DEVELOPMENT AND PERFORMANCE EVALUATION OF A RECONFIGURABLE PLATFORM FOR A MICROSATELLITE PRODUCTION

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Abstract: The process of satellite assemblage, integration and testing is challenging and requires considerable care. Hence, the provision of a support platform which provides easier mechanical mounting and unmounting within a design facility and its sources of mechanical disturbance. The provision of an indigenous low-cost platform to cater for microsatellite production is sought, and in this research, a partially reconfigurable platform to produce microsatellites (mass range between 10-100 kg) was developed. The materials used for the fabrication were carefully selected, and considerable attention was placed on the overall weight of the platform based on various criteria such as Density, Yield strength, Modulus of elasticity. The Computer-Aided-Design (CAD) for the various parts using Solid works was done, which also involved the design in accommodating the two standardized diameters for launch vehicle adapters for microsatellites. The design calculations necessary for the completion of this work and weight budget analysis was done to determine the final weight of the platform. Analysis System (ANSYS) software was used to validate the structural integrity of the platform under loading conditions. The stress analysis of the platform in a vertical position which is the most critical loading case was performed. Also, there was no mechanical damage or failure found on the platform. The procedure adopted for the development of the equipment can be used on the development of similar projects. The developed machine is both manually and mechanically operated. The developed platform has the capacity of carrying a maximum satellite weight of 150 kg.

Keywords: Construction, Design, Microsatellite, Vehicle adapter, Von Mises stress,

1. INTRODUCTION

Satellite configuration design should provide easy access between the satellite, the launch vehicle adapter and mechanical ground support equipment (MGSE). To provide this, the satellite configuration must include suitable mechanical interfaces to meet both launch vehicle and the mechanical ground support equipment (Kim *et al*., 2013). The configuration of a satellite's primary structure can be characterized by its architecture, type and packaging scheme (Griffin and James 1999). Generally, Satellites are attached to both the ground support equipment and the rocket during launch phase by means of the satellite adapter which form part of the separation system (Kim *et al.*, 2013). They are mainly flight modules because they are ejected together with the satellite at the launching phase.

Microsatellites, two standard diameter adapters are provided as standards for all missions. Usually made from aluminum alloys, the standard adapter diameter for microsatellites is the 8inch and 15inch diameter mounting port and cuts across all missions that involve microsatellites Both are standard small satellite interfaces compatible with light band

separation systems, in keeping with the Operationally Responsive Space (ORS) objectives of standardization and common Interface Control Documents (ICDs) (Maly *et al*, 2009).
The process of satellite assembly, integration and test is exacting and requires considerable care (Bruno 1993). Hence, the provision of a support platform which provides easy mechanical mounting and decoupling within an assembly facility and its sources of mechanical disturbance is desired. These support platforms allow handling within the facility while protecting the satellite or primary subsystem from harm (Maly *et al*., 2009). It also allows for the excellent accessibility of the satellite structure from all sides and the tilting and rotation of the structure. It also supports satellite or dedicated modules during horizontal or vertical integration (Griffin and James, 1999). Existing mechanical ground support equipment (MGSE) for this function are purposely designed for sizes of satellites as required by client space-faring countries according to planned space missions.

Reconfigurable machines (RM) form a new class of machines that are designed around a specific part family of products and allow rapid change in their structure. They are designed to allow changes in machine configuration according to changes in production requirements. The reconfiguration may be related to changes in machine functionality or scalability, i.e. the change in production volume or speed of operation (Yee, 2005; Chandrupatla and Belegundu, 2007; Abdelal *et al*., 2013). Reconfigurable machines represent a new class of machines that bridges the gap between the high flexibility and high cost of totally flexible machines and the low flexibility and low cost of fully dedicated machines. The design principles of reconfigurable machines follow a similar philosophy, which was derived for a reconfigurable manufacturing system and present an approach for the design of machines to be used mainly in high volume production line (Katz, 2006; Choi et al., 2015).

1.1 Purpose of the study

This research, a partially reconfigurable platform to produce microsatellites (mass range between 10-100 kg) was designed, fabricated, and the performance evaluated.

2. METHODOLOGY

2.1 Design concept

The design methodology surveys the principle of satellite configuration and tends to understand the various configurations and approaches. Solid Works design software was used for the CAD design while Ansys Multiphysics was used for the structural analysis to determine the structural integrity at critical loading case. The conceptual design and the structural analysis is shown in figure 1 to 4. Also, a physical or otherwise termed workshop loading case was done using dummy models whereby various loading cases was examined to verify the functionality of the platform. The mechanical arm of the structure responsible for the tilting motion of the platform was connected to a threaded shaft that is driven by a set of spur gears connected to a well sized electric motor by a worm gear; a plunger stand to increase the stability of the equipment while in operation is manually operated by turning the handle either clockwise or anticlockwise for extension and retraction respectively. section should include what, where and how of your research work. It should cover the details about location of your research work, sample size, tools you will be using (such as interviews, surveys or experiment/s) to collect data. The Research Questions (RQs) and Hypothesis (H) that will explore and tested should also be listed here. It will also mention statistical analysis tools and technologies that will be used to analyze data. In a nutshell, this section should provide insight into how you will conduct your research work.

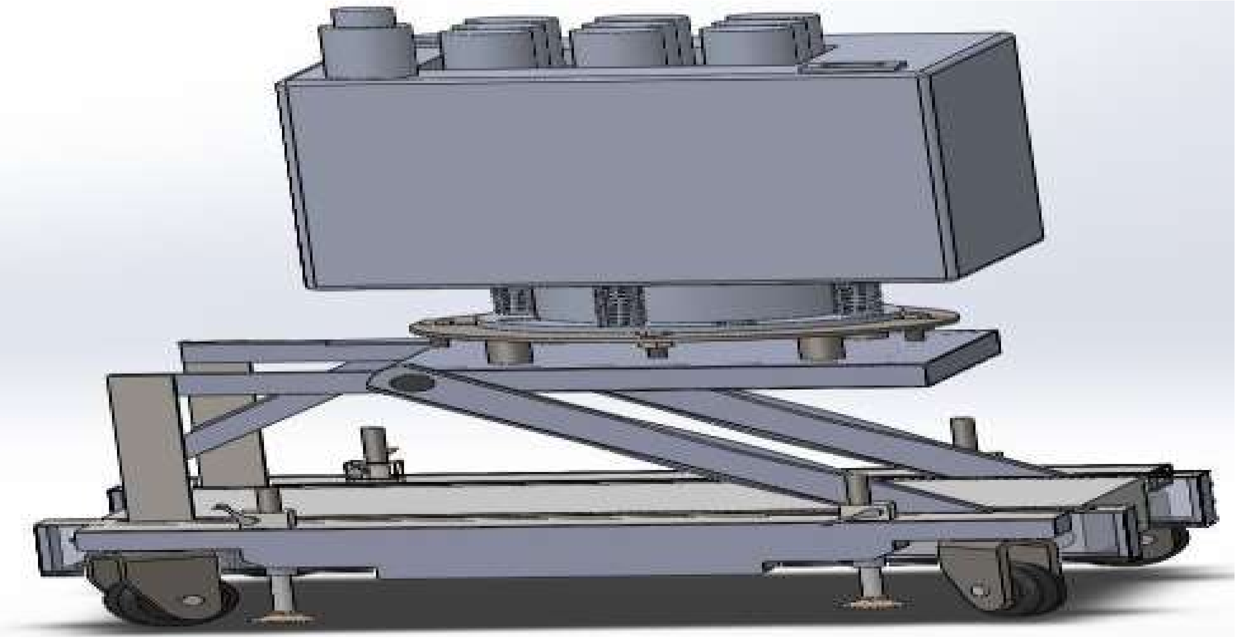
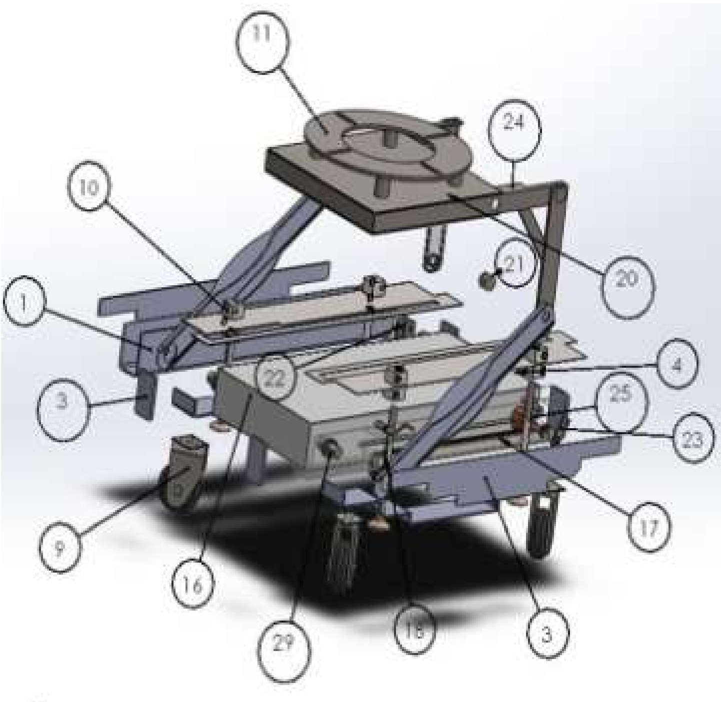


Figure 1: Isometric view



Part No.	Part Name	Qty.
1	Slotted U-channel	2
2	Rail unit	2
3	Rail unit cover	2
4	I beam	2
9	Wheel assembly	1
10	Mast link a	1
11	Clamp	1
16	Centre table	1
17	Rack spur rectangular iso	1
18	Plunger shaft	1
20	Table-support sub	4
21	Roller pin	1
22	Roller assembly	1
23	Brace adjustment	1
24	Link brace	1
25	I Beam bearing housing	2
29	Plunger handle	4

Figure 2: exploded view and machine part list

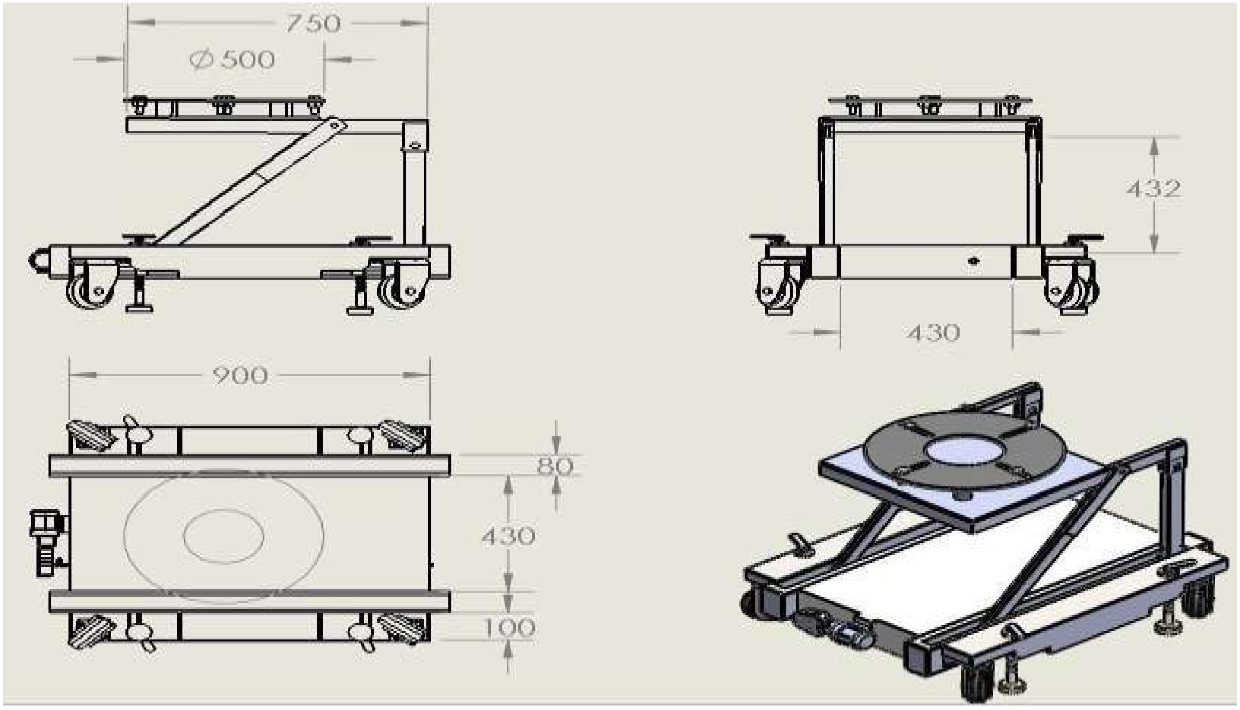


Figure 3: Orthographic view of the designed machine

2.2 Design analysis

The platform is designed to carry mass range between 10 kg to 100 kg. Different design factors were taken into consideration, as follows: the size of the machine, i.e. the dimensions of the machine: Length = 900 mm, breadth = 650 mm and height = 600 mm

Design for threaded shaft

Determination of Minimum Diameter of Shaft. According to Khurmi and Gupta, (2009). Bending moment M_b of the shaft

$M_b = (\sigma_b \times \pi \times d^3) / 32$ (2)

M_b is the bending moment (Nm)

$d = \sqrt[3]{((32 \times M_b)/ (\pi \times \sigma b))}$ (3)

$d = 0.036m$

Determination of Maximum Permissible Tensile Load on Threaded Shaft

When the distance between loading points is short, according to Slocum, (2008). It is necessary to examine the permissible tensile load following equation 4 independently of the supporting method.

$P= \sigma \cdot A$ (4)

Where P is the Permissible tensile load (N)

σ is the Permissible stress of standard mild steel

A is Sectional area at screw shaft root diameter (mm^2) $= \pi (d / 2)^2$ (5)

$P = 2.5 \times 10^4$ N is way greater than the designed weight of 1.47×10^3 N, indicating a permissible design to avoid the bending of the shaft while in operation.

Determination of maximum buckling load on the threaded shaft

Joseph *et al*. (2001) stated that when the screw shaft is subject to compression load, it is necessary to take measures to prevent buckling following equation 5;

a is the Safety factor (0.5)

Z is the distance between loading points (mm)
I is the Minimum secondary moment of screw shaft cross-section (mm4)
$I = \pi/64d^4$ (6)
Where d is the screw shaft root diameter (mm)
n = factor determined by supporting the method of ball screws
n = 1 (Both ends supported)
$I = 2485\ mm^4$
Hence, permissible axial load to buckling $P = (n.\pi^2.E.I)/Z$ (7)
$P_b = 8.04\times 10^5\ N$
This value indicates the maximum buckling load on the threaded shaft by the load.
Determination of critical speed of threaded shaft
It is necessary to examine critical speed using equation 8 so that the number of revolutions of the nut with the natural frequency of the screw shaft.
$N = (60\lambda^2)/ (2\pi L^2) \sqrt{((EIg) / (\lambda.A))}\ \alpha$ (8)
N = 0.14rpm
Gearing System
Determination of transmission ratio (tr) of spur gears
Transmission ratio denotes a relationship between the input and output directions of a set of gear. According to Slocum, (2008).
$T_R = -1$
A positive T_R value indicates the same rotation direction in both the output and input gears, but a negative T_R value as obtained above indicates that the output rotation direction is opposite to the input rotation direction.
Determination of pressure angle to avoid interference
According to Hall *et al*., 1998, to maximize the load-carrying capacity of a set of gears, a 20º Pressure angle tooth shape should be chosen because it is stronger and has a better tooth because of its broader base especially on gears with the small number of teeth hence it can withstand more force.
Determination of the minimum number of teeth in contact
The circular pitch can be obtained, according to Avallone *et al*., 2007. Using the equation 10,
$P_c = \pi d/T$ (10)
Where d is the diameter of the gear (125mm)
T is the number of teeth (20)
the number of pairs of teeth in contact is given by equation 11,
(Length of the arc of contact)/ (circular pitch (Pc)) (11)
∴ Minimum number of teeth in contact = 3 or more pair of teeth in contact
Determination of the circular pitch of the Spur Gear
Due to design considerations, build and ease of maintenance, according to Avallone *et al*., 2007, an ISO Spur Gear of 20 number of teeth at 18º between corresponding teeth was chosen as this presented a strong set of teeth for loading.
Number of teeth = 20 at 18º between each one.
Pitch Circle = 20mm, (0.787 inch)
Gear Pitch = (number of teeth)/ (Pitch Circle diameter) = 20/20 = 1 Pitch gear in mm (12)
Circumference of pitch circle = $2\pi r$ (13)
Where r = pitch circle radius (20/2)
Circumference of pitch circle = $2 \times \pi \times 10 = 62.83$ mm (2.47 inch)

Therefore, Circular pitch = (circumference of the pitch circle) / (no of teeth) (14)
Circular Pitch = 2.47/20 = 0.1235 circular pitch.
So, a 0.1235 circular pitch spur gear is chosen for the design.
Determination of velocity of the driver gear. The gear ratio of gear can be obtained using equation 15,
Gear ratio = (number of teeth on the driven) / (number of teeth on the driver) (15)
Gear ratio = 50/15 = 3.33
The gear ratio of 3.33 determines the driver speed of the motor.
Known, Gear velocity = Pitch diamter×0.262 rev (of the gear) per min(rpm) (16)
Assume Gear rotates at 0.5 revolutions per second hence (0.5 x 60) revolutions per minute, i.e. 30 rpm.
Gear Velocity = 0.02 m × 0.262 × 30 rpm = 0.1572 m/s
Determination of Torque required to be developed by gear to overcome the load. The torque required to be developed to overcome load is obtained using equation 17,
Torque $T = T_{(w)} (w_{sat} + w_{st}) \times r_{in}$ (17)
Where, $T_{(w)}$ = Total weight (Operating weight)
w_{sat} = weight of the microsatellite
w_{st} = Weight of the structural members of the platform
r_{in} = radius of driver gear, i.e. input gear.
Torque =102 Nm
Determination of the Power transmitted by spur gear. The circumference of the pitch circle can be obtained using equation 18,
C_{pc} = No of teeth $\times P_{sg}$ (18)
Where, No of teeth = 34
P_{sg} = Pitch of the spur gear (20)
C_{pc}= 34 x 20 = 680mm
The velocity V of the driver gear can be obtained using equation 19,
$V= C_{pc} \times s_{dg}/60$ (19)
Where, s_{dg} = speed of the driver gear (30 rpm)
C_{pc} = Circumference of the pitch circle
V= 0.68 × 30/60
V = 0.34 m/s
The Power P transmitted by the spur gear is obtained using equation 20,
$P= F_{(tan)}\times V$ (20)
Where V = Velocity of the driver gear
$F_{(tan)}$ = Tangential force to overcome (Operating weight)
Tangential force to overcome = (150 + 190) kg = 340 kg = 3335.4 N
Power transmitted = (3335 x 0.34) W = 1.13 kW
Design of plunger stand
Determination of the minimum allowable diameter. Recall that the operating weight for the design is 340 kg and in terms of newton force (F), i.e. 340×9.81 = 3334.4 N, this value denotes the overall weight of the equipment together with the microsatellite in place. However, to calculate the minimum diameter of the plunger stand, the overall weight was divided by 4 because there are 4 plunger stands in the design and it is assumed that the weight was distributed evenly on all 4 stands.
Thus, weight acting on each stand was, 3335.4/4 = 833.85 N

The bending moment M_b of the plunger stand is obtained using equation 21,

$M_b = m_p \times l$ (21)

Where m_p = load carried by each plunger stand

l = length of the plunger stand (290 mm)

$M_b = 833.85 \times 0.290 = 241.82$ Nm

According to Khurmi and Gupta 2009, the minimum allowable diameter d of the shaft is obtained by making d the subject of formula in equation 22,

$M_b = (\sigma_b \times \pi \times d^3) / 32$ (22)

$d = \sqrt[3]{(32 \times M_b) / (\pi \times \sigma_b)}$ (23)

Where σ_b is the bending stress (N) of mild steel 165×10^5 N/m^2

$d = \sqrt[3]{(32 \times 241.82) / (3.142 \times 165 \times 10^6)}$

$d_{min} = 0.025$m

This Value represents the minimum diameter the plunger stand can be designed to avoid failure in design at the yield strength of mild steel of 2.5×10^8 N/m^2, however, in other to minimize the weight of the equipment and therefore final cost, a more considerable diameter was considered that resulted in a Von Mises stress value lesser than the Yield strength of Mild steel.

Determination of the Minimum effort required to be applied at plunger handle. The effort P_m required at mean radius r_m of the thread to lift load is obtained using equation 24,

$P_m = W\tan(\alpha+\varphi)$ (24)

Where α is the Helix angle and obtained from equation 25,

$\tan \alpha = P/\pi d$ (25)

Where p = pitch of gear (10)

d = mean diameter (60mm)

$\tan \alpha = 10 / (\pi \times 50) = 0.053$,

The value φ is obtained from the Coefficient of friction between screw and nut in equation 26,

$\mu = \tan \varphi$ (26)

where $\mu = 0.15$

Thus $\varphi = \tan^{-1} \mu = \tan^{-1} 0.15$

$\varphi = 8.5^{o}$

From equation 3.27, effort P_m required at mean radius r_m of the thread to lift load is calculated thus,

$P_m = W \times \tan(3.04° + 8.5°)$

Where W = operating weight (3335.4 N)

$P_m = W\tan(11.54°) = 3335.4 \times 0.203 = 678.6$N

Effort P_1 required at the end of the handle may be obtained from the relation in equation 27,

$P \times 90 = P_m \times r_m$ (27)

Where, r_m = mean radius (30mm)

$P = (P_m \times r_m) / 90$

$P = 226.2$ N

An effort of value 226.2N is required to be applied at one of the plungers stands to raise the load.

Electric motor sizing and selection

The power required as expressed by Khurmi, (2001) is given by equation (28);

Power=W×2πR/12×Revolutions per minute (28)

=42411.5ftpounds/min

In terms of Horsepower,
1 HP = 33,000ftpound/min
Power = 1.28 Hp
In terms of wattage ratings,
1Hp = 0.7457 kW
Hence 1.28 Hp = (1.28×0.7457) kW = 0.96 kW
Assume a service factor of 1.5 thus, (0.96×1.5) kW = 1.43 kW
Therefore, a 1.5 kW AC Motor was selected for this design.
The primary power source for the equipment is a 1.5 Hp electric motor. This supplies the power required and the necessary speed and torque required to operate the microsatellite platform. The main components of the equipment are as follows: clamp, table top, plunger stand, rotatable wheels, roller bearings, threaded shaft, spur gears, worm gear and electric motor.

2.3 Materials

For the adequate performance of this equipment, several materials were considered based on some factors and adequate tradeoff analysis was done to prune down the list of available material for the design. Considering the nature of the equipment and the purpose it was designed for, the metal family group was deemed fit for the design of the platform. Under the metal family, however, it became imperative to consider the best material based on the following attributes: Density, Yield strength, Modulus of elasticity and finally desired weight of final designed equipment (Joseph et al.,2001). With these in mind, the steel material family was then chosen because it met practically all these attributes satisfactorily. Specific materials used were: Mild steel for the majority of the structural members, Cast Iron for the plunger stand, iron and mild steel for bolts and nuts; butyl rubber for the wheels. The machine part list and the selected materials are shown in Table 1.

3. RESULTS

3.1 Construction process

Fabrication which included the cutting to size of materials and permanent joining of structural parts was carried out. Figure 4, Plates 1 show different views of the fabricated adaptable microsatellite platform. The mainframe was fabricated from standard length angle iron of dimensions 1200 x 1040 x 940 mm. Following the design specifications, the angle iron was cut into appropriate sizes and welded together to form the main structure on which other parts were welded accordingly to form the equipment.

Figure 4: Plate 1. The assembled platform

4. Performance Evaluation

Performance tests were carried out on the fabricated platform on two significant fronts viz: Structural integrity test and adaptability of the two standard launch vehicle adapters to the designed clamp. The standard launch vehicle adapter is the 8in (203 mm) and 15in (381 mm) standard launch vehicle adapters for any microsatellite (Malik, 2014). The suitability test, the designed slot was fabricated to be able to accommodate both launch vehicle adapters suitably. The structural integrity test, a hollow dummy microsatellite was fabricated with dimension 600 mm x 600 mm x 600 mm and carefully loaded with dead weights until the various weights were achieved and the equipment tested accordingly. Six different weight values were considered for this test. The weight values were chosen as replicas of actual microsatellites launched between 2003 and 2011 as in Table 1.

Table 1. Showing microsatellite comparison

SPACECRAFT	COUNTRY	DIMENSION (mm)	LAUNCH WEIGHT (kg)	LAUNCH YEAR
NigeriaSAT-1	Nigeria	600 x 600 x 600	98	2003
BilSAT-1	Turkey	700 x 700 x 700	129	2003
AlSAT-1	Algeria	600 x 600 x 600	88	2003
DEIMOS-1	Spain		91	2011

UK-DMC 2	UK	120	2011

Together with NigSAT-1 launched in 2003, BilSAT-1 and AlSAT-1 were also launched as part of the global Disaster Monitoring Constellation (DMC) Satellites all launched from Russia (Adetoro and Aro, 2007). Increasing microsatellite load versus execution time. Execution time, i.e. time it took from the start of the horizontal position of the satellite bed to 45° angle of the bed of each operation was recorded accordingly as each weight was tested and the time recorded accordingly. Table 2 shows the results obtained for both test cases.

The results obtained in Table 2, The simulated test case gave a better load- time response compared to the workshop test case. The reason for this variation lies in a number of factors which include but not limited to; the variation in some of the materials used for the fabrication as compared to the CAD model, some fabrication detailing which could be different to the CAD model due to the non-availability of some parts on shelf and finally could be due to friction and loss of energy within the system which is avoided during the simulation phase. . The graph of the microsatellite loads and execution time for both cases of workshop tests and simulation result using Solid works motion study was plotted as shown in Figure 4.

Table 2. Load and estimated time

LOAD (kg)	SIMULATED TIME (seconds)	WORKSHOP TIME (seconds)
88	35	40
91	40	45
98	41	46
120	44	51
129	51	58
150	62	69

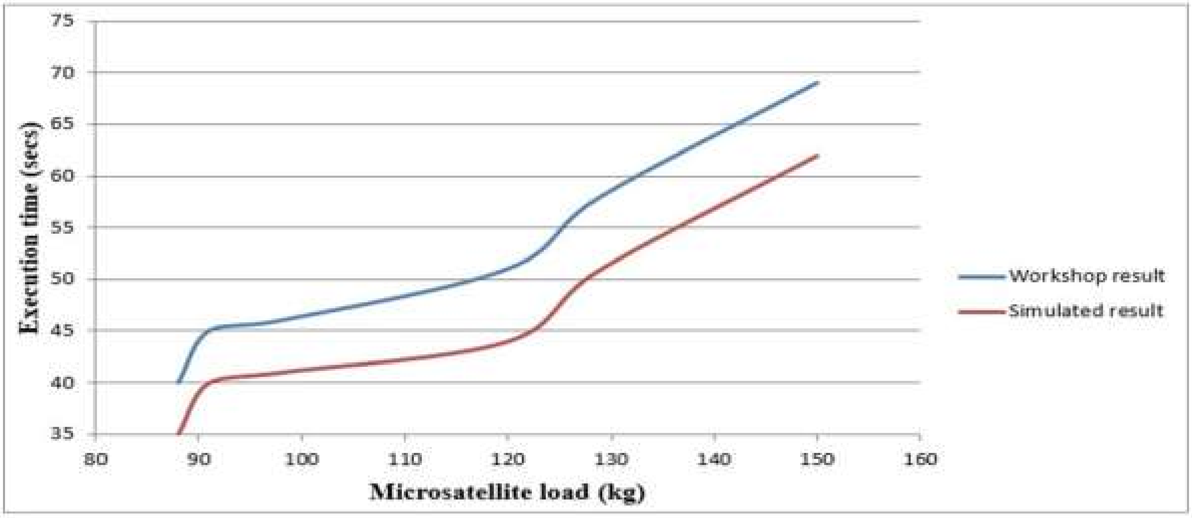


Figure 5: Increasing Load Vs Execution time

5. CONCLUSION

The design and fabrication of the adaptable platform for a microsatellite production have been completed and its objective fully achieved. The structural integrity evaluation indicated acceptable results from both the test cases and structural integrity evaluation. Results from the simulated and workshop loading test gave a machine efficiency of about 88%. The maximum stress values obtained for the critical loading case of 150 kg for some of the critical structural supports like the clamp was 3.07×10^{7} N/m^{2} and for the plunger, the stand was 1.59×10^{7} N/m^{2} which are both well below the yield stress for ASTM A36 steel, i.e. 2.5×10^{7} N/m^{2}. Thus, indicating that the design is safe for operation.

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