

Mechanical Analysis of Locally Developed Hand-Pushed Vibratory Soil Compactor for Usage Acceptability

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Abstract. The developed machine was designed by the authors to have a chassis mass of 32.68 kg, water reservoir of $6.57 \times 10^{-3} \text{ m}^3$, the discharge rate for soil wetting is $3.49 \times 10^{-3} \text{ m}^3 \text{ s}^{-1}$, and compacting rate of the vibrator is 5000 vpm with a power of 4.08 KW (5.5HP). The machine chassis was analyzed using solid works tools. The actual chassis model information is: mass 30.78 kg, volume 0.003921 m^3 , density of 7850 kg/m^3 and weight is 301.647 N. The material properties are: Name (1023 Carbon Steel sheet), Yield strength ($2.82685 \times 10^8 \text{ N/m}^2$), Tensile strength ($4.25 \times 10^8 \text{ N/m}^2$), Elastic modulus ($2.05 \times 10^{11} \text{ N/m}^2$), Poisson's ratio (0.29), Mass density (7850 kg/m^3), Shear modulus ($8 \times 10^{10} \text{ N/m}^2$) and Thermal expansion coefficient ($1.2 \times 10^{-5} \text{ Kelvin}$). The maximum allowable stress and strain that the vibrator seat could be subjected to are 958,428,688 N/m^2 and 2.983×10^6 respectively. Stress and strain above these will cause the failure of the seat.

Key words: design, soil compactor, mechanical analysis, local content, usage acceptability.

Introduction

Soil improvement by means of compaction is used increasingly for the solution of different types of foundation problems in coarse-grained soil deposits, in particular where the foundations will be subjected to dynamic and cyclic loading (Massarsch and Fellenius, 2002). In soil mechanics literature several terms like compaction, consolidation, etc., are used to describe the change in soil volume and increase in density. Compaction of soil refers to the increase in density of a soil by a dynamic load. Consolidation is the gradual decrease in volume of voids with increase in density under the action of continuously acting static load over a period of time (Ghildyal and Tripathi, 1987: 125-129).

Soil compaction is a vital part of the construction process. It is used for support of structural entities such as building foundations, roadways, walkways, and earth retaining structures to name a few. For a given soil type certain properties may deem it more or less desirable to perform adequately for a particular circumstance (David and McCarthy, 2006: 595). Properly compacted soil is a key component of virtually every construction project. It provides the firm, dense base needed to support footings and foundations, slabs, and pavements. When the underlying soil is not solid or dense enough, excessive soil settlement can lead to a variety of structural problems such as basement walls that crack and leak, pipes that leak and break, slab cracks, and foundation erosion (Hooker, 2012).

The hand-pushed vibratory soil compacting machine is a device used to compact soil, broken stones (non-cohesive materials), concrete or asphalt coatings, as well as residuum at the ecological ramps. The compactor uses a mechanism (driven by petrol or a diesel engine) which creates a descending force added to the equipment's static weight. One or two eccentrically weights turning around usually form the vibrating generator. The resulted vibrations generate the equipment's advancing movement (Morariu-Gligor, 2014: 189).

Building failure is a common phenomenon in our society today. One of the fundamental problems that associated to this is weak foundation. One of the major factors responsible for weak foundation is the looseness of the soil that the foundation is made of. This is as a result of the presence of holes, air, seepages in the soil. This problem could be eliminated by subjecting the soil to an effective compaction using appropriate soil compacting machine. Soil stands as the main component of the foundation. Once the foundation of a building or any footing construction work is well-designed, structured and built, the building on it is secured. The major problems inherent in the existing soil compaction machine are highlighted below.

Most of the soil compacting machines use in the preparation of the foundation of building and footing construction works are not having an integrated means of water supply (water reservoir). Water is essential in the compacting of soil, most especially mixture of granular and cohesive type of soil. It aids the excitation caused by vibration from the soil compacting machine which subsequently cause the soil particles to be bonded together. This water has to be supplied at regular rate and with right volume. Alternative source and application of water has been the usual practice in which water are applied irregularly across the cross-sectional area of the space in question. This problem could be eliminated having a soil compacting machine with an integrated means of water supply.

Also, the current soil compacting machine in use for building and road constructions most often put the health of the operators in danger as a result of vibrating signal being transmitted from the handle of the machine to them (Hand-arm vibration). The effect of this is very eminent. This hazard could be averted by the introduction of anti-vibration tools at the joints or along the handle of the compactor.

Procurement cost of the imported type is unaffordable to the local contractors. This project has price comparative advantage over the existing one despite the additional components. The market price of the existing soil compacting machine is higher. The materials used in this project are locally sourced and readily available.

Materials and Methods

A required power is the power required by the vibrator for efficient performance and it is given by

$$P_r = F_c \times v \quad (1)$$

$$\text{Recall: } v = \omega r = 2\pi f$$

Where; F_c is centripetal force, v is linear velocity, ω is angular velocity, r is the amplitude and f is maximum frequency. Hence; Design power, P_d which is the power that gives room for the safety of the machine is the product of the required power and service factor. This can be obtained as given in equation (2).

$$P_d = P_r \times s.f \quad (2)$$

According to Carlisle 2011 the service factor (s.f) for machine with normal service and normal duty is 1.3.

The Analysis of the eccentric Rotary shaft

Based on the equations (1) and (2) above, the eccentric rotary shaft was designed. The eccentric rotary shaft is the component that transmits rotational motion and power from the two stroke petrol engine to the vibrator through belt and pulley system. In the process of transmitting power at a given rotational speed, the shaft is inherently subjected to a torsional moment or torque. Thus, the torsional shear stress is developed in the shaft. Also, it carries a power-transmitting component (belt shelves) which exerted forces on the shaft in the transverse direction (perpendicular to its axis). This transverse forces cause bending moments to be developed in the shaft, thereby requiring analysis of the stress due to bending (Robert and Mott, 1992: 282-317). The position of forces acting on the shaft are as indicated in Fig.1.

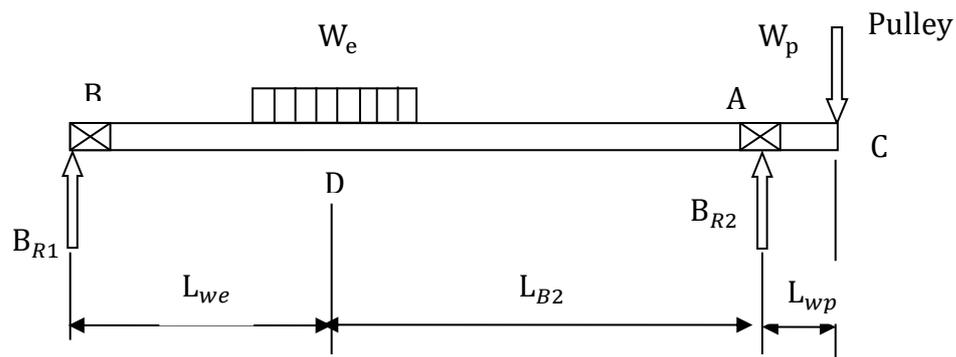


Fig. 1: The eccentric shaft arrangement

Considering the figure above; B_{R1} and B_{R2} are the reactions at the bearings that support the shaft at A and B respectively; W_p is the weight of the pulley at point C; L_{ecc} is the length of the eccentric weight; L_{we} , L_{B2} , L_{B2} and L_{wp} are the lengths of the eccentric shaft between point BD, DA and AC.

The uniform distributed load, F_u over the eccentric weight, W_e was obtained by;

$$F_u = \frac{W_e}{L_{ecc}} \tag{3}$$

$$W_e = 2M_w \cdot g \tag{4}$$

Where; g is the acceleration due to gravity.

According to A. Ogaga and A. Sule (2000: 60-72) the distributed load can be resolved to a single force, F_e using the equation below.

$$F_e = F_u \cdot \frac{L_{ecc}}{2} \tag{5}$$

The masses of the pulley and the bearing were measured using weight balance.

The weight of pulley, W_{pulley} and bearing, $W_{bearing}$ were obtained as follows;

$$W_{pulley} = M_{pulley} \times g \tag{6}$$

$$W_{bearing} = M_{bearing} \times g$$

The reactions at the bearings were obtained by resolving the forces acting on the shaft.

$$B_{BR1} + B_{BR2} = W_e + W_p \tag{7}$$

Taking moment about point B_{BR1} , we have;

$$L_{we} \cdot L_{B2} \cdot B_{BR2} = L_{we} \cdot W_e + L_{we} \cdot L_{B2} \cdot L_{wp} \cdot W_p$$

$$B_{BR2} = \frac{1}{L_{we} \cdot L_{B2} \cdot B_{BR2}} (L_{we} \cdot W_e + L_{we} \cdot L_{B2} \cdot L_{wp} \cdot W_p) \tag{8}$$

The negative sign indicates that the direction of the reaction at B_{BR1} is in opposite direction.

The maximum bending moment was obtained from the bending moment diagram in Fig. 2.

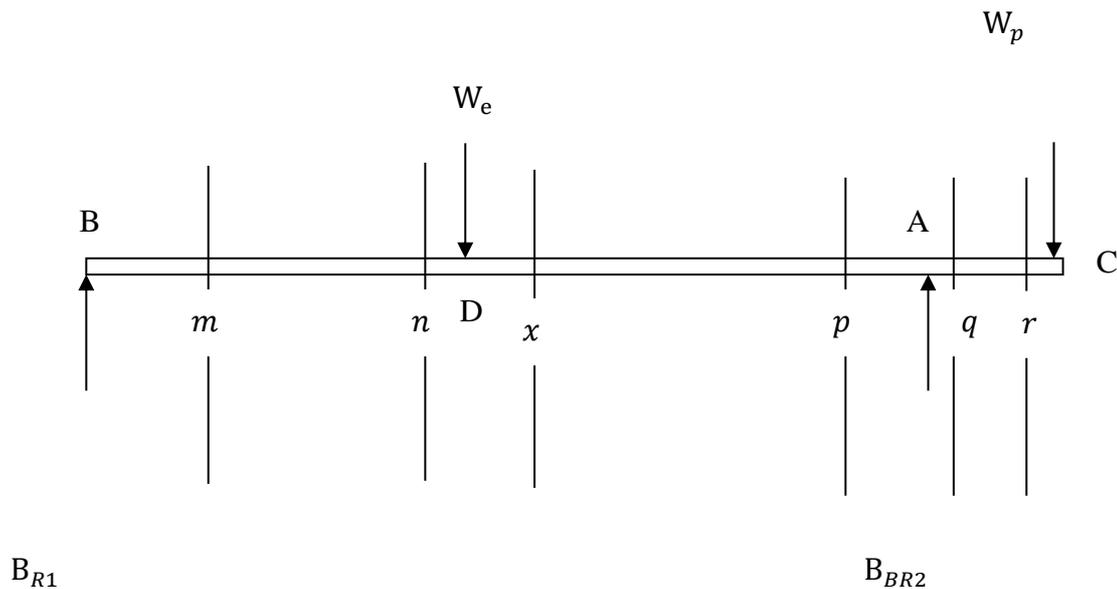


Fig. 2. Diagrammatical representation of the resolved distributed load with selected elemental parts

The bending moment was calculated over the selected elemental parts, $m, n, x, p, q,$ and r .

The maximum bending moment, M_{bm} was obtained.

The torsional moment on the eccentric shaft was obtained as follow;

$$M_{Ts} = (t_1 - t_2) \cdot \frac{D_c}{2} \tag{9}$$

Where; t_1 is tension in the tight side of the v-belt; t_2 is tension in the slack side of the v-belt; D_c is diameter of the pulley attached to the crankshaft of the two stroke petrol engine. Hence, the diameter of the eccentric shaft was calculated thus;

$$d_{es}^3 = \frac{16}{\pi S_s} \sqrt{\{(k_b M_{bm})^2 + (k_t M_{Ts})^2\}} \tag{10}$$

Where; S_s is allowable shear stress (N/m²) = 55 x 10⁶ N/m²; k_b is combined shock and fatigue factor applied to bending moment = 1.5; k_t is combined shock and fatigue factor applied to torsional moment = 1.0

Hence, the angle of twist between bearing and the pulley was determined using equation (11);

$$\theta = \frac{584 M_{Ts} x}{G d_{es}^4} \tag{11}$$

Where; θ is angle of twist; M_{Ts} is the torsional moment; x is the length of shaft between the applied and resisting torque; G is torsional modulus of elasticity = 80 GN/m² and d_{es} is diameter of eccentric shaft.

Design Analysis of Pulley and Belt System

The V-belt pulleys (also known as vee belt sheaves) are devices which transmit power between shafts by the use of a v-belt, a mechanical linkage with a trapezoidal cross-section. The combination of these devices offers a high-speed power transmission solution that is resistant to slipping and misalignment. The bore diameter of the pulley is geometrically designed to match with the mating shaft. The keyway on the bore of the

pulley and face of the vibrator shaft ensures an intimate fitting along the shaft to transfer torque between components.

Diagram showing the parameters involved in belt drive system design for the machine are shown in Fig. 3 and Fig.4 respectively.

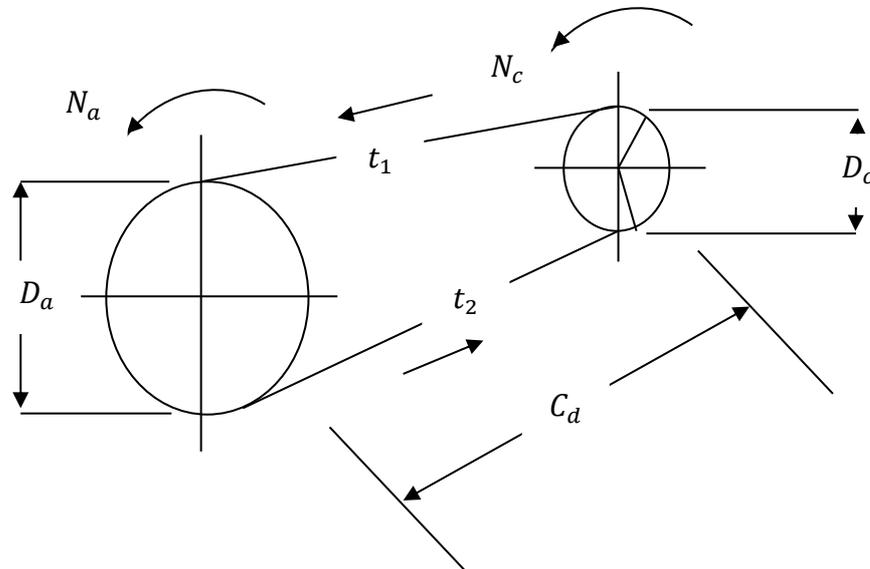


Fig. 3. Pulley and belt drive system



Fig. 4. Image of the pulley

D_a is diameter of the pulley attached to the shaft of the vibrator = 13 cm; D_c is diameter of the pulley attached to the crankshaft of the two stroke petrol engine = 5 cm; N_a is number of revolution of the pulley attached to the shaft of the vibrator; N_c is number of revolution of the pulley attached to the crankshaft of the two stroke petrol engine = 3500 rpm; t_1 is tension in the tight side of the v-belt; t_2 is tension in the slack side of the v-belt; C_d is centre distance between the two pulleys; θ_c is angle of contact of the belt on the pulley of two stroke petrol engine and θ_a is angle of contact of the belt on the pulley attached to the shaft of the vibrator.

The relationship between the number of revolutions and diameters of two pulleys connected by belt according to Khurmi and Gupta, (2009) is given as:

$$\frac{N_c}{N_a} = \frac{D_a}{D_c} \tag{12}$$

$$N_a = \frac{N_c \cdot D_c}{D_a}$$

The maximum, C_{max} and minimum, C_{min} centre distance between two pulleys is as given in equations (13) and (14) respectively in reference to Joseph and Charles, (2001).

$$C_{max} = 3(D_a + D_c) \tag{13}$$

$$C_{min} = 0.55(D_a + D_c) \tag{14}$$

The centre distance, C_{ac} between the two pulleys was calculated using equation (15).

$$C_d = \frac{C_{max} + C_{min}}{2} \tag{15}$$

The belt pitch length was determined according to Khurm and Gupta, (2009) using the equation (16a).

$$L = 2C_d + \frac{\pi}{2} (D_c + D_a) + \frac{1}{4C_d} (D_c - D_a)^2 \tag{16a}$$

In reference to Khurmi and Gupta, (2009) the width and thickness for a standard v-belt transmitting power between 2 kW to 15 kW are 1.7 cm and 1.1 cm respectively.

Therefore, the belt area was calculated using equation (16b).

$$A_b = w \times t \tag{16b}$$

Where; w is width and t is the thickness of the belt.

In reference to www.gatemictrol.com the belt mass, M_b was obtained from the specific belt weight, W_b and belt length, L and width, b .

$$M_b = \frac{W_b \cdot L \cdot b}{g} \quad \text{Or} \quad M_b = A_b \times L \times \rho \tag{17}$$

where; A_b is the area of the belt, ρ is the density of the belt.

The speed of the belt according to Khurmi and Gupta, (2009) is given as;

$$V_{bc} = \frac{\pi D_c \times N_c}{60} \tag{18}$$

The centrifugal tension in the belt was determined using equation 19.

$$t_c = M_b \cdot V_{bc}^2 \tag{19}$$

The maximum tension in the belt, T was calculated applying equation 20.

$$T = \sigma \cdot A \tag{20}$$

Where; σ is allowable tensile stress in the belt and it is given as 2 MPa.

The tension in the tight side of the belt is calculated thus;

$$t_1 = T - t_c \tag{21}$$

To calculate the tension in the slack side, t_2 the equation (22) was used and t_2 was made the subject of the formula.

$$e^{\mu\theta} = \frac{t_1 - M_b \cdot V_{bc}^2}{t_2 - M_b \cdot V_{bc}^2} \tag{22}$$

$$t_2 = \frac{t_1 - M_b \cdot V_{bc}^2}{e^{\mu\theta}} + M_b \cdot V_{bc}^2 \tag{23}$$

Where; θ is the angle of contact of the belt in the smaller pulley i.e. the pulley attached to the crankshaft of the two stroke petrol engine; μ is the coefficient of friction between the belt and the groove of the pulley attached to the shaft of the vibrator.

According to Khurmi and Gupta (2009) the coefficient of friction between belt made of canvas and a steel pulley operating under dry condition is 0.25.

Therefore; the angle of contact must be gotten before the tension on the slack side of the belt was calculated.

The angle of contact was calculated using equation (24);

$$\theta = 180^\circ - \frac{57.3(D_a - D_{ca})}{C_{ac}} \tag{24}$$

From equation (23)

$$t_2 = \frac{t_1 - M_b \cdot V_{bc}^2}{e^{\mu\theta}} + M_b \cdot V_{bc}^2$$

The power transmitted according to M. Joshi (1976: 173-370) by two stroke petrol engine is given as;

$$P = (t_1 - t_2)V_{bc} \tag{25}$$

To determine the angular velocity of the two pulleys equation (26) was used.

$$VR = \frac{D_a}{D_c} = \frac{N_c}{N_a} = \frac{\omega_c}{\omega_a} \tag{26}$$

Where; VR is velocity of the two pulleys; ω_c is angular velocity of the pulley attached to the crankshaft of the two stroke petrol engine and ω_a is the angular velocity of the pulley attached to the shaft of the vibrator.

In reference to R. Khurmi and J. Gupta (2009: 520-1096) the angular velocity of the pulley connected to the crankshaft of the petrol engine was obtained using equation (27).

$$\omega_c = \frac{2\pi N_c}{60} \tag{27}$$

From equation (28) the angular velocity, ω_a of the pulley attached to the shaft of the vibrator was determined.

$$\omega_a = \frac{\omega_c}{VR} \tag{28}$$

Engineering Drawings of the soil Compactor

The engineering drawing of this machine includes the orthogonal, the isometric and the expanded views as shown in Fig. 5 and Fig. 6.

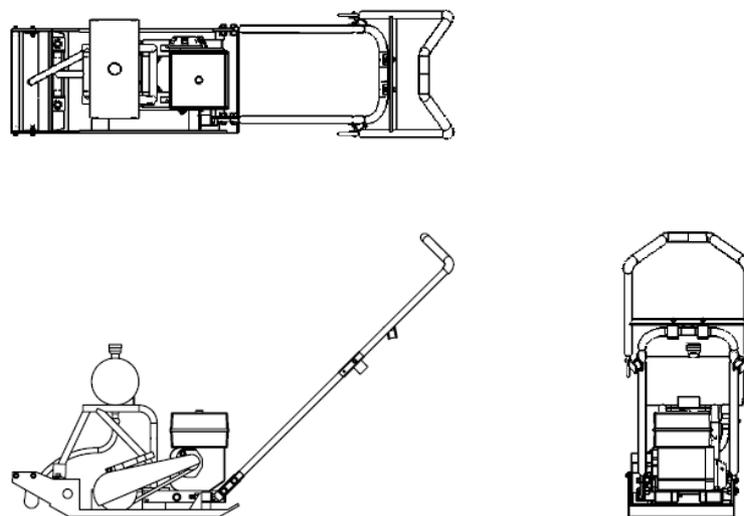


Fig. 5. Orthogonal drawing of Hand-pushed vibratory soil compacting machine

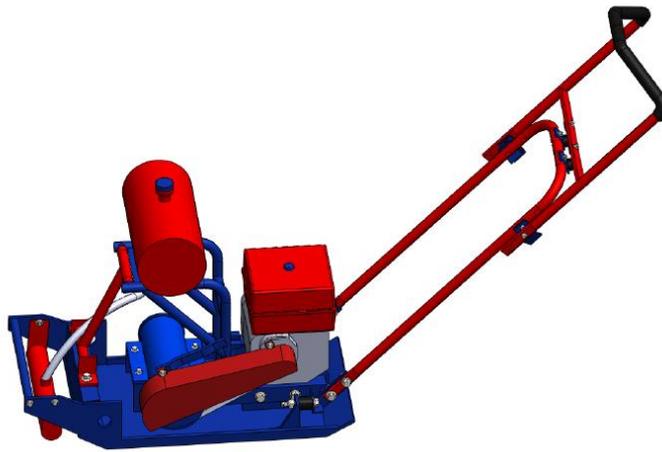


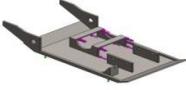
Fig. 6. The isometric drawing of the hand-pushed vibratory soil compacting machine

Results

The results of the Analysis of the Machine Chassis using solid works tools are presented below.

Model Information of the chassis is systemized in Table 1.

Table 1. Model Information of the chassis

 Model name: Chassis Current Configuration: Default			
Solid Bodies			
Document Name and Reference	Treated As	Volumetric Properties	Document Path/Date Modified
Cut-Extrude9 	Solid Body	Mass:30.7803 kg Volume:0.0039211 m ³ Density 7850 kg/m ³ Weight:301.6469 N	C:\Users\Esan\Documents\SOLIDON\ESAN\Base.SLDPRT May 25 08:32:02 2016

Material Properties Data are available in Table 2.

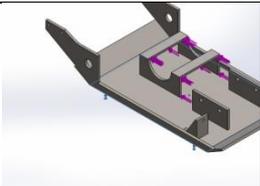
Table 2. Material Properties Data.

Model Reference	Properties	Components
	Name: 1023 Carbon Steel Sheet (SS) Model type: Linear Elastic Isotropic Default failure criterion: Max von Mises Stress	Solid Body 1(Cut-Extrude9)(Base)

Yield strength:	2.82685e+008 N/m ²	
Tensile strength:	4.25e+008 N/m ²	
Elastic modulus:	2.05e+011 N/m ²	
Poisson's ratio:	0.29	
Mass density:	7850 kg/m ³	
Shear modulus:	8e+010 N/m ²	
Thermal expansion coefficient:	1.2e-005 /Kelvin	

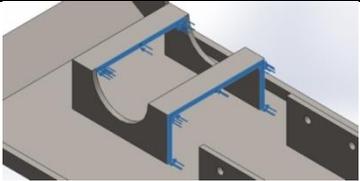
Fixture Data are systemized in Table 3.

Table 3. Fixture data

Fixture name	Fixture Image	Fixture Details		
Fixed-1		Entities:	1 face(s)	
		Type:	Fixed Geometry	
Resultant Forces				
Components	X	Y	Z	Resultant
Reaction force(N)	169.977	0.202674	0.00665736	169.977
Reaction Moment(N.m)	0	0	0	0

Load Data are systemized in Table 4.

Table 4. Load Data

Load name	Load Image	Load Details		
Force-1		Entities:	2 face(s)	
		Type:	Apply normal force	
		Value:	85 N	
		Phase Angle:	0	
		Units:	Deg	

Mesh Information is given in Table 5.

Table 5: Mesh Information

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Automatic Transition:	Off
Include Mesh Auto Loops:	Off

Jacobian points	4 Points
Element Size	15.3089 mm
Tolerance	0.765445 mm
Mesh Quality	High
Mesh Information - Details	
Total Nodes	21760
Total Elements	10486
Maximum Aspect Ratio	9.8572
% of elements with Aspect Ratio < 3	94.5
% of elements with Aspect Ratio > 10	0
% of distorted elements(Jacobian)	0
Time to complete mesh (hh:mm:ss):	00:00:03

The Result of the Mesh are presented in a pictorial outlook (Fig. 7)

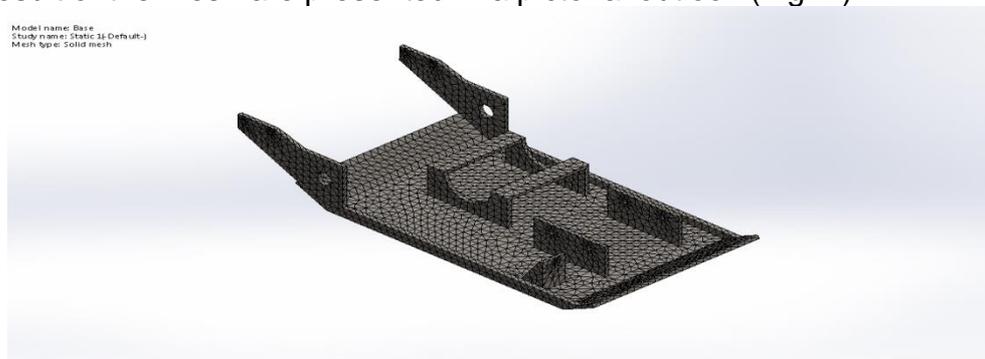
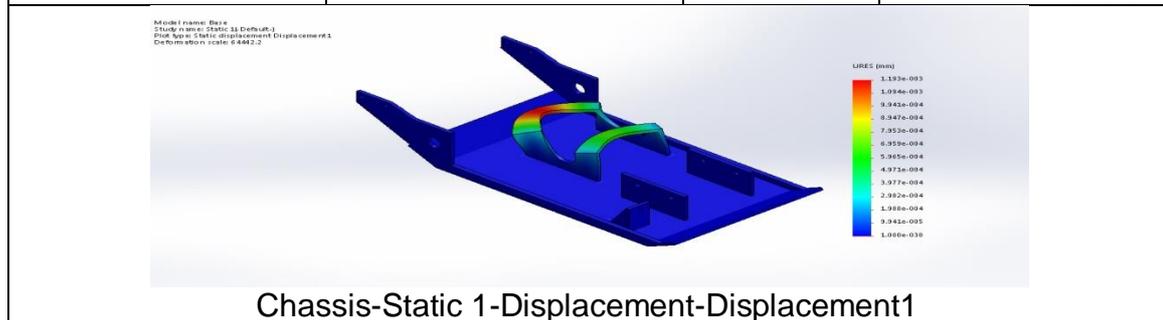


Fig. 7. Pictorial outlook of the result of the mesh of the chassis of the machine

The Displacement Analysis of the Chassis was summarized in Table 6.

Table 6. The Displacement Analysis of the Chassis

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 1030	0.00119297 mm Node: 6188



The Stress Analysis of the Chassis can be observed in the Table 7 below.

Table 7. The Stress Analysis of the Chassis

Name	Type	Min	Max
Stress1	VON: von Mises Stress	0.000174237 N/m ² Node: 14221	1.04556e+006 N/m ² Node: 6294

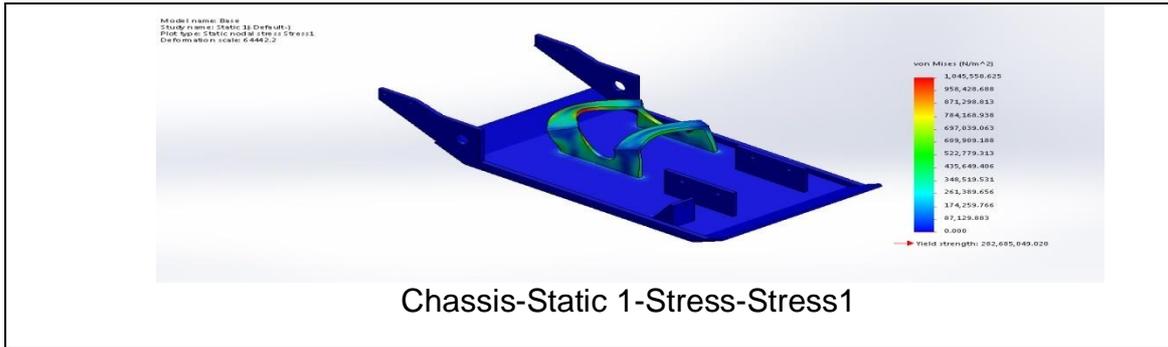


Table 8 presents the strain analysis of the chassis.

Table 8. The Strain Analysis of the Chassis

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	4.23467e-016 Element: 6913	3.25391e-006 Element: 9563

Model name: Base
Study name: Static 1 (Default)
Plot type: Static strain (Strain1)
Deformation scale: 0.444272

ESTRN

- 3.254e-006
- 2.983e-006
- 2.712e-006
- 2.440e-006
- 2.169e-006
- 1.898e-006
- 1.627e-006
- 1.356e-006
- 1.085e-006
- 8.135e-007
- 5.423e-007
- 2.712e-007
- 4.235e-016

Chassis-Static 1-Strain-Strain1

Discussion

The model of the chassis is as shown in Table 3.1. It gives the summary of the required model information for the chassis of the machine. The weight of the chassis is 301.6469 N and its density is 7850 kg/m³. The arrow symbols indicate the direction of the movement of the machine when in operation. With or without any force from the operator the engine tends to move in forward direction.

Table 3.2 spells out the summarized properties of the chassis of the machine. Its tensile strength, elastic modulus and shear modulus are 4.25 x 10⁸ N/m², 2.05 x 10¹¹ N/m² and 8 x 10¹⁰ N/m² respectively.

Table 3.3 shows that the components that made up of the chassis are fixed as they are welded together. At the vibrator seat the prevailing force that is acting there is the one that tend to move the machine forward though the weight of the vibrator is acting perpendicular to the axis of the vibrator. Therefore, the minimum force that can cause the movement of the machine is rated 85 N. the reaction force on x, y and z axes are 169.977 N, 0.202674 N and 0.00665736 N.

Mesh gives specification as regard the allowable tolerance between the welded components of the chassis and this is given as 0.765445 mm. If the Jacobian value is negative it means that the structure will fail. The aspect ratio an element is defined as the ratio between the longest edge and the shortest normal dropped from a vertex to the opposite face normalized with respect to a perfect tetrahedral. By definition, the aspect ratio of a perfect tetrahedral element is 1.0. It checks assumes straight edges connecting the four corner nodes. Its check is automatically used by the program to check the quality

of the mesh. The chassis is designed such that the percentage of aspect ratio that is less than 3 of its elements is not less than 94.5.

The maximum allowable displacement of the vibrator seat is 0.00119297 mm. The significance of this is that the vibrator seat should be rigid with zero degree of movement.

The maximum allowable stress that the vibrator seat could be subjected to is 958,428,688 N/m². Stress above this will cause the failure of the seat. Hence, the seat is to be designed in respect of this fact.

Conclusion

The machine was designed to have a chassis mass of 32.68 kg, water reservoir of 6.57 x 10⁻³ m³, the discharge rate for soil wetting is 3.49 x 10⁻³ m³ s⁻¹, and compacting rate of the vibrator is 5000 vpm with a power of 4.08 KW (5.5HP). The machine chassis was analyzed using solid works tools. The actual chassis model information is: mass 30.78 kg, volume 0.003921 m³, density of 7850 kg/m³ and weight is 301.647 N. The material properties are: Name (1023 Carbon Steel sheet), Model type (Linear Elasticity Isotropic), Default failure criterion (Max. von, mises stress), Yield strength (2.82685 x 10⁸ N/m²), Tensile strength (4.25 x 10⁸ N/m²), Elastic modulus (2.05 x 10¹¹ N/m²), Poisson's ratio (0.29), Mass density (7850 kg/m³), Shear modulus (8 x 10¹⁰ N/m²) and Thermal expansion coefficient (1.2 x 10⁻⁵ Kelvin). The mesh information is: Mesh type (solid mesh), Mesher used (standard mesh), Automatic transition (off), Include mesh auto loops (off), Jacobian points (4), element size (15.3089 mm), Tolerance (0.765445 mm), Mesh quality (High), total nodes (21760), Total elements (10486), Maximum aspect ratio (9.8572), % of elements with aspect ratio < 3 (94.5), % of elements with aspect ratio >10 (0), % of distorted elements Jacobian (0), Time to complete mesh (hh: mm: ss) (00:00:03). The maximum allowable displacement of the vibrator seat is 0.00119297 mm. The significance of this is that the vibrator seat should be zero degree of movement. The maximum allowable stress that the vibrator seat should be subjected to is 958,428,688 N/m². Stress above this will cause the failure of the seat. The maximum allowable strain that the vibrator seat could be subjected to is 2.983 x 10⁻⁶. Strain above this will cause the failure of the seat. The results of these analyses proved that the machine developed is good enough for use and can be released to the market.

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