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Development of a drainage and waste clearing machine

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ABSTRACT

Solid waste remains a critical environmental challenge globally, necessitating innovative solutions to alleviate its adverse impacts. This study aimed at enhancing solid waste management systems and mitigating the improper disposal of waste, particularly in drainage lines and the environment. The proposed solution involves the development of a waste and drainage clearing machine, designed to efficiently remove solid waste from drainage systems and roadside areas. Design analysis of the physical framework which included the buckling load analysis and structural integrity of the column, static load analysis of major components of the system, cantilever beam analysis of the looms, structural stability of the system was carried out. Results showed that the total weight of the machine is 384.75 kg and capable of clearing solid wastes from drainages at the rate of 300 kg per hour at 5 kg per operation, and can handle a maximum load of 10 kg in a single operation per minute. The portability of the machine also makes it suitable for narrow roads without undue obstruction to traffic. The innovation of this study is a potential source of employment and cost-effective service delivery with an overall benefit of guaranteeing a clean and healthy environment.

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1. INTRODUCTION

Solid waste management remains a significant challenge in urban areas worldwide, especially in rapidly growing cities of developing countries [1]. The combination of high population growth and increased per capita income has led to a substantial increase in solid waste generation, posing serious threats to both environmental quality and human health. It is estimated that more than 3.4 billion tons of municipal solid waste will be produced globally annually, with around 2.6 billion tons of greenhouse gas (GHG) emissions and consequently contributing immensely to global warming [2], [3]. Kaseva et.al. [4], delineate waste as encompassing all types of refuse originating from human and animal activities, commonly known as rubbish, trash, garbage, or junk. This includes unwanted solid materials generated from residential, industrial, and commercial sources within a specific area. Effective waste collection is a crucial aspect of solid waste management, ensuring regular removal from the source to prevent disposal on streets, drains, and the surroundings. Waste can be categorized as solid, liquid, or gaseous, each requiring different disposal and management methods in ensuring a clean and healthy environment [5].
According to Tchobanoglous et.al, [6] solid waste management involves the supervision and handling, collection, transportation, treatment, and disposal of waste in a manner that protects the environment and public health. Various methods, from ground-level dumping to containerization and direct loading onto waste trucks, are used for waste collection. Waste disposal aims to mitigate the adverse impacts of waste on human health and ensure

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environmental safety [7]. The whole process of collecting, transferring, treating, recycling, recovering resources, and disposing of solid waste in metropolitan areas is referred to as the Municipal Solid Waste Management systems [8]. It is, therefore, necessary for wastes to be managed properly and disposed of in a most acceptable manner to reduce its resultant effect on humans, plants, animals, and the environment in general.

In several rural and metropolitan areas of developing countries, governments face a major challenge in the management of solid waste [9]. Studies by the Federal Environmental Protection Agency, FEPA [10] reveals ongoing challenges with poor waste management in specific cities in Nigeria, including those designated as major tourist destinations. Some cities, such as Ibadan, Lagos, Onitsha, and Aba, have been identified and characterized with widespread littering of solid waste on roads, drainage lines, and unoccupied areas. Various disposal methods, such as dumping sites, incineration, recycling, and composting, have been employed for years. However, improper waste disposal practices, such as dumping waste in drainage lines, roads and unauthorized areas, are common in many developing countries [11]. Improper waste disposal leads to the generation of leachates which are contaminated liquids that are generated from water percolating through solid waste disposal sites and moving into subsurface areas during rainfall and pollute water bodies and contribute to environmental degradation [12], [13]. Improper disposal of solid wastes could lead to flooding and erosion of natural and occupied landscapes. It may cause hazardous substances and garbage from landfills, contaminated sites, tank farms, and other sites of environmental concern to be released into the ocean and rivers, jeopardizing water bodies, fish and wildlife [14].

The challenges of solid waste management in most developing and underdeveloped countries have increased in recent years due to the increase in population, industrialization, urbanization, and globalization [15]. One of such challenges has to do with building of sustainable cities which requires a substantial effort to reduce the production of municipal solid waste and improve its collection, transport and treatment systems [16]. As more cities become industrialized, the congenital problem of solid waste comes along with it. Technological and economic advancement has made the types and kinds of solid waste very diverse and their management much more complex. The complex nature of disease outbreaks such as cholera and diarrheal in recent times corroborate this fact. Furthermore, the changing economic trends and rapid urbanization complicate solid waste management in developing countries. Consequently, solid waste is not only increasing in composition but also changing in quantity from a few kilograms to tonnage proportions in recent times [17].

The most common arrangement in the few urban communities where a system is in place is for waste management authorities to collect refuse from households and public containers on a regular basis using collection trucks [18]. Unfortunately, operations managed by the waste management authorities have mostly been inefficient and ineffective as evidenced by mounds of decomposing refuse that have become a regular site in many urban areas [19]. Karande et.al. [20], developed an Automatic Garbage Collector Machine which helps to reduce human effort in garbage clearing on drainage lines utilizing an automated system. Based on the project design, the machine is placed on the drain so that solid wastes which float on the surface of the water bodies get lifted by the teeth connected to the chain and stored in a storage or collection bin. The system is powered by hydraulic turbines which generate electrical energy (electricity) for the running of the motor attached to the chain. The limitations of such a concept is that it would require highly skilled personnel to operate the machine. In addition, the machine contributed to environmental pollution as a result of generated sound and vibration.

Due to numerous problems associated with waste disposal, it is evident and pertinent that affordable and easy to operate waste disposal and drainage clearing machine is inevitable to facilitate proper and easier disposal of waste. A preliminary survey shows that available machines for clearing waste are usually very large and too expensive to purchase and maintain. Hence, the need for developing waste disposal and drainage clearing machine to enhance a sustainable environment at cheaper operating costs.

The primary objective of this study is to develop a drainage and waste disposal clearing machine that can be hauled with a conventional low-duty vehicle such as a tricycle. Major parts were produced from affordable and available sourced materials. The machine was designed to take-up solid waste from drainages and deposit them in designated containers for further disposal.

2. MATERIALS AND METHOD

2.1 Description and operation of the drainage clearing machine

The isometric view of the drainage clearing machine is shown in Fig. 1, and Table 1 presents the part list of the drainage clearing machine.

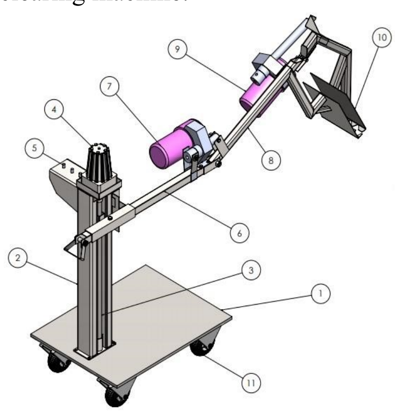

Fig. 1 *Isometric view of the drainage clearing machine.*

Table 1 - *Part list of the drainage clearing machine.*

S/No.	Parts	Quantity
1.	Base	1
2.	Column	1
3.	Threaded shaft	1
4.	Electric motor	1
5.	Control switch	3
6.	First loom	1
7.	Loom actuator	1
8.	Second loom	1
9.	Refuse parker actuator	1
10.	Refuse parker	1
11.	Wheel	4

The machine operates on the principle of electromechanical systems. The machine consist of the following main components; the column, base, looms, electric motor, linear actuators and control switches. The electric motor generates enough power to overcome external loads during operation. The machine was designed to handle about 5 kg solid waste in a single operation, and up to a maximum of 10 kg. The 1 hp electric motor acts as the primary source of mechanical power, transforming electrical energy into rotational motion, which is pivotal for propelling the vertical movements of other components of the machine via a threaded shaft attached to the column at a gear ratio of 3:1. The loom actuator harnesses its rotational force and converts it into a linear motion, enabling the system to maneuver a platform or container along an angular trajectory for efficient waste collection. Simultaneously, the opening and closing of the refuse packer was facilitated by the refuse parker actuator. The refuse parker can open to a maximum range of up to 135^o. The developed machine was test-run with a generating set.

2.2 Design analysis of the physical framework

2.2.1 Buckling load analysis of the column

The column shown in Fig. 1 is the major part of the machine on which the load of all other elements is hinged on. The column was designed for its load carrying capacity and structural stability according to the principles of buckling of columns under the support conditions of a fixed and free column [21]. The Euler's critical load and Euler's stress when buckling occurs are expressed by Eqs. (1) and (2) respectively.

$$P_{cr} = \frac{\pi^2 EI}{(KL)^2 F_S} \tag{1}$$

$$\sigma_{cr} = \frac{\pi^2 E}{(KL/r_G)^2 F_S} \tag{2}$$

where $E =$ Young's modulus of elasticity, $I =$ moment of inertia, $K =$ the effective length factor = 2 for a fixed and free column, $L =$ length of column $= 940\ mm$, $F_s =$ factor of safety, and $r_G =$ radius of gyration. The moment of inertia of a plane rectangular cross-section about the centroid: horizontal XX and vertical YY axes can be determined by:

$$I_{XX} = \frac{bh^3}{12} \tag{3}$$

$$I_{YY} = \frac{hb^3}{12} \tag{4}$$

where, b is the width of the cross section and h is the depth of the cross section. The cross section of the column is of hollow square section with external dimension$150\ mm \times 150\ mm$, and thickness $5\ mm$. For a hollow square section with thickness t, and $h = b$, the moment of inertia can be determined by:

$$I_{XX} = I_{YY} = [b^4 - (b - 2t)^4]/12 \tag{5}$$

Therefore the cross sectional area and moment of inertia of the column's cross section were obtained as area $A = 2900\ mm^2$, $I_{XX} = I_{YY} = 10.174 \times 10^6\ mm^4$. This indicates that the column has equal tendency to fail under buckling load on either of the axes of the cross section. The radius of gyration was determined by:

$$r_G = \sqrt{\frac{I}{A}} \tag{6}$$

the radius of gyration r_G was obtained as $59.23\ mm$, and for carbon steel, $E = 200\ GPa$. Typical factor of safety for carbon steel under steady load = 4 [22]. Consequently, the Euler's critical load and Euler's stress were determined from Eqs. (1) and (2) respectively as $P_{cr} = 1.42\ MN$ and $\sigma_{cr} = 490\ MPa$.

2.2.2 Structural integrity of the column

The ability of the column to perform optimally under load was analysed as follows [21]:

If $\frac{KL}{r_G} > 200$, then the column is too slender for safe use (7)

If $\frac{KL}{r_G} > 4.71\sqrt{\frac{E}{\sigma_y}}$, then $P_{all} = 0.525\sigma_{cr}A$ (8)

If $\frac{KL}{r_G} < 4.71\sqrt{\frac{E}{\sigma_y}}$, then $P_{all} = \frac{0.658^{(\sigma_y/\sigma_{cr})}\sigma_y A}{1.67}$ (9)

where, P_{all} = allowable load and σ_y = yield strength. The material properties of square hollow section (SHS) steel was determined by Keykha et. al. [23] as Modulus of Elasticity $E = 200\ GPa$, yield strength $\sigma_y = 280\ MPa$ and ultimate strength $\sigma_u = 340\ MPa$. Consequently, by substituting the values of K, L and r_G in Eqs. (7)-(9), determined values are; $KL/r_G = 31.74$, while $4.71\sqrt{E/\sigma_y} = 125.88$, Since $KL/r_G < 4.71\sqrt{E/\sigma_y}$, Eq. (9) together with previously determined values of σ_{cr} and area, A were therefore used to determine the allowable load for the column, and was obtained as $P_{all} = 0.383\ MN$. The structural integrity is assured since $P_{all}\ (0.383\ MN) < P_{cr}\ (1.42\ MN)$.

2.2.3 Static load analysis of major components of the system

The dead weight or mass of various members of the system were estimated from the geometric volume of each member and density of carbon steel ρ_s = 7850 kg/m^3, and known masses of other unit accessories that make up the entire machine.
The static load analysis of major components of the system is presented in Table 2. This analysis provides the necessary input data for further structural evaluation and optimization of the system under operational conditions.

Table 2 - *Static load analysis of major components.*

S/No.	Name of part	Geometrical dimensions (mm)	Volume, V (m^3)	Mass $(m = \rho_s \times V)$ (kg)	Weight $(W = m \times 9.81)$ (N)
1.	Base	$800 \times 600 \times 5$	2.4×10^{-3}	18.84	184.82
2.	Column (square hollow)	External $150 \times 150 \times 940$ Internal $145 \times 145 \times 940$	1.39×10^{-3}	10.91	107.03
3.	Electric motor			2.6	25.51
4.	First loom (square hollow)	External $40 \times 40 \times 667$ Internal $38 \times 38 \times 667$	104.1×10^{-6}	0.82	8.04
5.	Loom actuator			2.27	22.27
6.	Second loom (square hollow)	External $40 \times 40 \times 483$ Internal $38 \times 38 \times 483$	75.35×10^{-6}	0.59	5.79
7.	Refuse parker actuator			2.27	22.27
8.	Refuse parker (made up of 7 different parts joined together to form a single unit)				
	Part 1	$40 \times 350 \times 2$	28×10^{-3}	0.2198	2.156
	Part 2	$40 \times 172 \times 2$	13.76×10^{-6}	0.1080	1.060
	Part 3	$40 \times 180 \times 2$	14.4×10^{-6}	0.1130	1.109
	Part 4	$40 \times 175 \times 2$	14×10^{-6}	0.1099	1.078
	Part 5	$40 \times 60 \times 2$	4.8×10^{-6}	0.0380	0.370
	Part 6	$40 \times 65 \times 2$	5.2×10^{-6}	0.0410	0.400
	Part 7 (2 in number)	$2 \times (148 \times 125 \times 1)$	37×10^{-6}	0.2910	2.850
	Total for refuse parker			0.9207	9.023

2.2.4 Cantilever beam analysis of the looms

The loading condition on the first loom having external dimensions of 40×40×667, and internal dimensions of 38×38×667 is the most critical since it has a longer length and same cross sectional area in comparison with the second loom of length, 483 mm. Furthermore, the first loom experiences more load than the second loom. The free body diagram of the first loom as a cantilever is hereby shown in Fig. 2.

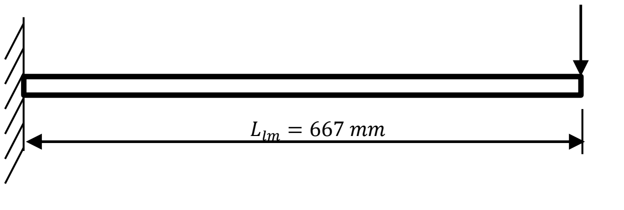

Fig. 2 *Free body diagram of the first loom as a cantilever.*

The most severe possible condition is to assume all the load extending from the column acts at the end of the loom. This load is the sum of the loads of parts serial number (4) – (8), 67.393 N in Table 2, and the maximum design load of the machine of 98.1 N, which totalled 165.5 N.
The maximum deflection of the loom can be obtained by:

$$y_{\max(lm)} = \frac{WL^3}{3EI} \quad (10)$$

Generally, the deflection limit for cantilever beams set by most design codes is $L/180$ for live load and $L/90$ for combined dead and live load [24]. The deflection of the beam is therefore 3.71 mm and 7.41 mm if subjected to either live load or combined dead and live load respectively. This indicates that the maximum acceptable deflection for the analysed loom is 7.41 mm. The moment of inertia for the loom can therefore be obtained from Eq. (10) as:

$$I = \frac{WL^3}{3Ey_{\max(lm)}} \quad (11)$$

By substituting the value for the deflection and other parameters in Eq. 11, the moment of inertia for the loom was determined as $I = 11.045 \times 10^3\ mm$. For the hollow square section with an outer width of $b = 40\ mm$, the minimum allowable thickness can be obtained by rearranging Eq. (5) as:

$$t = \left[b - (b^4 - 12 \times I)^{1/4}\right]/2 \qquad (12)$$

Therefore, by substituting the values of I and b in Eq. (12), the acceptable thickness of the loom was determined as $t = 0.53\ mm$. Consequently, the used thickness of $t = 2\ mm$ with the width of $b = 40\ mm$ affirms the structural integrity of the loom.

2.2.5 Structural stability of the system

The combined dead weight of the base and the column as presented in Table 2 are respectively $184.82\ N$ and $107.03\ N$, which amounts to $291.85.\ N$. This load is greater than the external load of $165.5\ N$ exerted on the base-column assembly. Consequently, the structural instability that may be caused by the external load to the base-column assembly is restrained, hence, the stability of the entire system during operation is enhanced. Furthermore, the structural stability of the system can be analysed by considering the external loads and reactive forces on the base; while the base itself is analysed as a structural unit subjected to its own weight considered as a uniformly distributed load.

The free body diagram of the base is shown in Fig. 3. Where, W_{CL} is the weight of the column, ω_b is the uniformly distributed load per unit area, consequent of the weight of the base, M_{CB} is the moment exerted at the intersection of the base and the column, L_b is the length of the base, t_b is the thickness of the base, while R_1 and R_2 represents the paired-reactive forces at the end of its length. The moment $M_{CB} = 110.39\ Nm$ was determined from the free body diagram of the first loom as a cantilever as shown in Fig. 2.

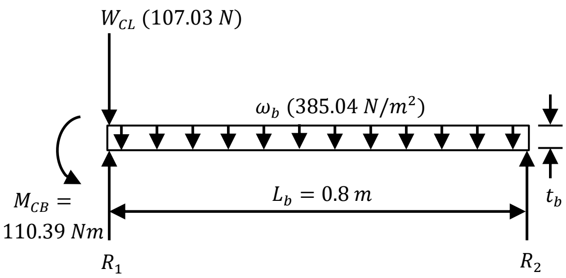

Fig. 3 *Free body diagram of the base.*

Applying the equations of static equilibrium to Fig. 3, the reactive forces were determined as $R_1 = 199.44\ N$ and $R_2 = 92.41\ N$ respectively.

The deflection analysis of the base shown in Fig. 3 results in equations for the slope and deflection, given as Eq. (13). These expressions are obtained based on classical beam theory and serve to evaluate the structural integrity and stiffness of the base under expected loading conditions.

$$EI\frac{dy}{dx} = 46.205x^2 - 110.39x + 34.3 \qquad (13)$$

$$EIy = 15.402x^3 - 55.2x^2 + 34.3x \qquad (14)$$

The solution to the deflection curve with applicable boundary conditions indicates that the maximum deflection will occur at $0.367\ m$ from the left support, and the slope ≈ 0 at this point. Hence, equation (14) reduces to

$$EIy_{max} = 5.915 \qquad (15)$$

The maximum allowable deflection of a beam is generally given as $1/360$ of the length [25]. However, for a detailed specification such as; a vertical deflection, live load, simple span of an elastic member, the maximum deflection could be limited to $1/240$ of the length [26]. A maximum deflection of $y_{max} = L_b/240$ was used for the design of the base. The moment of inertia of the base structure can be expressed from Eq. (15) as:

$$I = (5.915 \times 240)/(E \times L_b) \qquad (16)$$

Consequently, for Modulus of Elasticity $E = 200\ GPa$ for a steel material, and a span length of $0.8\ m$, the moment of inertia for the base was determined as $I = 8.87 \times 10^{-9}\ m^4$ or $8.87 \times 10^3\ mm^4$. From the moment of inertia of a plane rectangular cross-section about the centroid, horizontal XX axis expressed by Eq. (3) where, the width of the base is $b = 600\ mm$ (Table 2), the minimum thickness of the base can therefore be expressed from Eq. (3) as:

$$t_b = h = \left(\frac{12I}{b}\right)^{1/3} \qquad (17)$$

The acceptable thickness of the base was therefore determined as $t_b = 5.62\ mm \approx 6.0\ mm$.

2.3 Selection of electric motor and actuators

The functionality of the physical framework of the machine was made possible by the use of an electric motor and two electric linear actuators operating on alternate current (AC) as shown in Fig. 1. The electric motor was responsible for the vertical motion of the first loom (which is horizontal) through the aid of a threaded shaft attached to the column and a control switch. The first loom bears all other components of the machine and external load. The first actuator known as the loom actuator facilitates angular displacements of the second loom in relation to the horizontal first loom, thereby allowing for pick-up and raising of solid waste off the ground. The second actuator also known as the refuse parker actuator causes the opening and closing of the jaws of the refuse parker. Both the electric linear actuators operates by converting the rotary motion of an electric motor into linear motion, hence facilitating forward and backward motions by means of control switches. The linear motions are converted into angular displacement motions through incorporated linkages. Available sourced one horse power (745 W) electric motor, 75 W loom actuator and 93.2 W refuse actuator were utilized for the operation of the machine. The specifications of electric motor and electric linear actuators are presented in Table 3.

Table 3 - *Specifications of electric motor and electric linear actuators.*

Electric motor	Loom Actuator	Refuse actuator
Model no: Gearhead-3GN180K	Model no: JS18-B N40048L-TN23-003	Model no: JS18-B N30048S-TM33-019
Phase: single	Phase: single	Phase: single
Power: 1hp	Power: 1/10 hp	Power: 1/8 hp
Voltage: 220V	Voltage: 220 V	Voltage: 220 V
Angular velocity: 500/600rpm	Angular velocity: 70/90 rpm	Angular velocity: 70/90 rpm
Frequency: 50/60Hz	Frequency: 50/60 Hz	Frequency: 50/60 Hz
	Gear ratio: 20:1	Gear ratio: 20:1

3. RESULTS AND DISCUSSION

3.1 Mechanical Framework and Structural Integrity of the Machine

Detailed design analysis of a developed drainage and waste disposal clearing machine was carried out for an operating load of 5 kg and a maximum load of 10 kg. An electric motor and two electric linear actuators were used to test-run and operate the machine. The buckling load analysis of the column, that is, the Euler's critical load and Euler's stress were obtained as $P_{cr} = 1.42\ MN$ and $\sigma_{cr} = 490\ MPa$ respectively. Determined results showed a reliable structural integrity of the assembly, since the allowable designed load is less than the Euler's critical load for the column: $P_{all}\ (0.383\ MN) < P_{cr}\ (1.42\ MN)$.

The static load analysis of components of the machine structure indicated various weights as follows; base = 184.82 N, column = 107.03 N, electric motor = 25.51 N, first loom (attached to the column) = 8.04 N, loom actuator = 22.27 N, second loom (connecting the first loom and the parker) = 5.79 N, refuse parker actuator = 22.27 N, and refuse parker = 9.023 N. The square geometry of the looms was satisfactorily designed as: thickness, $t = 2\ mm$ and width, $b = 40\ mm$ respectively. And within design limits, the thickness of the base was determined as $t_b = 6.0\ mm$. Although the $5.0\ mm$ thickness used to fabricate the base of the machine due to availability of material is reliable, since other design criteria were not exceeded.

3.2 Performance of the Machine

The waste disposal and drainage clearing machine was designed using analytical design principles and Solid-works CAD software to develop the geometry. The machine was fabricated and tested, and found to work to expectation by carrying solid waste which weighed 5 kg from one location to another satisfactorily. The movement of the machine parts were controlled by three control switches, which worked for the movement of the horizontal loom on the threaded shaft attached to the column and controlled by an electric motor. The angular displacement of the second loom, and the opening and closing of the jaws of the refuse parker were respectively controlled by individual electric linear actuators. All the various movements of the parts can be done simultaneously without any interference from any part of the machine. The machine is easy to operate and less expensive compared to heavy duty machines used for parking solid wastes. The portability of the machine also makes it suitable for narrow roads without undue obstruction to traffic. Furthermore, the machine can easily be hauled along the sides of the road or drainage with a light duty vehicle such as a tricycle, thereby requiring minimal energy for overall operation. At an operating rate of 1 minute to carry solid waste of 5 kg from one location to a disposal container, the developed machine is capable to clear drainages at the rate of 300 kg in 1 hour and 1 ton of solid waste within 3 hours and 20 minutes.

4. CONCLUSION

The solid waste disposal and drainage clearing machine developed in this study is a portable machine weighing a total of 384.75 kg as indicated by the static load analysis of major components presented in Table 2. The machine can easily be hauled during operation via castor wheels attached at the 4 corners of its base. The designed machine can successfully handle a maximum load of 10 kg (about $100\ N$), since designed parts such as the column can withstand an allowable load of $P_{all} = 0.383\ MN$ which is less than the Euler's critical load of $P_{cr} = 1.42\ MN$. The flexibility of the operation of the machine is such that the three major movable parts namely the horizontal first loom, the second loom and the parker can operate independently with their respective switches, electric motor and linear actuators. The machine which was designed to pick-up and transfer solid waste from a site and deposit into a disposal container can operate and clear solid wastes from drainages at the rate of 300 kg per hour which amounts to about 1 ton of solid waste in a duration of about 3 hours and 20 minutes. The design herein provided can be up scaled for higher capacity and performance. The developed machine which was test-run using a generating set could be designed and adapted to use a DC power source of a vehicle. This innovation is a potential source of employment and service delivery at a cost effective rate and also to ensuring a clean and healthy environment.

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