DESIGN AND MODIFICATION OF BALE CUTTING MACHINE TO IMPROVE ITS PRODUCTIVITY

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Abstract

The aim of this paper is to present the results of the successful implementation of the redesigned bale cutting machine in Falcon tyre manufacturing company. Modification of hydraulic bale cutting machine is carried out to increase the productivity of the machine, to reduce the time and stress on the workers. The system is consisting of modified shear blade, suitable blade holder, hydraulic system according to new tonnage capacity and conveyor system. Unigraphics is used to model the blade and blade holder, meshing and analysis is carried out by using hypermesh and nastran. The parts are manufactured accordingly to new design parameters, and the following are installed and the performance characteristics to be determined are time, cost and productivity for this new modified design relative to the old system. It is established from the test results taken for a period of three months, that there is reduction in the time and cost of 70-80%, and 50-60% which in directly increased the productivity of machine and making the process economical.

Keywords: *shear blade, blade holder, hydraulic system, conveyor system*

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1. INTRODUCTION

A tyre is a ring shaped wrap that fits over the wheel's rim to secure it and to validate better vehicle performance". It is built upon a strong inner carcass with metal hoops embedded in it where it sits on the rim. Rubber is the basic raw material used in the production of tyres, the rubber in form of both natural and synthetic types are used along with some additives, tyres are manufactured according to standardized processes like mixing, extrusion, calandering, bead winding, bias cutting, tyre building ,raw cover painting, hauling, tyre curing, trimming, and inspection. Before being adding the rubber to the mixer it is needed to cut into small required sizes on a bale cutter machine. The bale cutting machine comprises of various parts shear blade, blade holder, main frame, table, arm, cylinder, and hydraulic system. The cutting of bale takes place when blade attached to the ram moves vertically downwards and shears the rubber placed on the table.

I. A. Daniyan, A. O. Adeodu and O. M, Dada from department of mechanical and mechatronics engineering worked on the design of conveyor system for the crushed lime stone using three roll idlers. They designed the material handling system in such a way that the process mist go cheaper providing supportive role to obtain good economical results, they worked on the system design which comprises the calculation for the values of roller diameter, belt length, belt thickness, belt inclination, spacing for idlers, mode of control, tension and power required, loading capacity of the conveyor system etc.

Shubham D Vaidya, Akash R samrutwar carried out their work on design of material handling system for an crushed biomass wood using v merge conveyor system, the project was comprising of designing and deriving the new values for various parameter in conveyor system as like belt diameter, belt thickness, belt inclination, and various all other parameters required to design the conveyor system was found out

Extractions from literature survey of conveyor system it shows that for designing of conveyor system various parameters needed to be finding out like belt length, inclination, belt capacity, roller diameter etc.

The objective of paper is to increase the productivity of bale cutting machine and to improve the easiness in material handling. In order to increase the productivity, the machinery is needed to modify and the necessary parts of it has to be redesigned, fabricating and installing those to the machine. As per the new required tonnage capacities to cut the maximum number of bales the supportive hydraulic system is needed to designed.

Designing of the blade involves finding out the new parameters rake angle, and shear angle. To hold the blade the suitable blade holder is needed to be designed with particular shape and dimensions, where as designing of hydraulic system is needed to select the suitable hydraulic equipments like hydraulic cylinder, control valves, pump, filters, type of oil used etc.

1.1 Specification and Productivity of Bale Cutting Machine (Before Modification)

The hydraulic machine in the industry was installed few years back it works on the hydraulic energy and the necessary hydraulic power is supplied by the hydraulic system, the rubber bale is cut when the ram moves down and the blade fixed to that pierces it, the loading and unloading is done manually and it is tedious and the productivity of machine is not up to the mark as it is having the capacity to cut only one number of rubber bale which is resulting in low production rates.

Fig -1: Bale cutting machine before modification

Table -1: specification of bale cutting machine

Make	Germany
Cutting stroke	500 mm
Diameter of cylinder	150 mm
Cylinder pressure	150 kg/sq cm
Blade width / stock thickness	400 mm / 150 mm
Time of cutting stroke	12 seconds
Motor power	15 hp
Number of bales cut per stroke	1 no's

The productivity of machine is too less the actual rubber required for the day plan is 120 tones and to cut the 120 tons of rubber the time required is 24.08 hours but after removing the unproductive time in a day(food ,shift change, intervals) is 2.75 hours, after removing the unproductive time from day schedule the working of machine is carried out for only 21.25 hours and the achieved cutting of rubber is only about 109.62 tons the remaining 10.38 tons on rubber is remained uncut.

2. DESIGN

The following are the parts identified for modification
1. Design of blade and blade holder.
2. Design of hydraulic system.
3. Design of conveyor system.

2.1 Design of Blade and Blade Holder

The design of blade involves finding out the new rake and shear angle of the blade.

Shearing is a mass separating process which involves cutting of various cross sections without formation of chips. In this process, a portion of work-piece is separated from the rest. The shearing process is used as a preliminary step in preparing stock for various other processes.

During shearing, the work piece, i.e. the rubber bale, is placed between the movable blade and the work table. As the blade is forced down, the rubber bale is penetrated to a specific portion of the thickness and starts to separate. Finally, as the rubber bale is completely penetrated, the bale is separated into two halves. This is caused due to the highly localized shear stresses that the material experiences.

The shearing force required to cut the rubber bale is as follows

$$Fs = \frac{S \times P \times T^2}{R} \times \left(1-\frac{P}{2}\right)$$

Where,
Fs = Shearing force (350×10^3 N)
S = Shear strength of rubber (50 MPa)
T = Stock thickness (250 mm)
R = Blade rake angle (degree)
P = Penetration of blade into material T (range: 0 to 1)

$R = 5.6^0$

Thus, the shear angle is $\beta = 75° + R/2$

$\beta = 77.8^0$

The total length of the blade is 1000 mm as it was the maximum possible dimension that could be given and its height is (T) 280 mm as it must be capable of cutting two rows of rubber bales (each of height 110 mm) stacked one over the other. The thickness of the blade is assumed to be 30 mm. Also, it is given taper of 1:14. This helps in easy retrieval of the blade during the return stroke.

The material of the blade selected is HCHC (high carbon high chromium). It is a cold working tool, provides good wear resistance and has moderate hardness. It is the most commonly used blade material. It is subjected to fracturing or chipping when shearing harder metals or heavier gauges as the material cut is rubber so the fracturing of blade is not a matter of worrying.

The blade is joined by five M12 bolts to the blade holder. The number of bolts and the type of bolt to be used is obtained as follows:
Density of HCC steel = 7.7×1000 kg/m^3
Blade Weight p_2 =48.972 kg = 480.42 N

From design data handbook, for an ordinary joint, initial bolt tension,

$$p_1 = 2805 * d \text{ in N}$$

Where,
d = bolt diameter (mm), assuming d = 12 mm
p_1 = 33660 N

Thus total load $F = p_1 + p_2$ in N

F = 34140.42 N

Tensile strength of bolt = 400 MPa

With a factor of safety of 4, allowable working strength is 100 MPa.

Therefore, we have 100 = F/ (N × A)

Where,
F = Load (N)
N = Number of bolts
A = Load cross sectional area in mm^2 (113.09)
N = 3.1

From data hand book the number of bolts selected is five

Thus, five number bolts have been proposed based on calculations of number of bolts. Based on the strength required, coarse grade M12 bolts have been proposed

To mount the blade on to the bale cutting machine, two blade holding devices were proposed and are as shown in the assembly of blade and blade holder figure below

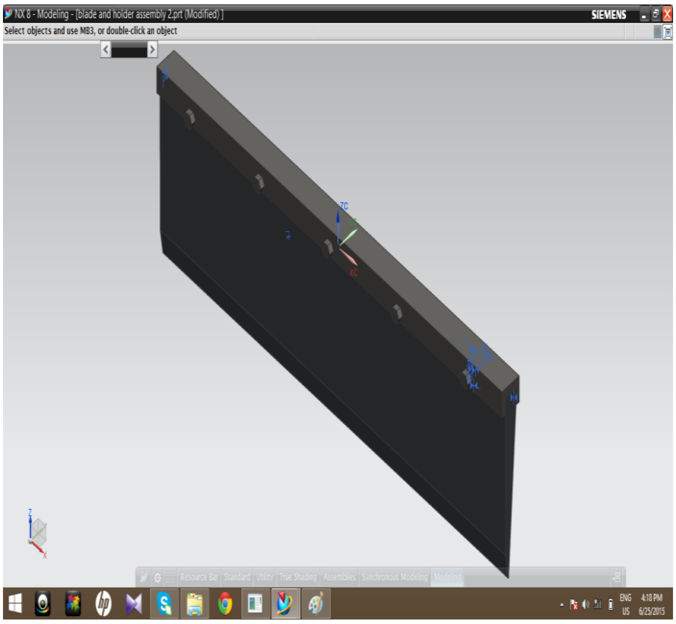

Fig -2: Blade and Blade holder assembly 1

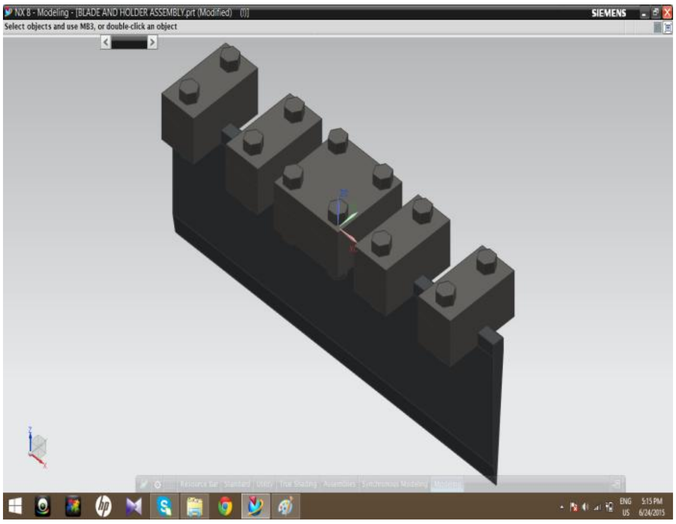

Fig -3: Blade and Blade holder assembly 2

Based on the pragmatics shown in table, design 2 was finalized. This design fulfilled the basic requirement, i.e. it constrains the blade movement in all directions. Also, the bolt life was highest in this design hence this leads to lesser machine down time.

Table -2: pragmatic analysis of proposed designs

Parameter	**Design 1**	**Design 2**
Fabrication ease	Easy	Difficult
Cost	Moderate	Moderate
DOF constrained	6	6
Bolt life	Minimum	Maximum
Material wastage	High	Moderate
Reliability	Low	High
Displacement variation	Minimum	Minimum
Stress variation	Minimum	Minimum

The raw material used for the fabrication of blade holder is forged Mild Steel with rectangular cross section. Mild Steel is one of the most commonly used metals and one of the least expensive steels used. It is weld-able, durable and hard, easily annealed. Various processes like gas cutting, lathe, shaping, drilling, tapping, grinding etc were carried out on the raw material to obtain the required dimensions.

The fabrication cost of blade and blade holder is rupees 47,160.

Fig -4: Bale cutting machine after modification

2.2 Design of Hydraulic System

Practical experiments were conducted on the existing bale cutting machine and it was determined that an average of 200 kg/cm² or 196.2 bars of pressure was required to shear rubber bales piled over each other.

Based on the pressure required to be developed to shear the rubber bales, from the catalogues of Bosch Rexroth, a standard cylinder was selected.

Standard cylinder: CD210 RE17017 (Tie rod type)
Nominal pressure = 210 bar
Piston diameter = 150 mm (range from 14-200 mm)
Piston rod diameter = 30 mm (range from 16-140 mm)
Maximum stroke length = 3000 mm
(We require a stroke length of 500 mm)
Maximum stroke speed= 0.5 m/s

Thus, the average cutting force from the cylinder required to shear the rubber bales piled over each other (Fc) in kN

F_c= Pressure required to shear × Piston area

Where,
Piston area = πr^2 in m^2
F_c = **346.714 KN**

Quantity of cylinder oil required for a stroke of 500 mm
= Cylinder effective area × cylinder stroke in m^3

Where,

Cylinder effective area = πr^2 in m^2 (diameter of cylinder 150mm)

Quantity of cylinder oil required = **8.835 × 10^{-3} m³** or **liter**
The type of oil used in hydraulic cylinder is ISO 68 (SAE 20W), density 865 kg/m^3

From the manual of Bosch Rexroth, a suitable variable displacement pump is selected which would efficiently aid in producing required pressure without any hesitation.

Type: A10VSO (Variable displacement pump)
Size: 10
Nominal pressure: 250 bar
Peak pressure: 315 ba2r
Displacement: 10 cm³
Speed: 3600 RPM
Flow@ max speed: 37.8 liter/min
Power@ 280 bar= 15.7 KW
Maximum toque= 41.7 Nm

A standard motor was chosen along with this pump from the Bosch Rexroth manual. This motor provides a peak power of 15.7 KW (21 Hp).

$$\text{Counter balance pressure} = \frac{\text{Net pressure acting on ram} \times g}{\text{Effective area of the piston}}$$

Where,
Net pressure acting on ram in kg/cm^3
g = acceleration due to gravity in m/sec^2
Effective area of piston = 0.0054 m^2

Counter balance pressure = 1.16 bar

Since, the acting force on the ram is enormous, a factor of safety of 10 is considered for the counter balance pressure. Thus, a counter balance pressure of 12bar is applied.

The proposed hydraulic circuit is shown in the figure 4.5. The circuit consists of various hydraulic components, namely, double acting hydraulic cylinder, counter balance valve (counter balance pressure= 12 bar), fixed displacement pump, 4/3 manually actuated spring centered open centered valve, filter, check valve and pressure relief valve (relief pressure= 200 bar).

The cost of hydraulic equipments and installation is rupees 69,180.

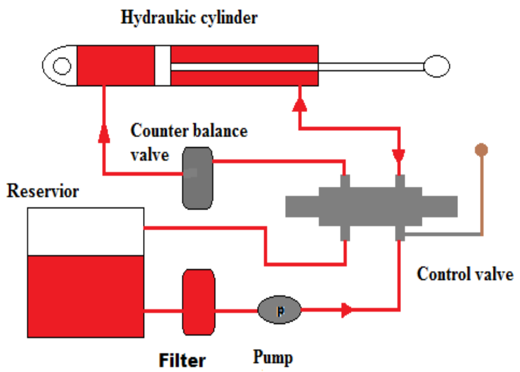

Fig -5: Designed hydraulic circuit

2.3 Design of conveyor system

The belt conveyor system is found to be well suited for the available floor space. The design of belt conveyor system involves determination of the correct dimension of the belt conveyor components and other critical parameter values so as to ensure optimum efficiency during loading and unloading conditions. Some of the components are: Conveyor belt, motor, pulley and idlers, rollers, pneumatic cylinder etc.

Length of the belt, $L = H \times N \times \pi$ in m

Where,
Vertical height of conveyor (H) = 350mm = 0.35m
Number of wayfers of belt (N) = 2
L = 2.19m

Inclination, $\tan\beta = \frac{\text{Height of conveyor}}{\text{Distance between rollers}}$ in degrees

Where distance between two rollersis 2000 mm

$\beta = 9.92^0 = 10^0$

Belt Capacity, c =Load bearing area of belt in $m^2 \times v \times \rho$

Where,
Belt Speed (v) = 0.5 m/s (desired speed)
Material Density (ρ) = 930kg/m^3
Load bearing area of belt = (length × breadth) in m^2
length of belt = 2.19 m
breadth of belt = 0.9 m
Load bearing area of belt = 1.971 m^2
Belt Capacity, c = 91.6515 kg/s = 330 x10^3 kg/hr

Belt Width, $B \geq Xa + 200$

Where,
X: Longest diagonal of irregular lump, (300 mm)
a: maximum linear size of the representative lump (2.5)
$B \geq 1150$mm
Belt width is selected to be 1200mm for our design.

Table -3: Diameter of roller

Classification	Diameter (Inch)	Belt Width (Inch)	Description
B4	4	18 – 48	Light duty
B5	5	18 – 48	Light Duty
C4	4	18 – 60	Medium duty
C5	5	18 – 60	Medium duty
C6	6	24 – 60	Medium duty
D5	5	24 – 72	Medium duty
D6	6	24 – 72	Medium duty
E6	6	36 – 96	Heavy duty
E7	7	36 – 96	Heavy duty

From design data hand book based on belt width the diameter of roller is selected to be of C6 classification.

i.e., D_{roller} = 152.4mm (6 inch)

Total Belt length = 2×L = 4.38 m

Power required by conveyor to produce lift, $P_L = \frac{c \times H \times 9.81}{3600 \times 1000}$

P_L= 0.315kW = 0.32 kW
W = weight of material + weight of belt, kg/m

where,
Weight of material, Wm = 77 kg/m
Weight of belt = 10 kg/m
W = 87 kg/m
Return side friction = $4F_e \times W \times L \times 9.81 \times 10^{-3}$
Where,
Equipment friction factor (Fe) = 0.0225
Return side friction = 0.168 N
Effective tension, T_e= Total empty friction + Load friction + Load slope tension
Total empty friction = $F_e \times (L + t_f) \times W \times 9.81 \times 10^{-3}$
Load friction = $F_e \times (L + t_f) \times (c/3600 \times v) \times 9.81 \times 10^{-3}$
Load Slope tension = $(C \times H \times 9.81 \times 10^{-3}) / 3600 \times v$
Where,
Terminal frictional co-efficient (t_f) $_=$60

Total empty friction=1.19 kN
Load friction = 2.52 kN
Load Slope tension = 0.63 kN
Effective tension, T_e= 4.34 kN
Belt tension at Steady State

The belt of the conveyor system always undergo tensile load due to rotation of the motor, weight of the conveyed materials and due to the idlers. The belt tension must be proper range to avoid slippage between the drive pulley and the belt. Tension of belt at steady state is given by:

$$T_{SS} = 1.37 \times f \times L \times g\,[2 \times M_i + (2 \times M_b + M_m) \times \cos\beta] + H \times g \times M_m$$

Where,
T_{SS} = Belt tension at steady state (N)
f = Coefficient of friction (0.02)
g = Acceleration due to gravity (9.81 m/sec^2)
M_i= Load due to the rollers (200 kg)
M_b = Load due to belt (20 kg total load of belt)
M_m= Load due to conveyed materials (168 kg)
β = Inclination angle of the conveyor (10^0)
H =Vertical height of the conveyor (0.35 m)
T_{SS} =1.9 kN
Belt tension while starting

During the start of the conveyor system, the tension in the belt will be much higher than the steady state. The belt tension while starting is (T_s)

$$T_s = T_{SS} \times k_s \text{ in kN}$$

Where,
k_s= Startup factor: 1.08
T_s = 2.052 kN

Unitary Maximum tension, T_U Max= $\frac{Tss \times 10}{B}$ in N/mm

T_U Max = 15.83 N/mm
Belt Power, $P_b = T_e \times v$

P_b = 2.17 kW

Effective pull, $F_u = \mu_T \times g\,(M_m + M_B/2) + \mu_R \times g \times (M_B/2 + M_i)$

Where,
μ_T= Coefficient of friction with support rollers (0.03
μ_R= Coefficient of friction with skid plate (0.33)
g= Acceleration due to gravity (9.8m/s^2)
M_m= Total load of conveyed materials (168 kg)
M_B= Mass of belt (20
M_i= Mass of rollers (200kg)

F_u = 737.45 N

Power at drive pulley, $P_p = \frac{F_u \times v}{1000}$ in kW

P_p = 0.368 kW

Minimum power of Motor (P_{Min}) = Pp/ η kW

Considering a factor of safety of 2, power required to drive pulley is 0.737 kW, efficiency of gear reduction (η) 0.8. So motor power P = 0.92 kW = 1.23 hp

Belt Breaking Strength, $B_{bs} = \frac{C_r \times P_p}{C_V \times v}$ in N

Where,
C_r = Friction factor (15)
C_V = Breaking strength loss factor (0.75)
P_p = Power at drive pulley (W)
V = Belt speed (0.5m/s)

B_{bs} = 14720 N = 14.7 kN
Acceleration, A = $(Ts-T_{ss})$ / L× $(2\times M_i+2M_B+M_m)$
A =1.02 × 10^{-4} m/sec^2

Cycle time of conveyor = $\frac{2L}{V}$

Cycle time of conveyor = 8.76 sec.

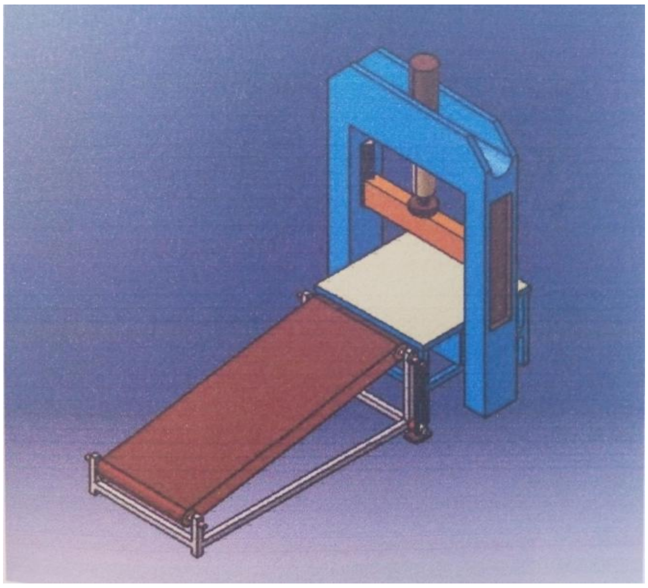

Fig -6: proposed conveyor system

Table -4: Design values for belt conveyor

Sl No	Parameter	Dimension
1	Belt Length	2.19m
2	Inclination	10^0
3	Belt Speed	0.5 m/ s
4	Belt Capacity	300x10^3 kg/hr
5	Belt Width	1200mm
6	Roller Diameter	6 inches
7	Total Belt Length	4.38m
8	Power Required to produce Lift	0.32 kW
9	Return Side Friction	0.168 N
10	Total Empty Friction	1.19 kN
11	Load Friction	2.52 kN
12	Load Slope Tension	0.63 kN
13	Effective Tension	4.34 kN
14	Steady State Belt Tension	1.9 kN
15	Belt Tension while Starting	2.052 kN
16	Unitary Maximum Tension	15.83 N/mm
17	Belt Power	2.17 kW
18	Effective Pull	737.45 N
19	Power at Drive Pulley	0.368 kW
20	Minimum Motor Power	23 hp
21	Belt Breaking Strength	14.7 kN
22	Acceleration	1.02x10^{-4} m/ s^2
23	Cycle Time of Conveyor	8.76 s

2.4 Modeling, Meshing and Analysis.

The modeling of a required blade and blade holder are done by using the Unigraphix software (UG-NX 8.0) product of Siemens, initially the individual parts were modeled as per the dimensions. After completion of modeling of all drawings it was assembled together.

Parts modeled are shown in figures below

Design 1

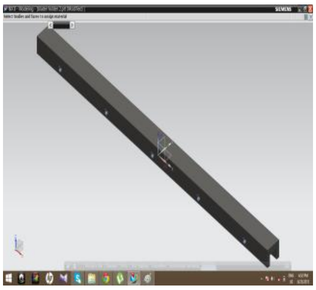
Fig -7: blade holder

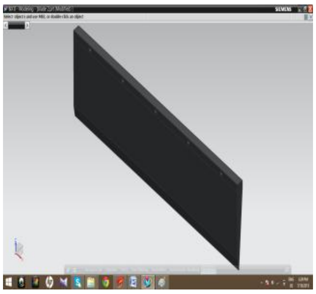
Fig -8: blade

The blade holder fits on the blade and they are assembled by using the hexagonal headed nut and bolt.

Design 2

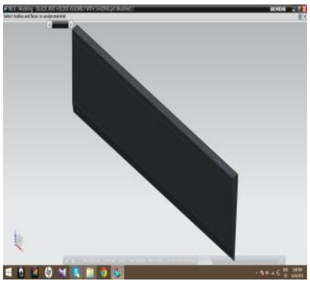
Fig -9: blade

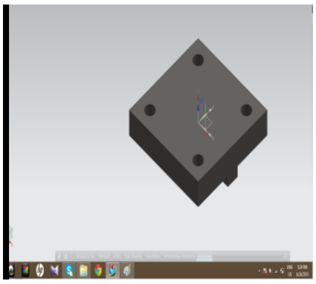
Fig -10: blade holder center top

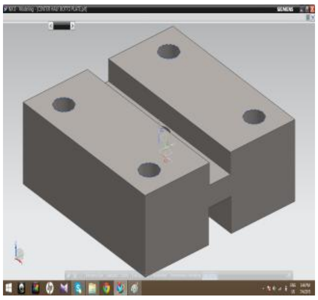
Fig -11: blade holder center Bottom

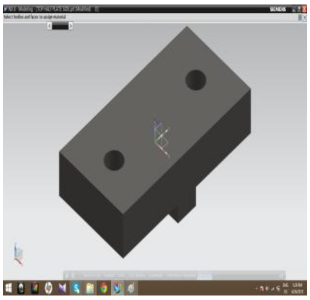
Fig -12: blade holder side top

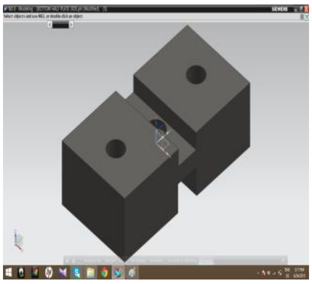
Fig -13: blade holder side bottom


Fig -14: Helen bolt

The top half blade holders are fixed to the ram, the blade holder bottom half fits on the blade and the threaded hole provided at the blade top matches the hole provided at the bottom half blade holder (both centre and side) the Helen bolt is used to fasten the blade to the bottom blade holder, and by using the hexagonal headed nut and bolt bottom half and top half are fastened. Assembly drawings of both models are shown in fig 2 and fig 3

Meshing is carried out by using the Hypermesh 11.0 As the model is more than 8mm in thickness so the Hexamesh was carried out, meshing divides complex body into various smalldivisions with maintaining systematic interconnectivity between the parts which helps in analyzing process and helps in obtaining accurate results

The meshed model of the designs are shown in figure below

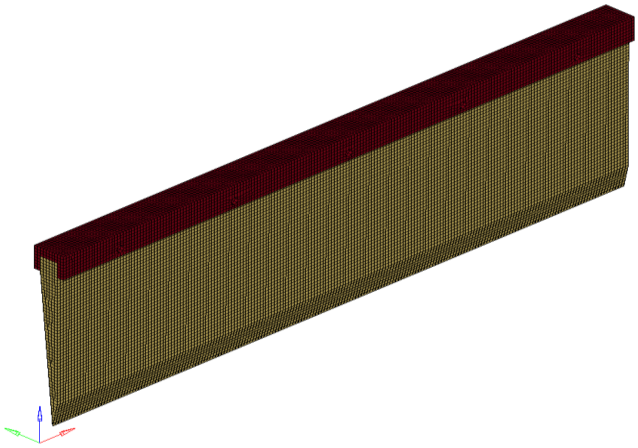
Fig -15: Mesh model of design 1

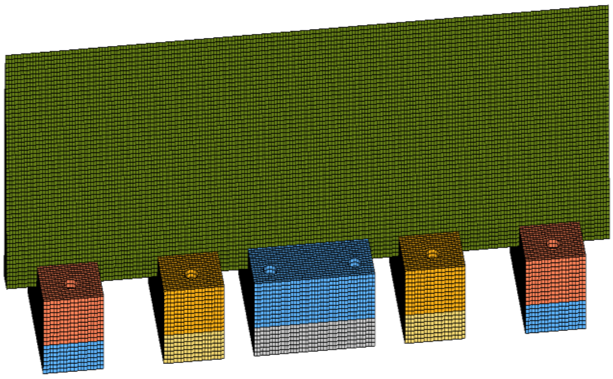
Fig -16: Mesh model of design 2

Analysis of the blade and holder designs were carried out using the Nastran pattern as the blade and blade holder are rigidly fixed to the ram so the static analysis is suitable for the case. The analysis comprises of finding out the stress and displacement analysis (deformation), the both models are made to act with an shear force (350×10^3) in N.

Displacement and stress analysis for design1

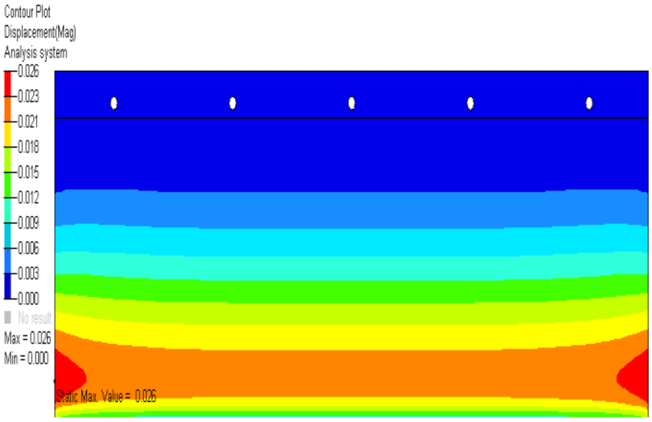

Fig -17: Displacement analysis of design 1

The results obtained shows the maximum displacement in the model is 0.026 mm and the design is safe to use.

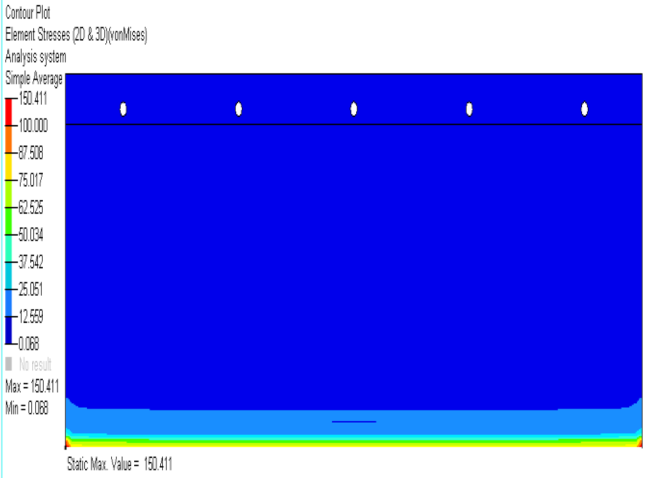

Fig -18: Stress analysis of design 1

The results obtained shows the maximum stress in the model is 152.41 N/mm^2 where as the allowable stress level is 960 N/mm^2 and the design is find safe to use.

Displacement and stress analysis for design 2

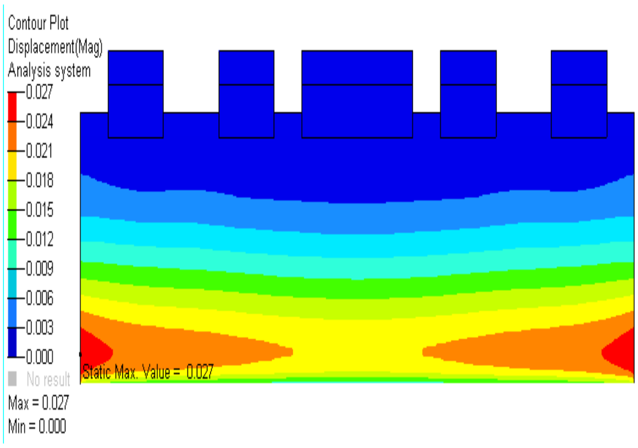

Fig -19: Displacement analysis of design 2

The results obtained shows the maximum displacement in the model is 0.027 mm and the design is safe to use

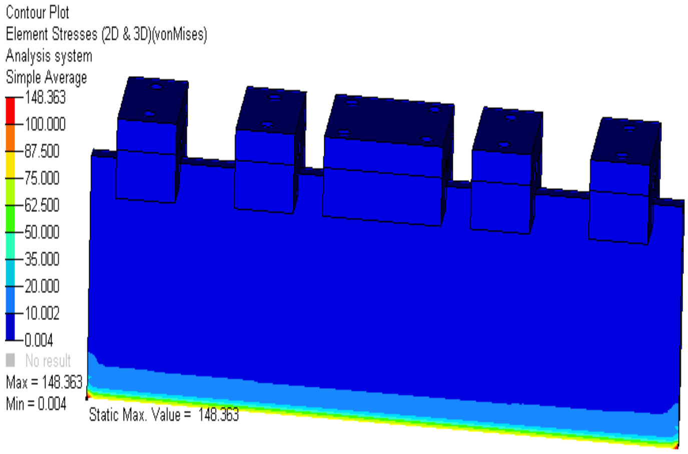

Fig -20: Stress analysis of design 2

The results obtained shows the maximum stress in the model is 148.363 N/mm^2 where as the allowable stress level is 960 N/mm^2 and the design is find safe to use.

3. RESULTS AND CONCLUSION

3.1 Performance Comparison

After the modification of the machine the results found were quite interesting the blade designed was successful in cutting four bales in a single stroke and the labour required for handling the process is increased from 3 to 4 till though the process is found more economical as show below.

Table -5: performance comparison

Performance of bale cutting machine before and after modification			
Production for 2890 bales	Before modification	After modification	
		Without conveyor	With conveyor
Time required in hours	24.08	6.82	5.61

Table -6: specification of bale cutter before and after modification

Make	**Existing**	**Modified**
Cutting stroke	500 mm	500 mm
Diameter of cylinder	150 mm	150 mm
Cylinder pressure	200 kg/sq cm	210 kg/sp cm
Blade width/stock thickness	400 mm/150 mm	1000 mm/280 mm
Time of cutting stroke	12 seconds	12 sec
Motor power	15 hp	21 hp
Number of bales cut per stroke	1 no’s	4 no’s
Material handling time	18 sec	16 sec
Stress on labour	More	Reduced
Material handling	Manual	Conveyor system

Table -7: comparison of cost incurred to run machine per month

Cost of running per month of bale cutting machine before and after modification		
Before modification	After modification	
	Without conveyor	With conveyor
Rs.129500	Rs.64280	Rs.52500

3.2 Conclusion

After modification that is., redesigning of the bale cutting machine resulted in savings of 70-80% in time, 50-65% of money, with achievement of 100% planned productivity. Whereas the original machine was giving about 90% productivity and increased expenditure further stress on the workers. And the overall modification cost (without conveyor) is Rs 116340 with payback period of 1.783 months, and the overall modification cost (with conveyor) is Rs 206340 and the payback period is 2.67 months.

Table -8: Overall performance of machine

	Time in hours	Productivity in no's	Money in Rs.
Estimated plan	24.08	2890	1,29,500
Performance achieved bale cutting machine	21.25	2610	1,29,500
Performance after modification	6.82 without conveyor (71.67% reduction) 5.61 with conveyor (76.70% reduction)	2890	64,280 without conveyor (50,36% reduction) 52500 with conveyor (59.45% reduction)

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