glycol/Water mixture
Boiling Point at 1 bar	100 °C	111 °C
Specific Heat, kJ/kg K	4.25	3.74
Thermal conductivity W/mK	0.69	0.47
Density at 20 °C, kg/m³	998	1057
Viscosity	0.89	4

From table 1. Shows water has a lower boiling temperature, higher heat capacity and thermal conductivity than a mixture of ethylene-Glycol. It also has lower viscosity and density.

3.0. METHODS AND MATERIALS

In order to show the basic working principles of the components of this cereal dryer design, the parts were also deeply discussed. The selection of materials and methods brought about the consideration of some certain properties: mechanical properties, thermal properties, and chemical properties. In this study, a cereals drier using waste heat from the cooling and exhaust systems is used to dry cereals. The research design is design and analysis and uses technical data and specify the proposed model and system(Barasa, 2018; Jeremiahi, 2019).

3.1 Overview of the design

Freshly harvested cereals are not always dry to a moisture content that can be considered suitable for marketing or long-term storage hence the need for drying to achieve the target safe storage moisture content. In the process of drying cereal, heat is required to evaporate moisture from the grain and air flow is needed to carry away the evaporated moisture. The two basic mechanisms involved in the drying process are the migration of moisture from the interior of an individual grain to the surface, followed by the evaporation of moisture from the surface to the surrounding air(FAO, 1994). The rate of drying is determined by the moisture content of the grain, temperature of the air in contact with the grain, relative humidity, and the velocity of the air in contact with the grain(Muthukumarappan, 2016).

The hot coolant leaving the engine jacket as well as the hot exhaust gases in the manifold were directed into a shell and tube heat exchanger. The coolant and exhaust gas flowed inside the shell while air was blown inside the tube.(Jack & Ojapah, 2013) The hot coolant and exhaust gases heated up the cold air in the tubes of the heat exchanger, which was blown in tubes of heat exchangers. The final and desired temperature of the dry air to be channeled into the dryer reached 55 ^{0}C. After drying cereal, some of the exhaust air from the dryer was expelled into the atmosphere while some of it was taken back to the column of the dryer as it was considered to have a good amount of heat that could dry the cereal further. In addition, this technique helped to reduce heat loss to the environment. Figure 2 below shows the proposed model of the dryer that uses the engines waste heat for cereal drying.

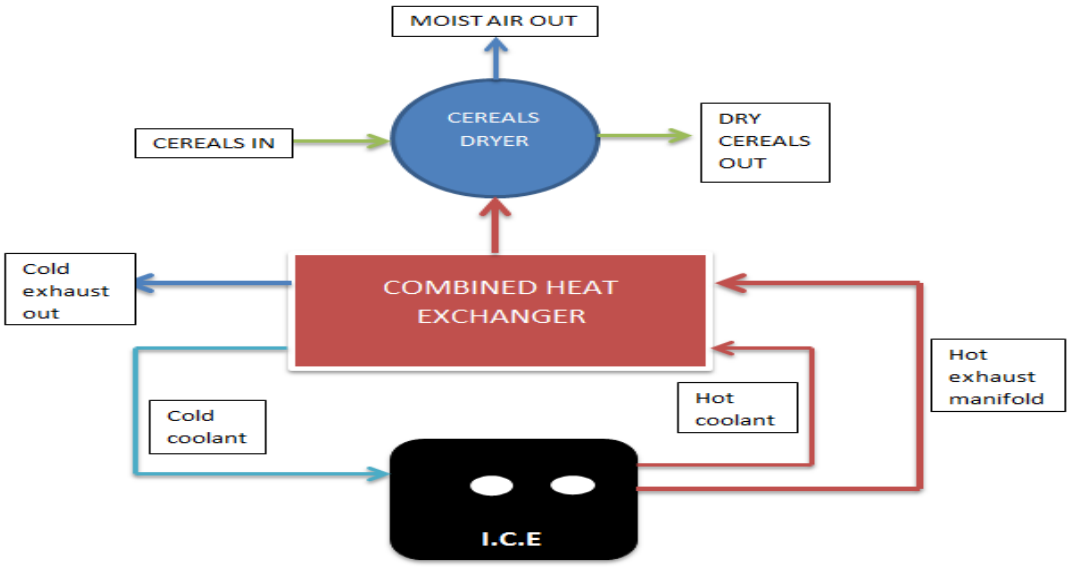

Figure 2: Proposed grain dying model.

Figure 2 is the proposed cereals dryer where heat is extracted from the exhaust and engine coolant via heat exchangers to heat up drying air for the cereals in a dryer.

3.2 Combined heat exchanger

Table 2 below is a table of materials that would be used in the design of the heat exchanger.

Table 2: Table of Materials and Equipment for the Heat Exchanger

Part	Material	Reason for choice
Shell, tubes, baffles, and header	Galvanized steel sheet	High thermal expansion coefficient, High thermal conductivity, machinability is easy and resistant to corrosion

The nominal diameter of the tubes made of galvanized steel would be 1 inch in diameter, as per the tubular standards of TEMA. The layout pattern of the tube is shown in appendix B. All the other dimensions of the specific parts were determined by the calculations below.

i.) Bulk temperature.

$T = \frac{Ti + To}{2}$, Where: Ti = inlet temperature of fluid, To = outlet temperature of fluid

ii.) Mass flow rates.

Referring to the energy balance equation:

$Q = \dot{m}C_p\,\Delta T$, We obtain the mass flow rate as follows $\dot{m} = \frac{Q}{Cp \times \Delta T}$

iii.) Determining Logarithmic Mean Temperature Difference (LMTD)

For the two heat exchangers, the flow was concurrent, therefore:

For engine coolant heat exchanger:

$\Delta T_{lm} = \frac{(Tci - Tao) - (Tco - Tai)}{ln\frac{(Tci - Tao)}{(Tco - Tai)}}$, For exhaust gas heat exchanger:

$\Delta T_{lm} = \frac{(Tei - Tao) - (Teo - Tai)}{ln\frac{(Tei - Tao)}{(Teo - Tai)}}$, Where: T_{ei} – temperature of inlet exhaust gas, Teo – temperature of outlet exhaust gas, Tci- Temperature of inlet coolant, Tco – Temperature of outlet coolant, Tao – temperature of inlet air T_{ai} – temperature of outlet air

iv.) Determining heat transfer area and tube count.

The equation below shows the heat transfer area.

$A = \frac{Q}{UoassFt\Delta Tlm}$ Where: Q = heat transfer / time, $Uoass$ = assumed value of overall heat transfer coefficient Ft = Correction factor and ΔT_{lm} = Logarithmic mean temperature difference. LMTD correction factors were read from graph.

The following factors were considered in the selection of the number of tubes.

a) Standard diameter, d = 1 inch
b) Pitch, Pt = 1.25d
c) Design length, l
d) Wall thickness as defined by Birmingham wire gauge

Tube count (N) is calculated as follows:

$N = \frac{A}{\pi dl}$ where, Where: A = heat transfer area, d= tube outside diameter and l = length of tube

v.) Determination of heat transfer coefficients of tube side and shell side

Tube side fluid (h_i)

$hi = 0.023(Re)^{0.8}(\mathrm{Pr})^{0.36}$, Reynolds' number = $\frac{4\dot{m}}{N\pi d\mu}$

Shell side fluid (h_o)

$$ho = 0.021(Re)^{0.84}(Pr)^{0.36}(\frac{Pr}{Prw})^{0.25}$$

Reynolds' number = $\frac{\rho Umaxdi}{\mu}$

Hence, the convective heat transfer helped to get the overall heat transfer coefficient.

$$Uo = \frac{1}{\frac{ro}{ri} x \frac{1}{hi} + \frac{ro}{ri} Rfi + \frac{roln\,(ro/ri)}{k} + Rfo + \frac{1}{ho}}$$

Where, h_f - tubes convective heat transfer, h_o - shell convective heat transfer, R_{fi} - fouling factor of the fluid in tube, R_{fo} - fouling factor of the fluid in shell

The difference between the assumed overall heat transfer coefficient and the calculated overall heat transfer would have been acceptable only if it were less than 30%. If not, some changes would be made until it was within the allowable limit.

vi.) Determination of the heat transfer overdesign

% overdesign = $\frac{area\ with\ standard\ tubes - area\ with\ required\ tubes}{area\ with\ required\ tubes}$ x 100

Finding the shell thickness

$t_s = \frac{PR}{\sigma j - 0.6P}$ + corrosion allowance, where: P = working pressure, R = Baffle spacing, j = the efficiency of the joint usually 0.8 and σ = maximum allowable stress for steel

vii.) Tube sheet thickness determination

The minimum tube sheet thickness to resist bending is given by.

$t_s = \frac{FDs}{3}\sqrt{\frac{P}{k\sigma}}$, Where: F = 1 for fixed tube sheet and k= 1 - $\frac{0.785}{(\frac{pt}{do})^2}$

Vital parameters like baffle spacing were considered in shell tube heat exchanger design. The baffle spacing was 0.5Ds and the cut 25% of Ds.

viii.) Pressure drop calculation.

Tube side pressure drop.

$\Delta P = f\, x\, \frac{Lt\rho V^2}{D}$

Shell side pressure drop.

$\Delta P = \frac{2fG^2N}{\rho}\, x\, (\frac{\mu w}{\mu b})$ x 0.14

For the two exchangers (combined), pressure drop was calculated separately to check if it was within the allowable fluids' pressure drop.

ix.) Effectiveness of the coolant heat exchanger

Effectiveness, $\varepsilon = \frac{rate\ of\ heat\ transfer\ in\ heat\ exchanger}{maximum\ possible\ heat\ transfer\ rate} = \frac{q}{qmax} = \frac{\dot{m}hCph(Thi - The)}{(Thi - Tci)}$

3.3 Design Procedure of the Cereal Dryer

Designing a cereal dryer was the focus of this research project. The dryer would be an in-bin type with an exhaust pipe for waste heat recovery. The materials, properties and parameters that were determined to develop an efficient dryer is summarized I table 4 below.

Table 3: Table of Materials and Equipment for Cereal Dryer.

Part	Material	Reason for Choice
Drying chamber, Upper perforated lid, Exhaust air tube and dryer floor	Stainless steel	To minimize contamination

Table 3 shows the materials prescription for the drying chamber. To minimize contamination from oxidation and rust stainless steel is used.

i.) Determination of final weight of grain after drying.

The first step in designing the cereal dryer was to determine the final weight of the maize after drying using equation shown below.

$$W_f = W_i\, (\frac{100 - Mci}{100 - Mcf})$$

Where: W_f = final weight of maize after drying

W_i = initial weight of maize before drying

M_{ci} = initial moisture content of maize before drying

M_{cf} = final moisture content of maize after drying

ii.) Determination of moisture content

The moisture content, Mc (%), in maize was determined to establish the amount of moisture that was to be removed from the freshly harvested maize.

$M_c = \frac{100(Wi - Wf)}{Wi}$

iii.) Determination of bulk density of maize

Obtaining the bulk density (TW_m) of maize was by use of an empirical formula which relates bulk density with moisture content for maize shown below.

$TW_m = 0.7019 + 0.01676\ M_c - 0.0011598\ M_c^2 + 0.00001824\ M_c^3$

iv.) Amount of moisture to be removed.

The equation below shows the amount of moisture (M_R) to be removed in kg.

$M_R = M(\frac{Q1 - Q2}{1 - Q2})$

Where: M = mass per batch to be dried

Q1 = initial moisture content in maize as a fraction, Q2 = final moisture content in maize as a fraction

v.) Quantity of air to effect drying.

This is denoted Qa and is computed in kg using the equation below:

$Qa = (\frac{MR}{Hr2 - Hr1})$

Where: Hr2 = final humidity ratio from psychometric chart at 15% relative humidity & 55ºC

Hr1 = final humidity ratio from psychometric chart at 70% relative humidity & 25ºC

vi.) of air to effect drying.

Volume of air to effect drying (Va) is computed in m^3.

$Va = \frac{Qa}{\delta a}$

Where: δa = density of air in kg/m^3

vii.) Quantity of heat required for effective drying (Hr) in kJ

$Hr = (M \times H_k) + (H_L \times M_R)$

Where: M = mass of maize to be dried per batch

H_L = Latent heat of vaporization = 1248.1 kJ/kg

$H_K = C_T (T_2 - T_1)$

C_T = Specific heat of maize = 1.8 kJ/kg ºC

T_1 = Ambient temperature = 25 ºC

T_2 = Drying temperature = 55 ºC

viii.) Actual heat used to effect drying.

$H_D = Ca\ Tc\ M_R$, Where: Ca = Specific heat capacity of air = 1.005 kJ / kg, Tc = Temperature difference

ix.) Design of the drying chamber (dimensions)

This was to be calculated from the bulk density of maize obtained. The volume occupied by 1kg of maize was determined and thereafter the volume occupied by the quantity of maize to be dried per batch. Since the dryer was to be cylindrical, using the formula for the volume of a cylinder,

$$V = \Pi r^2 h$$

The height and diameter of the drying chamber were determined.

x.) Results and Discussions

The information from Mantrac Kenya Limited (appendix G), Tetrapak and National Cereals and Produce Board in Donholm was utilized in computational design of the rest of the subsystems, which are, the combined heat exchanger and the design of the cereal's dryer as per the research objectives and with the accordance to the TEMA standards. The feasibility of the project was then determined through cost benefit analysis and at the end, discussions were made to see if the objectives were successfully met.

3.4. Design of the waste heat recovery system

The proposed recovery system consisted of two heat exchangers combined using pipes. One heat exchanger is for preheating using the coolant from the engine before the radiator. The second heat exchanger was for final heating of air using the hot exhaust manifold. The hot air was then directed to the designed cereals dryer.

i.) Air preheating heat exchanger

DATA

Coolant inlet temperature, T_{ci} 100°C

Coolant outlet temperature, T_{co}	30°C
Air inlet temperature, Tai	25°C
Air outlet temperature, T_{ao}	35°C (desired)
Allowable pressure drops of air	14kPa
Allowable pressure drops of coolant	68.9kPa

ii) Finding the mass flow rates

At 81.33% engine load, 78.30kW of heat is rejected to jacket water. Referring to energy balance equation mass flow rates are obtained as follows.

Energy balance equation:

$$Q = \dot{m}_a C_{pa}\, \Delta T_a = \dot{m}_c\, C_{pc}\, \Delta T_c$$

Mass flow rate of air = 7.79kg/s

Mass flow rate of coolant = 0.27kg/s

iii) Logarithmic Mean Temperature Difference (LMTD)

$\Delta T_{lm} = \frac{(100-35)-(30-25)}{ln\frac{(100-35)}{(30-25)}}$ = 23.39°C. Hence tube specifications are ;Single tube pass, 14BWG, OD = 25.4mm, ID = 21.184mm, Thickness = 2.108mm, Pitch P_t = 1.25OD = 31.75mm, Length of tubes = 4.2m

$A = \frac{Q}{UoassFt\Delta Tlm}$ And F_t is 1 for single tube pass

To initiate the design an overall heat transfer coefficient of 90w/m^{2}K was assumed.

$A = \frac{78300}{90 \times 23.39}$ =37.20m^2. $N = \frac{A}{\pi dl} = \frac{37.2}{\pi \times 0.0254 \times 4.2}$ =110.997 approx. 111tubes

A single tube pass heat exchanger, 1in OD and 1.25 in pitch, should have 112 tubes.

iv) Determination of heat transfers coefficients of tube side and shell side

Tube side fluid-air

Reynolds' number = $\frac{4x7.79}{111 \times 0.1846 \times 10^{-6} \times \pi \times 0.021184}$ = 2.28 × $10^{7,\ Re}$ > 2300 hence the flow is turbulent.

$hi = 0.023(Re)^{0.8}(\mathrm{Pr})^{0.36}$ = 15626 w/m^{2}k

Shell side fluid-water

Reynolds' number = $\frac{\rho Umaxdi}{\mu}$, $U_{max} = \frac{Pt}{Pt-di} \times U$ =1 m/s as the recommended minimum velocity of water, d = $\sqrt{\frac{4\dot{m}}{\rho\pi\mu}}$ = 0.019m = 19mm. Since Standard pipe nearest to 19mm is 20mm. Recalculating to get U.

$U = \frac{4 \times 0.27}{980.39\pi \times 0.02^2}$ = 0.8766m/s; $U_{max} = \frac{31.75}{31.75-21.184} \times 0.8766$ =2.634m/s;

$Re = \frac{980.39 \times 2.634 \times 0.0021184}{430 \times 10^{-6}}$ =1.2722x10^5 since Re>1000, the flow is therefore turbulent.

$ho = 0.021(Re)^{0.84}(Pr)^{0.36}(\frac{Pr}{Prw})^{0.25}$ Where P_{rw} = Prandtl number at the wall,

$Q = \dot{m}_a C_{pa}$ (Tw - T) = $\dot{m}_c\, C_{pc}$ (Tc-Tw) = 7.79 × 1.0049 (Tw-30) = 0.27 × 4.188 (65-Tw)

Where, Tw = 59.38°C, Prandtl number of waters at 59.38 degrees = 2.97

$ho = 0.021(1.2722 \times 10^5)^{0.84}(2.74)^{0.36}(\frac{2.74}{2.97})^{0.25}$ = 573.96 w/m^{2}k

Where: h_f - tubes convective heat transfer; ho - shell convective heat transfer; R_{fi} - fouling factor of the fluid in tube R_{fo} - fouling factor of the fluid in shell $Uocall = \frac{1}{\frac{25.4}{21.184} \times \frac{1}{15626} + \frac{25.4}{21.184}0.0072 + \frac{0.0127ln(25.4/21.184)}{30} + 0.0004 + \frac{1}{573.96}}$

U_{ocall} = 91.44 w/m^{2}k

Checking for error in the overall heat transfer coefficients: % error= $\frac{Uocal-Uoass}{Uoass} \times 100 = \frac{91-90}{90} \times 100$ = 1.6%

1.6 % < 30% (within the min allowable error). The heat transfer coefficient is acceptable.

v) Determination of the heat transfer overdesign

% overdesign = $\frac{112-110.997}{110.997} \times 100$ = 0.9%, 0.9% < 10% - typical min area overdesign.

The above design is therefore within the acceptable criteria.

vi) Finding the shell diameter

According to the TEMA standards, for 112 tubes the recommended shell diameter is 17.25in =17.25 × 25.4 = 438.15mm

vii) Baffle spacing and baffle cut

Recommended baffle spacing is between 0.4Ds - 0.6Ds, Baffle spacing = 0.5Ds = 0.5 × 438.15 = 219.075mm, recommended baffle cut is 25% and Baffle cut = 0.25 × 438.15 = 109.5375mm

viii) Finding the shell thickness

The working pressure is 200kPa, but the design pressure is taken as 10% higher than working pressure i.e. 1.1 × 200 = 220kPa
Maximum allowable stress (σ) for steel is 100MPa.
$t_s = \frac{PR}{\sigma j - 0.6P}$ + corrosion – allowance. Where, j is the efficiency of the joint usually 0.8.
Hence ts = (220 × 219.08) / (100× 10^3 × 0.8 – 0.6 x 220) + corrosion - allowance = 0.6035 + corrosion- allowance
And corrosion allowance for galvanised steel is 1.5mm i.e., t_s = 0.6035 + 1.5 = 2.1035mm
For standard sheet thickness, the nearest galvanised sheet metal of normal bore of 20mm, t_s=2.35mm
ix) Tube sheet thickness determination
The minimum tube sheet thickness to resist bending is given by.
$t_s = \frac{FDs}{3}\sqrt{\frac{P}{k\sigma}}$, Where: F= 1 for fixed tube sheet and k= 1 - $\frac{0.785}{(\frac{pt}{do})^2}$, k= 1 - $\frac{0.785}{(\frac{31.75}{25.4})^2}$ = 0.4976 Therefore
$t_s = \frac{1\times 438.15}{3}\sqrt{\frac{220}{0.4976\times 100000}}$ = 9.7mm approx. 10mm
x) Pressure drop calculation
Tube side pressure drop- air is ΔP=f × $\frac{Lt\rho V^2}{D}$
Friction factor for flow with Re= 2.28 x 10^7 and relative surface roughness ε/D= $\frac{0.002}{21.184}$ = 9.47 × 10^{-5} is 0.0085 reading from the Moody chart diagram. Therefore, ΔP=0.0085 × $\frac{4.2\times 1.177\times 30.24^2}{0.021184}$ = 1813.85Pa since 1.81kPa < 14kPa we deduce that Tube side pressure is within the max allowable pressure drop.
Shell side pressure drop - coolant (water)
ΔP = $\frac{2fG^2N}{\rho}$ × $(\frac{\mu w}{\mu b})$ × 0.14, f = $\frac{0.044+0.008(\frac{Pt}{D})}{(\frac{Pt-D}{D})^{0.43+1.13\frac{D}{Pt}}} \times Re^{-0.15}$, f = $\frac{0.044+0.008(\frac{31.75}{25.4})}{(\frac{31.75-25.4}{25.4})^{0.43+1.13\frac{25.5}{31.75}}} \times (1.2722\ x\ 10^5)^{-0.15}$,
f= 0.1567, Gmax = ρU_{max} = 2.634 × 980.39 = 2582.35 hence N = number of transverse rows = 6, μw= 463 × 10^{-6} at wall temperature of 59.38 degrees μb = 430 × 10^{-6} at bulk temperature
Therefore, ΔP=$\frac{2\times 0.1567\times 2582.35^2\times 6}{980.39}$ × $(\frac{463}{430})$ x 0.14 =1640.98Pa, therefore 1.64kPa< 68kPa so the Shell side pressure drop is within the maximum allowable pressure.
xi) Effectiveness of the coolant heat exchanger
Effectiveness, ε = $\frac{rate\ of\ heat\ transfer\ in\ heat\ exchanger}{maximum\ possible\ heat\ transfer\ rate}$, ε = $\frac{q}{qmax} = \frac{\dot{m}hCph(Thi-The)}{(Thi-Tci)}$

- Hot fluid – 0.27 × 4.188 = 1.13076
- Cold fluid – 7.79 × 1.0049 = 7.828171

Hence if, ṁcCpc > ṁhCph, then: ε = $\frac{Thi-The}{Thi-Tci} = \frac{100-30}{100-25}$ = 0.93
By definition, ε is a dimensionless quantity and should be between 0 and 1.
Final air heating heat exchanger
DATA

Exhaust inlet temperature, T_{ei}	599.9°C
Exhaust outlet temperature, T_{eo}	79°C(desired)
Air inlet temperature, T_{ai}	35°C
Air outlet temperature, T_{ao}	55°C (desired)
Allowable pressure drops of air	14kPa
Allowable pressure drop of exhaust	68.9kPa
Mass flow rate of air is	7.79kg/s

The properties of air and exhaust were taken at the bulk temperatures.
ii) Energy balance
Q = $\dot{m}_a C_{pa}$ ΔTa = $\dot{m}_e$ C_{pe} ΔTe where ṁe = 0.29kg/s
iii) Determining heat transfer area and tube count
Single tube pass, and for 14BWG; OD = 25.4mm, ID = 21.184mm, Thickness = 2.108mm, Pitch Pt =1.25OD = 31.75mm and Length of tubes = 4m

Logarithmic temperature difference is calculated as.
$\Delta T_{lm} = \frac{(599.9-55)-(79-35)}{\ln[\frac{(599.9-55)}{(79-35)}]}$ = 199.05°C, A= $\frac{Q}{UoassFt\Delta Tlm}$ Ft is 1 for single tube pass
To initiate the design an overall heat transfer coefficient of 22w/m^2K was assumed.

A= $\frac{7.79 \times 1.0063 \times 10^3(55-35)}{22 \times 199.05 \times 1}$ =35.8m^2, N = $\frac{A}{\pi dl}$ = $\frac{35.8}{\pi \times 0.0254 \times 4}$ = 112

A single tube pass heat exchanger, 1in OD and 1.25 in pitch, should have 112 tubes.

iv) Determination of heat transfers coefficients of tube side and shell side

Tube side fluid-air

Reynolds' number = $\frac{4 \text{ x } 7.79}{112 \times \pi \times 0.1962 \times 10^{-6} \times 0.021184}$ = 2.13 ×10^7

Re >2300, the flow is turbulent.

$hi = 0.023(Re)^{0.8}(\text{Pr})^{0.36}$=0.023(2.13 × 10^7)$^{0.8}$(0.7071$^{0.36}$=14757.2w/m^{2}k

Shell side fluid-exhaust

Reynolds' number = $\frac{\rho Umaxdi}{\mu}$

U_{max} =$\frac{Pt}{Pt-di} \times U$ and U= $\frac{4 \times 0.29}{0.5774\pi \times 0.1^2}$ = 63.94m/s hence U_{max} = $\frac{31.75}{31.75-21.184} \times 63.94$ =192.13m/s

Re = $\frac{0.5774 \times 192.13 \times 0.0021184}{3.05564 \times 10^{-5}}$ = 7.69 × 10^4 .since Re>1000, the flow is therefore turbulent.

$$ho = 0.021(Re)^{0.84}(Pr)^{0.36}(\frac{Pr}{Prw})^{0.25}$$

P_{rw} = Prandtl number at the wall

Q= $\dot{m}_a C_{pa}$ (Tw-T) = $\dot{m}_e C_{pe}$ (T-Tw) substituting gives 7.79 × 1.0063(Tw-45) = 0.29 × 1.0539(339.45-Tw)

8.1446Tw = 456.498 hence Tw = 56.05°C

Prandtl number of air at 56.05 degrees = 0.704

$ho = 0.021(7.69 \times 10^4)^{0.84}(0.68)^{0.36}(\frac{0.68}{0.704})^{0.25}$ = 230.57w/m^{2}k

$$Uocall = \frac{1}{\frac{25.4}{21.184} \times \frac{1}{14757.2} + \frac{25.4}{21.184}0.0072 + \frac{0.0127ln\,(25.4/21.184)}{30} + 0.029 + \frac{1}{230.57}}$$

Uocall = 23.74 w/m^{2}k. Thus, Checking for error in the overall heat transfer coefficient

% error = $\frac{Uocal-Uoass}{Uoass}$ × 100 = $\frac{23.74-22}{22}$ × 100 = 7.91%

7.91 % < 30% (within the min allowable error). The overall heat transfer coefficient is acceptable.

v) Determination of the heat transfer overdesign

% overdesign = $\frac{112-112}{112}$ × 100 = 0.0% since 0.0% < 10% - typical min area overdesign.

The above design is therefore within the acceptable criteria.

vi) Finding the shell diameter

According to the TEMA standards, for 112 tubes the recommended shell diameter is 17.25 in

17.25 × 25.4 = 438.15mm

vii) Baffle spacing and baffle cut

Recommended baffle spacing is between 0.4Ds-0.6Ds

Baffle spacing = 0.5Ds= 0.5 × 438.15= 219.075mm

Recommended baffle cut is 25%

Baffle cut = 0.25 × 438.15= 109.5375mm

viii) Finding the shell thickness

The working pressure is 200kPa, but the design pressure is taken as 10% higher than working pressure i.e., 1.1 × 200 =220kPa

Maximum allowable stress (σ) for steel is 100MPa.

t_s=$\frac{PR}{\sigma j-0.6P}$ + corrosion – allowance, where j is the efficiency of the joint usually 0.8.

Therefore t_s = (220 × 219.08)/ (100 × 10^3 × 0.8 – 0.6 × 220) + corrosion allowance = 0.6035 + corrosion allowance, and corrosion allowance for galvanised steel is 1.5mm. Hence i.e., t_s=0.6035 + 1.5 = 2.1035mm

For standard sheet thickness, the nearest galvanised sheet metal of normal bore of 20mm, ts=2.35mm

ix) Tube sheet thickness determination

The minimum tube sheet thickness to resist bending is given by.

$t_s = \frac{FDs}{3}\sqrt{\frac{P}{k\sigma}}$, where: F= 1 for fixed tube sheet and k= 1 - $\frac{0.785}{(\frac{pt}{do})^2}$, k = 1 - $\frac{0.785}{(\frac{31.75}{25.4})^2}$ = 0.4976

$t_s = \frac{1x438.15}{3}\sqrt{\frac{220}{0.4976\ x\ 100000}}$ = 9.7mm approx. 10mm

x) Pressure drop calculation

Tube side pressure drop- air.$\Delta P = f \times \frac{Lt\rho V^2}{D}$

Friction factor for flow with Re= 2.13×10^7 and relative surface roughness ε/D= $\frac{0.002}{21.184} = 9.47 \times 10^{-5}$ is 0.0085 reading from the Moody chart diagram.

Therefore, $\Delta P = 0.021\ x\ \frac{4 \times 1.086 \times 30.24^2}{0.021184}$ = 3.938kPa hence 3.938kPa < 14kPa

Tube side pressure is within the max allowable pressure drop.

Shell side pressure drop- exhaust.

$\Delta P = \frac{2fG^2N}{\rho} \times (\frac{\mu w}{\mu b})$ x 0.14, f= $\frac{0.044+0.008(\frac{Pt}{D})}{(\frac{Pt-D}{D})^{0.43+1.13\frac{D}{Pt}}} \times Re^{-0.15} = 0.1690$

$G_{max} = \rho U_{max}$ = 0.5774 x 192.13 = 2582.35, N = number of transverse rows = 8, μw= 1.962 x 10^{-5} at wall temperature of 56.05 degrees,μb = 3.05564 x 10^{-5} at bulk temperature

Therefore. $\Delta P = \frac{2 \times 0.1690 \times 110.94^2\ x\ 8}{0.5774} \times (\frac{1.962}{3.05564})$ x 0.14 =5.181kPa therefore 5.181kPa < 68.9kPa

Shell side pressure drop is within the maximum allowable pressure.

xi) Effectiveness of the exhaust heat exchanger

Effectiveness, $\varepsilon = \frac{rate\ of\ heat\ transfer\ in\ heat\ exchanger}{maximum\ possible\ heat\ transfer\ rate} = \frac{q}{qmax} = \frac{\dot{m}hCph(Thi-The)}{(Thi-Tci)}$

- Hot fluid – 0.29 × 1.0539 = 0.305631
- Cold fluid – 7.79 × 1.0063 = 7.839077

Hence if, $\dot{m}_c C_{pc} > \dot{m}_h C_{ph}$, then: $\varepsilon = \frac{Thi-The}{Thi-Tci} = \frac{599.9-79}{599.9-35} = 0.92$

By definition, ε is a dimensionless quantity and should be between 0 and 1.

4.1.2 Design of the cereal's dryer

The outdoor air temperature and relative humidity had to be known. The ambient air temperature was 25 degrees and relative humidity was 70%.

i) Determination of final weight of grain after drying.

$W_f = W_i\ (\frac{100 - Mci}{100 - Mcf})$, Where: W_f = final weight of maize after drying, Wi = initial weight of maize before drying = 40000 kg, M_{ci} = initial moisture content of maize before drying = 20%, M_{cf} = final moisture content of maize after drying = 13% Therefore $W_f = 40000\ (\frac{100-20}{100-13}) = 36781.61$ kg

ii) Determination of Moisture Content

$M_c = \frac{100(Wi - Wf)}{Wi} = \frac{100(40000-36781.61)}{40000}$ Mc = 8.046%

iii.) Determination of bulk density

$TW_m = 0.7019 + 0.01676\ (8.046) - 0.0011598\ (8.046)^2 + 0.00001824\ (8.046)^3$ = 0.7712 g/cm^3 or 771.2 kg/m^3

iv.) Amount of moisture to be removed.

M = mass per batch to be dried = 40000 kg, Q1 = initial moisture content in maize as a fraction = 0.20

Q2 = final moisture content in maize as a fraction = 0.13; MR = $40000(\frac{0.20-0.13}{1-0.13})$ = 3218.39 kg

v.) Quantity of air to effect drying.

Hr2 = 0.017 (final humidity ratio from psychometric chart at 15% relative humidity & 55ºC)

Hr1 = 0.014 (final humidity ratio from psychometric chart at 70% relative humidity & 25ºC)

Qa = $(\frac{3218.39}{0.017 - 0.014})$ = 1,072,796.67 kg

vi.) Volume of air to effect drying.

Va = $\frac{Qa}{\delta a}$ Where: δa = density of air in kg/m^3, ρ_a = 1.294 kg/m^3, Va = $\frac{1072796.67}{1.294}$ = 829,054.61 m^3

ii.) Quantity of heat required for effective drying (Hr) in kJ.

M = mass of maize to be dried per batch, HL = Latent heat of vaporization = 1248.1 kJ/kg

$H_K = C_T (T_2 - T_1)$, C_T = Specific heat of maize = 1.8 kJ/kg ºC, T_1 = Ambient temperature = 25 ºC

T_2 = Drying temperature = 55 ºC,H_K = 1.8 (55 - 25) = 54, Hr = (40000 × 54) + (1248.1 × 3218.39) = 6,176,872.56 kJ

v.) Actual heat used to effect drying.

$H_D = Ca\ Tc\ M_{R,\ Where}$: Ca = Specific heat capacity of air = 1.005 kJ / kg, Tc = Temperature difference, HD = 1.005 × 30 × 3218.39 = 97,034.46 kJ

vii.) Design of the drying chamber (dimensions)

From bulk density obtained, 771.2 kg/m^3, 1kg of maize therefore occupies $\frac{1}{771.2}$ = 0.001296680498 m^3.

Hence 40000kg occupies 40000 × 0.001296680498 = 51.87 m^3
Taking a height of 9 meters for the cereal dryer, the diameter is obtained as follows:
$V = \Pi r^2 h$ hence $51.87 = \Pi r^2 \times 9$, r = 1.355 m
Diameter = 2.71 m

Therefore, dryer dimensions are 2.71m diameter by 9m height.

4.2. Blower specifications

A blower powered by a 3-phase induction motor of 415V, 22kW, 44 Amps and a frequency of 50-60 Hz. The blower was to operate at a desired mass flow rate of 7.79 kg/s. The assumption made was using thermodynamic properties of dry air for the exhaust. We know that exhaust gas is a mixture of carbon monoxide, carbon dioxide, water vapour, nitrogen, and sulphur. The assumption was done because determining the exact properties of the exhaust gas would be a long and tedious process. Nitrogen is the largest compound of the exhaust as it is in air, using properties of air was the best approximation. The ideal designs are done when there are no heat losses to the atmosphere. The heat losses are there, and they reduce the efficiency of the system. Heat is lost through radiation and conduction. The loss can be avoided by glass or rubber insulation of the tubes. In simplifying the calculation, another assumption was made; head loss was neglected across pipe fittings. Pressure drops in the reducers, valves and elbows, pipe to blower connection had to be considered (Pekmez, 2017; Rivier et al., 2018).

5.0.SUMMARY AND CONCLUSION

The design consists of two heat exchangers and a drier. These are the exhaust heat recovery heat exchanger, the cooling waste heat recovery heat exchanger which supply heat to the drier for cereals drying. He main parameters for the heat exchanger specifications are number of passes, length of tubes, the inner and outer diameter of tubes, baffle spacing and number of tubes. These are summarized in table 4 below.

Table 4: Coolant heat exchanger specifications (Author's analysis).

Parameter	Specifications
Number of passes	1
Length of heat exchanger tubes	4.2m
Shell diameter	438.15mm
Tube inner diameter	21.184mm
Tube outer diameter	25.4mm
Baffle spacing	219.075mm
Number of tubes	112

Table 4 above summarizes the specifications for the coolant heat exchanger. It has 112 tubes of length 4.2 m and tube outer diameter of 25.4 mm with tube inner diameter of 21.184 m. table 6 below summarizes the specifications for the exhaust heat exchanger.

Table 5: Exhaust heat exchanger specifications. (Author's summary)

Parameter	Specifications
Number of passes	1
Length of heat exchanger tubes	4.0m
Shell diameter	438.15mm
Tube inner diameter	21.184mm
Tube outer diameter	25.4mm
Baffle spacing	219.075mm
Number of tubes	112

Table 5 gives a summary of the exhaust heat exchanger specifications. It is a shell-tube type of heat exchanger with 112 tubes.

The main parameters of the drier are the height, diameter, volume, and discharge time which are summarised below. The drier specification is summarized in table 6 below.

Table 6: Cereal's dryer specifications. (Author's summary)

Height, meters	9
Diameter, meters	2.71
Mass of grains (maize), tones	40

Volume, cubic meters	51.87
Discharge time, hour(s)	1

From table 6, the drier overall height is 9 m, diameter is 2.71 m, the design capacity 40,000 kg of maize and volume 51.87 m^3.

The design temperatures are 599.9 degrees Celsius for exhaust and 100 degrees Celsius for coolant. Hot air was sourced from the two heat exchangers and found to be 55 degrees Celsius as the recommended temperature for drying. For the heat exchanger, each was found to have an effectiveness of 92% and 93% for energy from the exhaust and coolant respectively; the design engine load range should be set at around 70% - 90%. With this load, the temperature of the exhaust manifold and the jacket water will be high enough to get heat from. Fouling in the coolant, water, heat exchanger due to scale formation and soot accumulation in the exhaust heat exchanger can cause insulating effects. It is therefore importance to clean the tubes of the heat exchangers for proper heat transfer between the fluids. Heat exchangers can be cleaned regularly using industrial chemical agents. The cereals dryer has designed capacity 40,000 kilogrammes of grains in 1hr (in a single operation). However, the discharge time is set to vary but not too much depending on the moisture content if it is above 20%. This study demonstrates that waste engine heat from both the exhaust and cooling systems can be extracted and used to dry cereal while the engine runs the drier. This will reduce the load on conventional fuel normally natural gas. This reduces cost ad contributes to reduction in use of fossil fuels in drying operations for mobile driers.

References

Bala, B. K. (2017). *Drying and Storage of Cereal Grains*(2 ed., pp. 340). doi:https://doi.org/10.1002/9781119124207.ch1

Barasa, M. J. K. (2018). Transformational Vs Transactional Leadership with Examples. *International Journal of Business & Management, 6*(5), 191-193. . Retrieved from http://www.internationaljournalcorner.com/index.php/theijbm/article/view/129786/90079

Bari, S., & Hossain, S. (2013). Waste heat recovery from a diesel engine using shell and tube heat exchanger. *Applied Thermal Engineering, , 61*(2), 355-363. doi:https://doi.org/10.1016/j.applthermaleng.2013.08.020

Bianchi. (2015). Exhaust Waste Heat Recovery in Exhaust Systems. In.

FAO. (1983). Food loss prevention in perishable crops *FAO Agriculture Services Bulletin, 43*, 89. Retrieved from http://www.fao.org/3/s8620e/S8620E06.htm

FAO. (1994). *Grain Storage Techniques*. Rome.

Jack, T. K., & Ojapah, M. M. (2013). Water-Cooled petrol engines:: A review of considerations in cooling systems calculations with variable coolant density snd specific heats. *International Journal of Advances in Engineering & Technology, 6*(2), 659-667.

Jeremiahi, M. B. K. (2019). Evolution of Project Management, Monitoring and Evaluation, with Historical Events and Projects that Have Shaped the Development of Project Management as a Profession. *International Journal of Science and Research (IJSR), 8* (12). doi:10.21275/ART20202078

Kabeyi, M. J. B. (2019). Geothermal electricity generation, challenges, opportunities and recommendations. *International Journal of Advances in Scientific Research and Engineering (ijasre), 5*(8), 53-95. doi:https://di.org/10.31695/IJASRE.2019.33408

Kabeyi, M. J. B. (2020a). Feasibility of Wellhead Technology Power Plants for Electricity Generation. *International Journal of Computer Engineering in Research Trends, 7*(2), 1-16. doi:https://doi.org/10.22362/ijcert/2020/v7/i02/v7i0201

Kabeyi, M. J. B. (2020b). Investigating the challenges of bagasse cogeneration in the kenyan Sugar Industry. *International Journal of Engineering Sciences & Research Technology, 9*(5), 7-64. doi:https://doi.org/10.5281/zenodo.3828855

Kabeyi, M. J. B., & Oludolapo, A. O. (2020a). *Design and Modelling of a Waste Heat Recovery System for a 250KW Diesel Engine for Cereal Drying*. Paper presented at the 2nd African International Conference on Industrial Engineering and Operations Management, Harare, Zimbabwe. http://www.ieomsociety.org/harare2020/papers/78.pdf

Kabeyi, M. J. B., & Oludolapo, A. O. (2020b). *Performance Analysis of an Open Cycle Gas Turbine Power Plant in Grid Electricity Generation*. Paper presented at the 2020 IEEE International Conference on Industrial Engineering and Engineering Management (IEEM), Singapore, Singapore. https://ieeexplore.ieee.org/stamp/stamp.jsp?tp=&arnumber=9309840

Kabeyi, M. J. B., & Oludolapo, A. O. (2020c, 5th – 7th October 2020). *Performance analysis of diesel engine power plants for grid electricity supply.* Paper presented at the 31ST Annual Southern African Institution for Industrial Engineering Conference, South Africa.

Kabeyi, M. J. B., & Oludolapo, A. O. (2021). Central versus wellhead power plants in geothermal grid electricity generation. *Energy, Sustainability and Society 11*(7), 23. doi:ttps://doi.org/10.1186/s13705-021-00283-8

Mitzlaff, K. (1988). *Engines for biogas*. (3-528-02032-6). Eschborn, Germany: Vieweg & Sohn Verlagsgesellschaft mbH Retrieved from https://biogas.ifas.ufl.edu/ad_development/documents/Engines%20for%20biogas.pdf

Muthukumarappan, S. (2016). *Grain Drying Systems.* (2 ed.).

Oludolapo, A. O., & Kabeyi, M. J. B. (2020). *Design and Modelling of a Waste Heat Recovery System for a 250KW Diesel Engine for Cereals Drying*. Paper presented at the 2nd African International Conference on Industrial Engineering and Operations Management Harare, Zimbabwe. http://www.ieomsociety.org/harare2020/papers/78.pdf

Pekmez, H. (2017). Cereal Storage Techniques: A Review. *Journal of Agricultural Science and Technology, 6* (2016), 67-71. doi:https://doi.org/10.17265/2161-6264/2016.02.001

Rettig, L., Lamare, Li, Mahadea, McCallion, Chernushevich. (2011). *Applicaton of Organic Rankine Cycles (ORC).* Paper presented at the World Engineers Convention, Geneva.

Rettig, L. L., Lagler, M., Lamare, T., Li, S., Mahadea, V., McCallion, S., & Chernushevich, J. (2011). *Application of Organic Rankine Cycles (ORC)*. Paper presented at the World Engineers Convention, Geneva. https://www.zhaw.ch/storage/engineering/institute-zentren/iefe/PDFs/orc-final-paper-wec2011-2011-07-30.pdf

Reykdal, Ó. (2018). *Drying and storing of harvested grain A Review of Methods*(pp. 20). Retrieved from https://www.matis.is/media/matis/utgafa/05-18-Drying-and-storage-of-grain.pdf

Rivier, M., Collignan, A., Meot, J. M., Madoumier, M., & Sebastian, P. (2018). Modeling a process combining a cereal dryer with a bioenergy unit for equipment design in developing countries. *Journal of Food Process Engineering, 41*(41:e12836.), 1-12. doi:https://doi.org/10.1111/jfpe.12836

Shareweb. (2019). *Drying the grain Its importance and common practices*(pp. 10). Retrieved from https://www.shareweb.ch/site/Agriculture-and-Food-Security/focusareas/Documents/phm_postcosecha_drying_grain_e.pdf

Tamil Nadu agricultural University. (2017). Drying- Grain drying , types, of drying , types of dryers and importance of drying. In (pp. 10).

Wilson, J. M. R., Singh, A. K., Singh, A. K., & Subramanian, L. R. J. (2017). Waste Heat Recovery from Diesel Engine. *Thermal Science, 21*(1B), 715-727. doi:10.2298/TSCI160426264W

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