The Design, Construction and Computer –Aided Simulation   of a Prototype Roofing Tile Machine

Okoronkwo C.A[1], Ezurike O.B[2], Igbokwe J.O[3]   and Oguoma O.N[4]

1,3,4, Department Of Mechanical Engineering, Federal University Of Technology, Owerri
P.M.B 1526 Owerri, Imo State Nigeria
2 Department Of Mechanical Engineering, Madonna University Enugu Campus

---***---

Abstract –

The design, analysis and fabrication of a roof tile production machine are presented. The machine with dimensions 250.2×103.1×50 cm, is consists of the conveyor system, hopper, roller, frame, cam-actuated cutter system, frame, 2 dc motors etc. Calculations involving various physical properties of the tile, pallet, etc. were carried out. Furthermore, analysis was done on component parts like camshaft, shafts, cutter, frame, pallet etc. to determine the effect of loads and varying boundary conditions on their mechanical properties hence deciding the most convenient working condition for which the machine must run in terms of power requirement, speed, components loading etc. Results obtained from the analysis, shows that the conveyor system speed is 0.164 m/s, the camshaft rotates at 117.95 rpm, the cutter has a maximum displacement of 16.062 mm. To further validate the paper analysis, a Computer Aided Analysis was used to test component parts thus eliminating some errors.

Keywords: Roofing Tiles, Prototype, Extrusion Chamber, Roller, Cutter

1. INTRODUCTION

One of the greatest challenges to architects, planners and designers must be to consider not only how the building will look in the immediate future, but also how it will look in 10, 20, 30, 40, 50 or 100 years' time. This is the greatest legacy that they can bequeath to future generations- an architecture that is as beautiful, if not more so, in decades to come as it was when first constructed.

Roof tiles are construction materials like slates used for roof covers. The tiles can be corrugated both for architectural beauty and working convenience. Much attention has been paid to developing the small-scale production of concrete roofing tiles as an affordable alternative to both traditional roofing materials, such as thatch, and modern, mass-produced, often inappropriate, galvanized iron sheeting or asbestos cement. These tiles are relatively low in cost, durable, aesthetically acceptable, able to offer adequate security and comfort, and provide protection from both the heavy rain and the hot sun. **[1].** besides their architectural beauty, roofing tiles of clay and concrete are useful in regulating room-temperature. It is a common observation that roofing tiles are recently becoming popular in modern housing constructions. Moreover, the overall economic growth in the country and climate change due to global warming are expected to create growing demand for the product. Concrete roof tiles are an outstanding example of a high quality, cost-effective solution for roofing. They have proved their worth over many years of trouble-free use, providing maximum protection against the elements. Concrete roof tiles are manufactured in an extensive range of profiles, colours and finishes which enhance the visual appearance of any roof and provide designers with a wide scope for expression. Tiles are made up of Sand, cement and water – combined with a rational production method. The sand, with a max particle size of 4mm, is mixed with standard Portland cement. The sand/cement mix proportion should be about 1:3.5 **[2].** Pigment is usually mixed into the concrete to give the tiles an even colour. The tiles can be of different kinds of profiles. The profile to be used for this design is the flat profile.

The roof is arguably the most exposed face of any structure. It has to withstand rain, ice, ultraviolet light and the effects of damaging acids caused by the atmosphere pollution. The asbestos, zinc and aluminium roofing sheets have been very prone to failure during harsh conditions. They offer little resistance to the damage caused by hurricanes, earthquakes and other weather extremes. They also offer conductance to heat, meaning that they make the homes colder in the winter and warmer in the summer. The cost of maintenance of these roof sheets is also very high. Roofing tiles are more preferred because

they are weather resistant, fireproof, carry a class A rating as insulators and help to keep homes warmer in the winter and cooler in the summer. However, the cost of producing these tiles is very high and therefore beyond the reach of so many people. The roofing tile machines in the market are very complicated and require highly skilled labour for their operation. As a result of the above, the authors aim to provide a cheaper and more economical way of manufacturing concrete roof tiles machine using locally made materials, parts and components, a roof tile machine that will require a lower technical-know-how to service and maintain.

2.0 SYSTEM CONFIGURATION

The system designed is shown in figure 1. This system has the same working principle as the one manufactured by hydraform tile making machine designed shown in fig 2 The system is consists of the conveyor system, hopper, roller, frame, cam-actuated cutter system, frame, 2 DC motors etc. it has dimension of 250.2×103.1×50 cm. The mode of operation involves many stages of operation. The first stage uses an automatic cement mixing machine which is situated away from the machine. The mortar is fed to the machine or with the use of conveyors the length of which depends on the situation of the mixing chamber. The next stage of the process is the first stage of the concrete roof tile machine – the compaction/pressure forming chamber. This is usually achieved via vibrating mechanisms. The size of this chamber is the width and length of the roof tile. Under this chamber is a conveyor system having pallets which carry the mix across all the chambers. The pellets are designed to be the same length and width of the roof tiles. The top profile also has the same profile as the underside of the roof tiles.
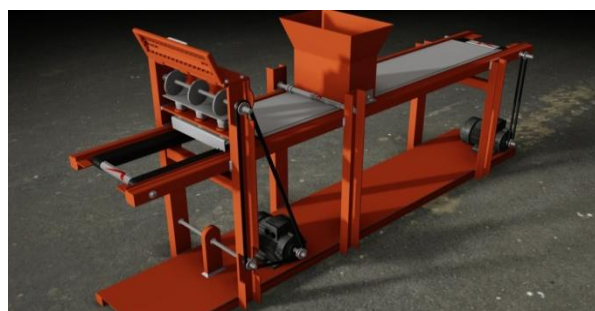

Fig. 1 the prototype of the roofing tiles making machine designed

 Fig. 2.0   Roofing tile making machine by hydraform[1]

The extrusion chamber has an opening the size of the thickness of the concrete roof tiles. The profile of the bottom of the extrusion nozzle is the same as the profile of the top of the roof tile. The extrusion gives the final compaction on the tiles. The conveyor moves the extruded mix on the pallets out of the extruded chamber.

The cutting system utilizes either a pneumatic actuator or a hydraulic actuator. This brings about high pressures enough to initiate the cutting action of the cutter. The conveyor system stops momentarily during this cutting action. The choice of motor is usually stepping motors that will be integrated with either the hydraulic or pneumatic system to interface with a microcontroller so that the start/stop action of the motor will coincide with the opening and closing of the valves of the actuator. Also the time between successive starts/stops will give an axial length equal to the length of the tile on the pallet.

After the cutting of the tiles, the concrete roof tile machine plays no further part in the process. Other conveyor systems can then be attached to take the tiles through needed chambers like paint spraying and sealing. From there the tiles which are still on the pallets are removed and sent to the curing chamber where they are dried finally to give it its required strength.

3.0 DESIGN OF THE MECHANICAL ELEMENTS

The knowledge of materials and their properties is of great significance in the design and analysis of machines and machine members, **[3].** The machine elements should be made of materials which has properties suitable for the conditions the machine will operate. In the design of the automated roof tile production machine, the following

important characteristics were carefully considered in selecting the materials (elements): Strength and durability of the materials, Flexibility and mass of the materials, Ability to withstand both varying load and high impact load, Corrosion resistance and ability to be cast, welded or hardened, Machinability and electrical conductivity of the material.

Above all, the most difficult challenges for the design of an automated roof tile production machine are how to select the best materials which would serve the desired objective at the minimum cost. Hence, the following factors were strictly considered when selecting the materials: Availability of the materials, Suitability of the materials for the working conditions in service, the cost involved in purchasing the materials.

Khurmi and Gupta [3] stated that the best materials are the ones which serve the desired objective at the minimum cost. The widely used materials in the design of mechanical members are steel an alloy of carbon and iron having carbon content up to a maximum of 1.5%.

3.1 Design Calculations and Analysis

The design calculations in this section are to provide the right sequence, geometry and form in which the interaction of forces is correlated to achieve a workable system. Some of the components designed in this section include: the conveyor, shaft belt and cam system etc.

3.1.1 The Conveyor

The conveyor system to be adopted in this design is the belt conveyor. The purpose of the conveyor is to move the pallet across the barrel (where there is a mass movement of mortar), vibrator (first compaction of mortar), roller (final compaction of mortar) and cutter (where the compacted mould is cut). The conveyor is primarily made of pulleys and flat belts.

3.1.2 Motional Resistances and Power required in the Steady Operating State of the conveyor

Phoenix [4] stated that to overcome the motional resistances in a conveyor belt system the required (mechanical) power is determined by:

$$Pmax = Fwmax * V \qquad\qquad 1$$

P_{max} = Total as a result of loading conditions in a steady operating state of necessary power at the periphery of the driving pulley

F_{wmax} = Total of the motional resistances in forward run/return run in a steady state operation + total acceleration force at start up

V = Belt speed

Also,

$$PMerf = (PWmax/ \eta ges)$$

P_{Merf} = total power required of the drive motors

P_{Wmax} = maximum power required at the periphery of the drive pulley(s)

η_{ges} = the overall efficiency of all transmission elements between motor and pulley shaft.

For the belts used in this design, the efficiency is taken to be 96% **[5]**

3.1.3 Motional Resistances, F_v

With the belt moving in a steady operating state, motional resistances arise from friction, weight and mass forces:

$$Fv = Fh + Fn \qquad\qquad 2$$

F_h = total primary – acting in top run/return run resistances along conveying flight

F_n = total secondary – locally limited to the head resistances and tail of the system.

3.1.4 Primary Resistances, F_h, of the Conveying Flight

The primary resistances of the conveying flight are composed of the parts occurring in the sections **Phoenix, [4].** These consist of flexing resistances of the conveyor belt as well as the bulk material and the rolling resistances of the idlers. The flexing resistance of the belt arises mainly from its indentation rolling resistance; its bending resistance is of secondary minor importance.

The main resistances of the system sections are simplified by using a linear dependency of the moving mass – split up for top run and return run – and are determined as follows:

$$Fhi = \mu i * li * g * [m'G + m'Li] \ldots\ldots\ldots 3$$

μ_i = coefficient of friction factor of a system section

l_i = belt length of a system section of the entire conveying length

g = acceleration due to gravity (g = 9.81 m/s2)

m'_G = length related mass of conveyor belt

m'_{Li} = length related mass of the conveyor belt of a section of the system the loaded run with an evenly distributed load on the conveying track.

3.1.5 Secondary Resistances, Fn, of Individual Conveying Sections

The total secondary resistances F_n result from the sum of locally limited motional resistances in the top run and return run, particularly at the head and tail of a belt conveyor system **Phoenix, [4]:**

The equation for calculating the secondary resistances in the different sections is given as follows –

$$F_{Ni} = \{\mu g(l)(m_{pi}|m_{ti})\}. \qquad 4$$

Where

μ = coefficient of friction between steel and mortar = 0.31

g = acceleration due to gravity = 9.81m/s^2

3.1.6 Acceleration Resistances, F_A at Start-up

This resistance is made up of the frictional resistances between the transmission systems, i.e. those between belts and pulleys. The equation for this is given as

F_A = A_{cc}* μ*Length related mass of belt in contact with pulley

This acceleration is so small that the Acceleration Resistance can be disregarded.

Therefore,

F_{wmax} = F_H + F_N............   5

3.1.7 Speed of Conveyor

Here an arbitrary time for between two cuts is assumed. This is the time it takes one pallet be cut. Therefore, the conveyor moves a distance equal to the length of the pallet in x second.

The velocity, v of the conveyor is thus given as:

$$v = \frac{0.42m}{x} \qquad 6$$

3.2 Motor Drive

Due to the low speed and subsequent low torque encountered in this design, the motor is chosen as the prime mover.

Diameter of motor shaft = 9.2 mm

Diameter of motor pulley = 72.7 mm

The RPM required of the motor based on the design so far is calculated as follows –

$$v = \frac{\pi DN}{60} \qquad 7$$

Where

V = velocity of the motor = 0.1640624m/s

π = 22/7

D = diameter of the motor shaft = 9.2*10^{-3}

N = RPM of the motor

From the equation 7, the speed is obtained as:

$$N = \frac{60v}{\pi D} \qquad 8$$

Substituting the values into the equation,

N = 340.5827209 RPM

The torque transmitted by the motor is gotten as follows –

$$T = \frac{30P}{\pi N} \text{ or } T = FR$$

Where

T = torque transmitted

P = P$_{Merf}$ =

N = RPM of the motor

π = 22/7

F = Force at the periphery of the drive

R= Radius of motor shaft

Solving for T gives us

T = 1.402 NM

3.3 Belts and Belt Tension

The v-belt is chosen here to transmit power from the motor to the conveyor shaft.

The tensions in the tight and slack sides of the belt are calculated as follows:

$$T = (T_1 - T_2)R \qquad 9$$

Where

T = torque transmitted by the motor =

T$_1$ = Tension in tight side of belt

T$_2$ = Tension in tight side of belt

R = Radius of motor pulley = 4.6*10^{-3} m

Also, the ratio of driving tensions for v-belts is given as

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu\theta cosec\beta \qquad 10$$

Where

μ = coefficient of friction between belt and pulley = 0.26

θ = angle of contact in radian between belt and pulley =

β = ½ of Groove angle of belt = 17^0 for 'A' belts **Khurmi and Gupta, [6]**

The difference in (T$_1$ – T$_2$) and F$_{max}$ is due to power loss in transmission.

The length of the belt is calculated as follows

$$L = (center\ to\ center\ distance, C\ between\ pulleys) + (2\pi R) \qquad 11$$

C = 0.557 m

R = radius of pulley (using the same pulley size) = 0.0727 m

π = 22/7

Therefore, L = 1.014 m

3.4 Shafts

The design of this shaft is based on the Maximum Shear Stress Theory as the shaft is subjected to combined twisting moment and bending moment.**[5]**

From the theory,

$$\frac{\pi}{16} \times \tau_{max} \times D^3 = \sqrt{M^2 + T^2} \qquad 12$$

Where

π = 22/7

τ_{max} = the maximum shear stress in the shaft

D = diameter of the shaft

M = Maximum bending Moment

T = Maximum torque transmitted

The left hand side of the equation is known as the Equivalent Twisting Moment

3.5 The Cam System

The design of the cam system is based on a circular arc cam with flat faced follower. 3 identical cams will be used with each equally laid out on the follower. Under the follower are three compression returning springs.

3.5.1 Cam Motion

For the cam motion, the case where the flat face of the follower has contact with the circular flank was adopted. **Khurmi and Gupta[6]** proved that in deriving the

expressions for displacement, velocity and acceleration of the follower for the above case, it is assumed that the cam is fixed and the follower rotates in the opposite sense to that of the cam.

From the geometry of the figure, the displacement of the follower, x at any instant for contact with the circular flank is given by

$$X = BA \qquad 13$$

By the use of trigonometry and geometry,

$$X = (R - R\_1)(1 - cos\theta) \qquad 14$$

Differentiating the above equation with respect to t, gives the velocity of the follower

$$V = \frac{dx}{dt} = \frac{dx}{d\theta} \times \frac{d\theta}{dt} = \frac{dx}{d\theta} \times \omega \qquad 15$$

Where $\frac{d\theta}{dt} = \omega$ = angular speed of the cam shaft

Therefore,

$$V = \omega(R - R_1)sin\theta \qquad 16$$

From the equation, it is observed that the velocity increases from 0 at the beginning of ascent to a maximum value θmax .This is when the contact of the follower just shifts from circular flank to circular nose.

$$V_{max} = \omega(R - R_1)sin\varphi \qquad 17$$

Differentiating the above equation with respect to t, the acceleration of the follower is obtained as:

$$A = \omega^2(R - R_1)cos\theta \qquad 18$$

$$A_{max} = \omega^2(R - R_1) \qquad 19$$

3.6 Cam Profile

In order to get the required dimensions, a first reasonable assumption will have to be made. Let the total angle of action on cam be 150^0, i.e. $2\square = 150^0$. Also, let the cam profile be split into two parts.

Part one is made up of the profile still having the circular arc of the base circle. There is no displacement of the

follower during this period. Let this arc be completed in (2.56-1) sec of the axial motion of the tile along the conveyor.

The axial distance, L = length of arc subtended by the total angle of the cam.

$$L = \frac{\theta}{360} \times 2\pi R \qquad 20$$

Where

R = radius of circular base

Using θ = (360 – 150) = 210^0

π = 22/7

L = 3.665191429R

This distance is to be moved in 1.56 seconds

The velocity, V of the cam is therefore

$$V_c = \frac{3.665191429R}{1.56}$$

R = $0.42562579V_c$

It is assumed that the cam covers an axial distance of 0.42m for one revolution in 4 seconds, therefore

$$V_c = \frac{0.42}{4} = 0.105 \; m/s$$

Putting this as the value of V_c in the previous equation, we get

R = 0.04469 m = R_1

The Part two is consists of the circular arc connecting the base circle and the nose circle together with the nose. It is at the start of this part that displacement commences on the follower. Due to parameters already established, any further assumption of parameters must satisfy all sets of conditions as stated below

- The total lift must be greater than the depth of the tile to be cut, i.e. $X > 11 \; mm$

- The forward displacement and the return run of the follower must be completed in the remaining (2.56-1.56) seconds of the conveying time of one tile.

The implication of the first condition is that – R_2 + OQ – R_1 > 11 mm with 11 mm being the intended thickness of the tile to be cut. Having gotten R_1, the above becomes – R_2 + OQ > 55.69 mm

The cam system is best designed first and fine-tuned before the pallet is designed. This is so because in the design of the cam system, there are assumptions which are made that might not be in sync with the geometry, especially te length of the pallet and subsequently tile. Trial and error is followed until the intended length of cut is gotten. Usually this is done by a using a paper cut profile of the cam and the profile that gives the best result is then used to produce the actual cam.

Also out of the 1 second allocated to the displacement, it takes the follower 0.5 seconds to go from the start of displacement to the maximum lift. This means that –

(The time taken from start of displacement to the nose contact) + (The time taken from nose contact to the nose peak) = 0.5 seconds.

Using the average velocity of the follower during these intervals, the above can be written as –

$$\left[\frac{\{2(R_1+OQ-R_1)-2([R-R_1][1-cos\varphi])\}}{\{\omega(R-R_1)sin\varphi\}}\right] +$$
$$\left[\frac{\{2([R-R_1][1-cos\varphi])\}}{\{\omega(R-R_1)sin\varphi\}}\right]$$

21

Also using a cam shaft diameter of 17mm, ω can be calculated as follows –

$$\omega = \frac{2V}{D} = 2 \times \frac{0.105}{0.017} = 12.35294117647059 \, rad/s$$

It is assumed that the cutter is to have a clearance of 20mm and that the cutter starts cutting the tile when the cam follower just makes contact with the nose of the cam, then

$$X_1 = (R - R_1)(1 - cos\varphi) = 20 \, mm$$

Using the above equation to substitute for $(R - R_1)$ in the previous equation and solving for φ, we get $\varphi = 61.121686^0$. From there, we solve for R and get the value to be 83.374321mm. The values of R_2 and OQ can then be chosen to meet all the conditions stipulated.

Note also that this process of cam profile parameter determination is an iterative one. The loop will go back to where the velocity of the cam was chosen, but instead of generating a new base radius, a new total angle of action of cam is gotten from the equation –

$$L = \frac{\theta}{360} \times 2\pi R = V_i T$$ (Where V_i is any assumed velocity and T is the time which is allocated to the base circle) and the iteration continues.

3.7 Power Considerations

The prime mover i.e. electric motor which is driving the cam system must be such as to overcome the weight of the cam, the frictional resistance between the cam and follower and the stiffness of the springs used. As the velocity of the cam has been assumed, the RPM of the motor can then be calculated. Thus using V = 0.105m/s and Diameter of motor shaft = 0.0092m, the RPM can be calculated as follows –

$$N = \frac{60V}{\pi D} = \frac{60 \times 0.105}{\pi \times 0.0092} = 217.9448149890682 RPM$$

The first condition for power transmission stated above can be expressed mathematically as follows –

$$F_T > KE_i + M_i G + \left[\frac{\mu M_i A1}{R} + \frac{\mu M_i A2}{R2}\right]$$ 22

Where R_2

F_T = Force needed to drive the cam system

K = Stiffness of spring, assuming steel alloy is used as spring, K = 5.144 N/mm

E_i = Extension = 16.2 mm

M_i = Mass of cams = 0.49*3 = 1.47 kg

G = Acceleration due to gravity = 9.81m/s^2

A1 = Acceleration of the follower at start of displacement = 5.903

A2 = Acceleration of follower at beginning of contact with nose of cam =

μ = Coefficient of rolling friction between cam and follower = 0.005

F_T = 267.015 N

The force needed by the motor to drive the entire system, including the transmission system is given as:

$$F_S = F_T + F_M \qquad 23$$

Where –

F_M = Motional resistances in transmission system

The motional resistances between belt and pulley are again of negligible value here as the acceleration of the cam system is very small.

From the above, the power needed to drive the system is given by

$$P_D = F_S \times V \qquad 24$$

Where –

V is the velocity of the motor.

P_D = 28.04 W

As always, there is always power loss in transmission systems, thus the actual power supplied by the motor is usually more than that calculated. The minimum power supplied by the motor is thus given as –

$$P_M = \frac{P_D}{\beth_{ges}} \qquad 25$$

Where $\beth_{ges}$ = Overall efficiency of all transmission elements between motor and pulley of cam shaft = 95%. Thus the power requirement of the motor for the cam system has been taken care of. P_M = 30 W (approximately).The motors used are 24 volts DC motors of 1hp. It is run with a 27 volts/550mA adaptor.

4.0 RESULTS AND DISCUSSION

The section shows the results of the computer aided analysis carried on the system. The fig 3 shows the effect of the applied load and boundary conditions on the beams when mild steel is used. Note: results may vary with other materials.
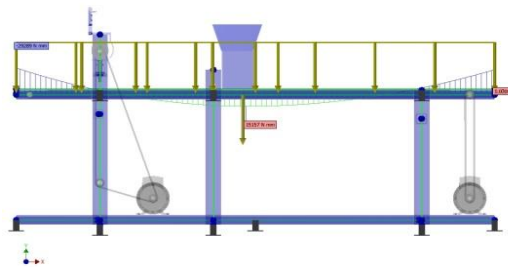

Fig: 3 bending moment diagram with and without extended belt support

The figure 3 shows the bending moment on the beams carrying the conveyor belt thus, invariably carry the loaded and unloaded pallet. For this analysis, the beam is extended using angle irons of dimension 30 × 60 × 3 mm to completely support the belt ,but from the analysis, it can be noted that this extension allows for excessive bending moment of the beam.
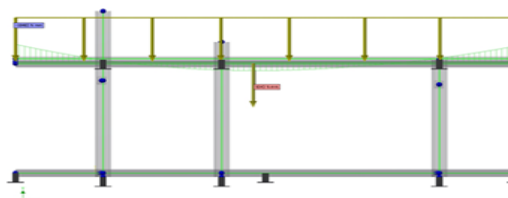

Fig:4 Bending moment diagram without extended belt support.

The figure 4 shows the bending moment diagram of the beam without an extended belt support. Though, this design allows for a minimum bending moment, it does not provide full support for the conveyor belt. This may lead to excessive slacking of the conveyor belt, which can be avoided by using idlers which would incur more cost. Hence, the selected beam is a compromise of conveyor belt support and bending moment requirements. But, since the belt must be adequately supported, the extended beam support is used and is reinforced using ribs.
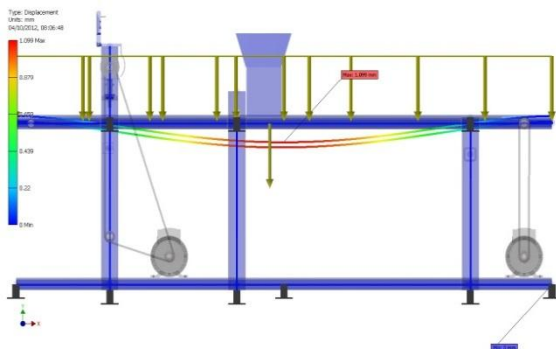
Fig: 5 maximum displacement without extended support

In agreement with the earlier result (bending moment graph), the maximum displacement for the beam without extended support is shown in fig 5.

The deflected beam as shown is not displayed to scale (i.e. the scale is not in correspondence with the values. It is magnified for clarity purposes).
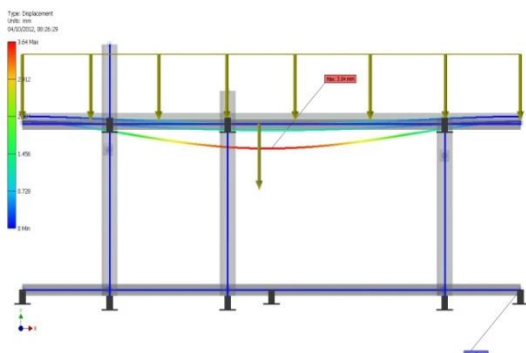

Fig: 6 maximum displacement with extended support.

From the displacement diagram above it is also seen that the extended support is subjected to more displacement than the beam support without extension.

5.0 CONCLUSION

The design, analysis and fabrication of a roof tile production machine have been successfully completed. The motivating factor in the present design is to reduce to the barest minimum the cost of producing a roof tile making machine. Though the cost of production and procurement of component parts was not included in this paper, it is believed that interested designers can procure the component parts from their local market without importing them. This is to stimulate the mass production of this machine in the third world countries for the betterment of the living condition of the populace. The

machine designed has dimensions 250.2×103.1×50 cm, is consists of the conveyor system, hopper, roller, frame, cam-actuated cutter system, frame, 2 dc motors etc. Calculations involving various physical properties of the tile, pallet, etc. were carried out. Furthermore, analysis was done on component parts like camshaft, shafts, cutter, frame, pallet etc. to determine the effect of loads and varying boundary conditions on their mechanical properties hence deciding the most convenient working conditions for which the machine must run in terms of power requirement, speed, components loading etc. results obtained from the analysis, shows that the conveyor system speed is 0.164 m/s, the camshaft rotates at 117.95 rpm, the cutter has a maximum displacement of 16.062 mm.

6.0 REFERENCES

[1] J Orujekwe, C Ubah , Charles, Ujomu L , Okpara P And Egwu E(2010) Design Of Roofing Tile Making Machine *Final Project Federal University Of Technology,Owerri, Nigeria*
[2] A.B Abece. (2007) Your global partner in roof tile production', Nykoping: Bybloo
[3]R.S Khurmi, and J.K Gupta, (2005) Textbook of Machine Design, New Delhi: Euracia
[4] Phoenix (2004) Phoenix Conveyor Belts Design Fundamentals, Hamburg

[5]Budynas and Nisbeth (2006) Shigley's Mechanical Engineering Design, USA*: McGraw-Hill*
[6]R.S Khurmi, and J.K Gupta, (2010) Theory of Machines, New Delhi: *Euracia Publishing* Communication Publishing.
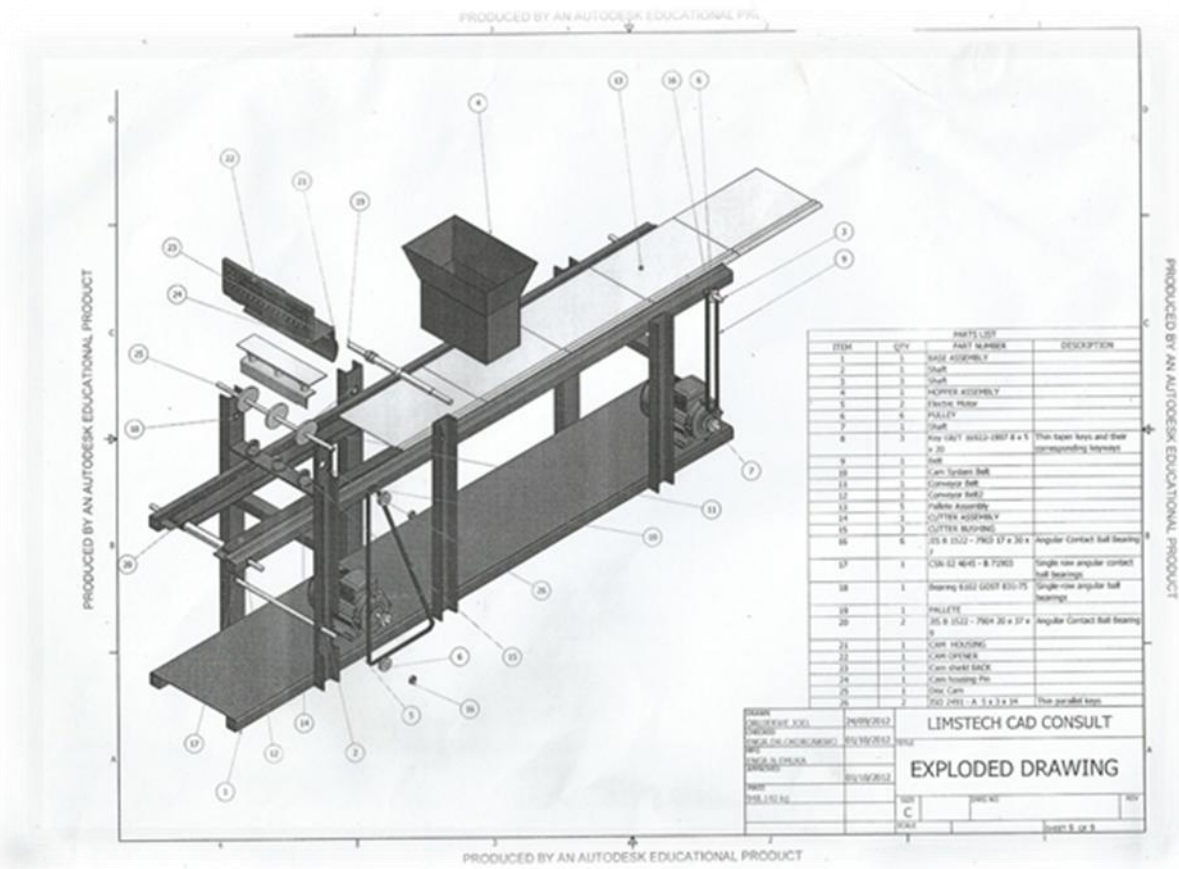   *House.*
    .

Appendix

FIGURE 7 THE EXPLODED VIEW OF THE ROOFING TILE MACHINE