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Design of Combined Press Tool for the Manufacturing of Rice Thresher Blade (Case Study at Amhara Agricultural Mechanization and Food Science Research Center-Ethiopia)

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ABSTRACT

Combination press tool is a die in which cutting operation and a non cutting operation on a part is accomplished in one stroke of the press. This paper deal about the development of combination press tool to be used for the manufacturing rice thresher blade. The press tool performs double lancing operation, which is bending and blanking, on a single stroke of the press tool. The case study was taken to replace the existing rice threshing machine drum which is produced currently by welding to replace by using combination press tool. The rice threshing machine is used to thresh rice of large amount with a little time. The initial design of the thresher machine is composed of stamped sheet metal as a blade for threshing. But the case company (Amhara Agricultural Mechanization and Food Science Research Center), presently uses a drum fabricated by welding, which is found to be less effective and less efficient relative to stamped blade threshing machines. This welded drum was used due to lack of press tool to manufacture the drum blade from sheet metal. Generally the existing system of manufacturing causes; production of less efficient threshing machine, larger weight and less portable threshing machine, shorter life of product due to structural instability and higher cost of manufacturing and lower productivity rate due to extended production lead time of the thresher machine up to three days.

This research focuses on designing of combined press tool to be used in the fabrication of the rice thresher blade, two dimensional and three dimensional modeling of the components, analyzing stress and displacement on the components. The modeling and structural analysis of the components was carried out on SolidWorks 2013 and is found to be acceptable. The design and selection of combined press tool components is carried out by following standard die design approach and analysis methods. The designed combined press tool system incorporates combined press tool, which is suitable to use on the existing press machine in case company. Also the simulation and stability of the combined press tool is validated using Logopress3. The rrice thresher blade which is manufactured by the combination tool will result in higher production rate by decreasing the production lead time from three days to one day this reduces the cost of manufacturing.

***KEY WORDS**: Combined Press Tool, Thresher Blade, Punch Design, Die Design*

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I. INTRODUCTION:

1.1 Back Ground of the Study

In the technology level we reach today, sheet-metal parts have already replaced many expensive cast, forged, and machined products. The reason is obviously the relative cheapness of stamped, or otherwise mass-produced parts, as well as greater control of their technical and aesthetic parameters. That the world slowly turned away from heavy, ornate and complicated shapes and replaced them with functional, simple, and logical forms only enhanced this tendency.

Considering the organization, Amhara Agricultural Mechanization and Food Science Research Center, it is one of the organizations which provide agricultural machines by redesigning the imported machineries or designing new machines to the local farmers. From the number of products manufactured at the company workshop, a rice thresher is one of them. This rice thresher machine has a sheet metal blade structure when designed by the patented organization. But due to the lack of press tool, the blade is temporarily replaced by a welded bar component.

1.2 Statement of the Problem

The rice threshing machine is used to thresh rice of large amount with a little time. But in this case company currently uses welded bars on the threshing drum, which is not effective and efficient relative to

stamped blade threshing drum machine. The imported original design of the thresher machine is composed of stamped sheet metal as a blade for threshing. The problems observed are listed below:

- The efficiency of the current welded drum is 44kg/hr, but if it were with the original patented design it will increase to 80kg/hr.
- The welded round bar which is used as blade increase the total weight of the machine by 10 kg (26kg to 36kg). This increment in mass will be an obstacle to move the machine from place to place for the farmers easily.
- The welded member is made of mild steel rods, which is easily corrosive material. This will cause food contamination.
- The strength and uniformity of the welded blade depends on the skill of the operator and would not create a uniform strength throughout the drum and shorten thresher life.
- The production lead time for completing each drum by welding bars on the drum takes up to 3 days, which cause an increase in manufacturing time.

1.3 Objectives of the Study

The general objective of this research is to design and validate by FEM analysis of compatible combined press tool to the existing hydraulic press machine at the case company (Amhara Agricultural Mechanization and Food Science Research Center).

1.4 Specification of the Press Machine

The available press in the case company is hydraulic press type punch with model number of CEU-13, which has a maximum punching capacity of 35tonnes.

Table 1 Specification of the available hydraulic press

Press Type	Tonnage	Shut height	Ram adj.	Stroke	Bed size (L x W)
Hydraulic	35ton	165mm	30mm	10mm	270mm x 250mm

1.5 Methodologies of the Study

The methods utilized are:-

- Literature review and data collection about die designing concept and existing system in the company
- Analytical design of combined press tool components by following standard approaches
- 2D and 3D modeling of components using Modeling software
- Simulation and validation of components using FEM software

II. LITERATURE REVIEW:

The design of die is variable according to the aim it is intended to fulfill. But the design of the press involves certain common principles and steps; those should be followed in order to have an efficient and acceptable outcome. Many researchers have conducted researches on designs of dies for sheet metal components. This works can be broadly classified considering the area to be applied as; food sector, automotive, aerospace, agricultural, medicine, electronics, etc.

The researchers found that it is relevant to give attention on die design literatures that are dedicated to blanking and bending of sheet metals, some of these works are discussed as follows.

'Design of Blanking and Piercing Press Tool for Manufacturing Washers of M12', (K.Kishore Kumar, May 2012). The methodology involved in the work includes manual design of parts for press tool components followed by 3D modeling with Pro/E software. One of the draw-back that is observed on that work is that the researcher did not validate or taste the manual design for functionality using simulation software or any other techniques.

Another literature found relevant is 'Press Too Design and Analysis for 49 Lever 5 Stage Tool', (H.Ameresh and P.Harishankar, July 2013). This work involves design of punching and bending tools used to manufacture 49 lever component used in thermostats. In this work manual design of die component was performed by following the standard calculation steps. Then the outcomes of the design were modeled in Pro/E software and imported to ANSYS for validation of the design under the given load conditions. The limitation that was observed, the validation of the design was performed on an imported model to ANSYS, Which was initially modeled in Pro/E. This may result in simulation errors due to the 3D model information transformations error.

This particular research dedicated to the design of combined bending and blanking tool used to manufacture thresher blade, focused on following the standard design steps in order to eliminate errors as much as possible. This aim is believed to be achieved by conducting deep literature survey in the die-design area to

select the best methodology which can solve the problem efficiently. Standard design steps for manufacture and assembly are conducted, and the outcomes of the design are first validated for strength and functional acceptance by simulating the SolidWorks 3D model of the components and solving structural analysis in SolidWorks simulation package. This will eliminate the occurrence of validation error due to exporting of 3D-models to simulation software.

III. DESIGN OF COMPONENTS OF COMBINATION PRESS TOOL:

3.1 Thresher Component Model Description

The blade is made of hot rolled medium strength steel sheet of 1.5mm thickness having 380N/mm^2 ultimate shear strength and 310 N/mm^2 yield strength.

The length of the component which is used to make a component having 6 blades is 554mm with 66mm width (fig 1a). This strip (554mm x 66mm) is first cut out from the sheet stock. Then the strip is ready for stamping operation. After 6 blades are provided using the combined tool, the stamped component is provided with 5mm edge bends from both of the length sides. This is for additional strength and structural stability. 12 components of the stamped sheet are welded on the round disk, which is connected to the shaft, and are used to make a single drum assembly (fig 1b and 1c).

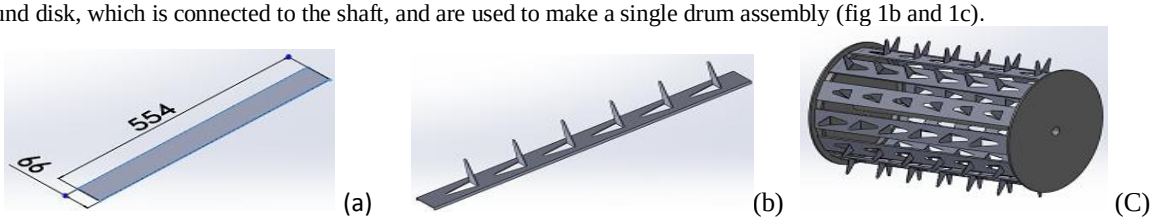


Figure 1: (a); Raw strip dimension; (b)Stamped sheet component ; (c)Drum assembly

3.2 Punch and Die Block Material

Appropriate tool material selection is essential for the functionality and manufacturability of a product. To obtain longer die life and hence higher productivity, tool steels are being widely used as materials for die components.

Selection of tool steel for a specific operation is based the following major considerations:

i. Predicting the performance requirement of the steel for an application
ii. Availability of tool materials
iii. Analyzing the limitations associated with the manufacturing of the tool

Table 2: Recommended tool steels for combined tooling [1]

Application	AISI Steel Type	Hardness, Rockwell C
Blanking dies and punches (long run)	A2	58-62
	D2	58-62
	M4	58-62
Bending dies and punches	O1	58-62
	A2	58-62
	D2	58-62

Since the punch and die selected should be capable of combining the blanking and bending performance requirements. So the tool steel materials in the first category of candidate materials will be O1, A2 and D2 as shown in table 2. The criteria to rank them is by giving relative weighting factor for wear resistance, toughness, machinability, non deforming property, safety in hardening and resistance to softening effect due to heat. AISI D2 is a high carbon, high chromium tool steel alloyed with molybdenum and vanadium characterized by high wear resistance, high compressive strength, good through hardening, high stability in hardening and good resistance to tempering back is selected for punch and die block.

3.3 Die Clearance Determination

For this particular thresher blade made of C40 carbon steel of thickness 1.5mm, the die clearance will have maximum and minimum values as indicated in table 3 is from 2.5% to 5 %.

Table 3: Die clearance for various materials [2, 3]

Material	Steel sheet	Aluminum	Brass
Clearance % of sheet thickness	2.5-5 %	1.5-3 %	1.5-3 %

Die clearance = thickness of the sheet X recommended percentage

$$\text{Minimum die clearance} = 1.5\text{mm X}\frac{2.5}{100} = 0.0375\text{mm}$$

$$\text{Maximum die clearance} = 1.5\text{mm X}\frac{5}{100} = 0.075\text{mm}$$

Since excessive clearance cause more burr and reduced clearance cause frequent reshaping and short tool life, the researchers select to take the intermediate clearance (value of 3.75%) value between the maximum and minimum clearances.

$$\text{Optimum clerance} = \frac{0.075 + 0.0375}{2} = 0.05625\text{mm}$$

3.4 Force Analysis

Cutting of metal strip takes place due to the shearing in blanking and piercing operations. The cutting operation of metal strip takes place due to the plastic deformation followed by shear and break. Forces that develop in the shearing operation can be represented by a triangle as depicted in fig 2. The vertical shearing force is represented by V while the horizontal lateral force is designated H. The resultant force is represented by R. The value of the vertical component V depends upon the shear strength of the material to be cut and the area to be sheared. Shear area is the product of the length of the cut and the sheet thickness [2, 4].

1. Blanking Force

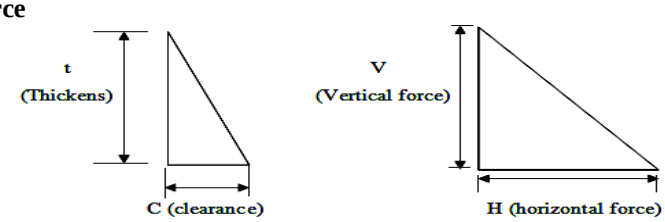


Figure 2 Decomposition of forces in shearing

$$V = \text{shear area x shear sterength of the sheet} \quad \ldots\ldots 1$$

$$\text{shear area} = \text{Cut length x Sheet thickness} \quad \ldots\ldots 2$$

$$V = 171.591\text{mm}^2\text{x}380\ \text{N/mm}^2 = 61404.2\ \text{N} = 6.26\ \text{ton}$$

The value of the horizontal or lateral force H depends upon the die clearance: the gap between the punch and die cutting edges.

Then the horizontal and resultant blanking force R_{blank} is given by;

$$H = \text{clerance \% with thickness x V} \quad \ldots\ldots 3$$

$$= \frac{3.75}{100}\text{x } 61404.2\ \text{N} = 2302.66\text{N} = 0.25\ \text{ton}$$

$$R_{blank} = \sqrt{V^2 + H^2} = \sqrt{(6.26)^2 + (0.25)^2} = 6.27\ \text{ton}$$

Since the operation is blanking the clearance is given on the punch while die is made to exact size.

$$\text{Punch size} = \text{Die bore} - 2\ \text{(die clearances per side)}$$

In order to avoid the jamming action the 3mm land is provided. And to relive internal pressure 2 degree angular clearance is given all around the die opening as shown on the fig 3.

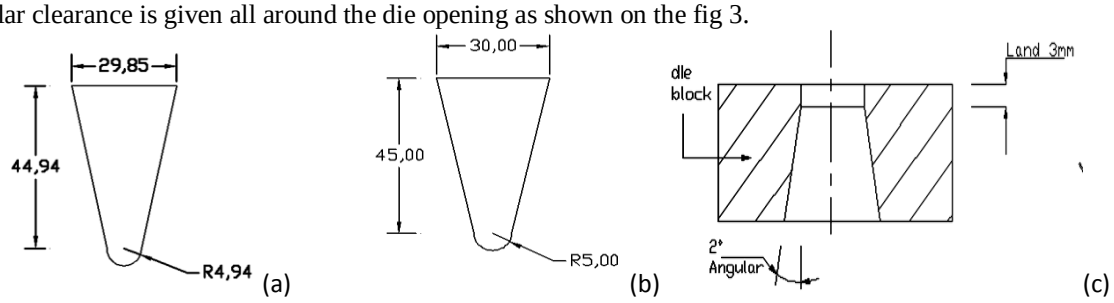


Figure 3 (a) Blanking punch profile; (b) Die profile; (c) Die land and angular clearance

2. Bending Force

Bending force required to bend a work piece depends up on thickness of the work piece, die opening, length of bend and amount of coining bottoming or ironing required. In order to find the wipe bending force required for this particular C40 steel sheet, it is required to find the bend radius to be used.

Table 4 Minimum inside bend radius for various materials [4]

Condition	Aluminum	Brass	Copper	Steel
Dead soft	0.25t	0.55t	t	0.5t
¼ hard	0.5t	t	1.5t	t

From table 4 the minimum inside bend radius to be used is't', because the C40 steel sheet is to be bent at hot rolled 1/4 hard condition. So R_{bend} equals to the thickness (1.5mm). The external bend radius for carbon steel sheet can be computed by adding the thickness to internal radius: i.e. $R_{bend\ out} = 3$mm. The bending force required can be calculated by using the following formula using die opening factor (K) equal to 0.33 for wipe bending [5].

$$F_{bend} = \frac{K\,L\,S\,t^2}{W} \quad \ldots\ldots 4$$

$$= \frac{0.33 \times 30 \times 480 \times (1.5)^2}{4.5} = 2376\text{N} = 0.3 \text{ ton}$$

L = length of bent or contact length which is 30 mm for single blade

S = ultimate tensile strength which is 480N/mm^2

t = thickness of the material which is 1.5mm

W = width of bend die, which is the summation of bend radius on the die and punch plus the die clearance $= 1.5 + 1.5 + 1.5 = 4.5$mm

3.5 Press Tonnage Requirement

The capacity of the press is the ability to deliver enough force necessary to perform the metal working operation. And the press machine should be capable of delivering about 33% more force than the required for consistent performance [2, 4]. As per the plan is to design a combination die for the lancing operation in a single stroke of the ram, the individual values for blanking and bending should be added.

$$\text{Total force per blade} = R_{blank} + F_{bend} \quad \ldots\ldots 5$$

$$= 6.27 \text{ ton} + 0.3 \text{ ton} = 6.57 \text{ ton}$$

Since the aim of the researchers are to design the combined tool which is capable of producing two blades per a single stoke of the ram, the total required tonnage for this operation will be 13.2 ton.

As stated above the press machine should be capable of providing 33% more force than the required for consistent performance.

$$\text{Total Tonnage Requirement} = 1.33 \text{ X } 13.2 \text{ ton} = 18 \text{ tonnes}$$

The total tonnage required is below the capacity of the hydraulic press available in the case company is 35 ton, hence it is acceptable.

3.6 Determination of Tool Shut Height

Tool shut height should be kept 5-10mm less than press shut height to provide a little height adjustment during press setting. The screw adjustment (of 10-20mm) for the ram should be available to take advantage of reducing tool shut height [4].

$$\text{Maximum tool shut height} = 165 - 10(\text{adj.}) = 155\text{mm}$$

$$\text{Minimum tool shut height} = 155 - 30 = 125\text{mm}$$

$$\text{Optimum tool shut height} = 125\text{mm} - 10\text{mm (rigrinding alloance)} = 135\text{mm}$$

The results show that it would be possible to use a combined tool on the existing hydraulic press machine keeping the shut height in the range of 125mm up to 155mm.

3.7 Die Block Design

The design of the die block basically depends up on the work piece size and thickness, the contour of the work piece and the die material [5]. Hardened shearing and bending dies can be safely stressed up to 160N/mm^2 compressive stress and 240N/mm^2 shearing stress [4]. The die thickness can be computed by using the following formula.

$$t_D = \sqrt{\frac{3V}{f_t}\left[\frac{\left(\frac{B}{A}\right)^2}{1+\left(\frac{B}{A}\right)^2}\right]} \quad \ldots\ldots 6$$

Where,

t_D = thicness of die , f_t = permissible tensile stress, B = width of the slot (30mm),

A = length of slot (50mm)

$$= \sqrt{\frac{3(61404.2)}{160}\left[\frac{\left(\frac{30}{50}\right)^2}{1+\left(\frac{30}{50}\right)^2}\right]} = 17.45 \text{ mm}$$

. So the final thickness including the reshaping allowance will be $17.45 + 3.55 = 21.00$ mm.

$$M = (1 \text{ to } 2) \text{ x } t_D = (2) \text{ x } 18 = 36 \text{ mm (Taking the highest value)}$$

In the design of die block the parameters that are to be computed are the die thickness (t_D), the margin (M), the die length (L_D) and the die width (W_D).

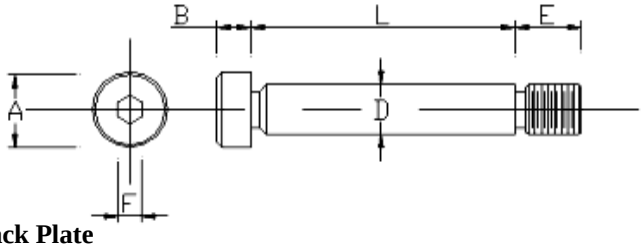
Now the die can be checked for shear stress as follows by calculating the die shear area first.

$$\text{Die shear area} = 2M \text{ x } t_D \quad \ldots\ldots\ldots\

chosen from the available standard spring series. As the selected spring 43 has internal diameter of 1/2in the corresponding standard bolt will have the following dimensions [7].

Table 5 Stripper bolt size

Diameter (D)	Length (L)	Thread diameter (C)	Thread length (E)	Head diameter (A)	Head height	Size hex. Hole (F)
9.525mm	36mm	10mm	12mm	14.3mm	6.35mm	4.76mm



ii. Design of Punch Back Plate

Backing plate is used to distribute pressure uniformly over the whole area (maintain uniform stress), it prevents the stress concentration on any portion of the punch holder. Also prevents the hardened punch from dig in to the soft surface of the upper bolster [5].

Backup plates are made of hardened steel of 3/8 in (9.5 mm) thick for general work, 1/2 in (12.5 mm) thick for heavy duty jobs. The hardness should be in the vicinity of 40 to 50 HRC. A2 steel is preferred to oil-hardened steels, which tend to be warped by the heat treatment, which affects their flatness. So 10mm A2 steel is used as a buck up plate and is provided with one M10 screw and 16mm diameter dowel pin to hold the each punch in place.

3.9 Design of Punch Holder

The punch plate is designed, dimensioned and manufactured similarly to the die block. There is one difference though, when considering the view location: the die block is always viewed of its top surface, where as the punch plate is seen from below, to determine the thickness and margin of the punch holding plate:

$$t_{PH} = \sqrt{\frac{3V}{f_t}\left[\frac{\left(\frac{B}{A}\right)^2}{1+\left(\frac{B}{A}\right)^2}\right]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 12$$

$$= 18 \text{ mm}$$

Where, t_{PH} = thickness of punch holder, f_t = permissible tensile stress, B = width of the slot, A = length of slot and V = the vertical force.

And the punch holder margin;

$$M_{PH} = (1 \text{ to } 2) \text{ x } t_{PH} = (1) \text{ x } 18 = 18\text{mm}$$

The length (L_{PH}) of punch holder for single punch can be computed

$$L_{PH} = A + 2M_{PH} = 50\text{mm} + 2(18\text{mm})$$

The width of the punch holder W_{PH} can be found using the margin (M_{PH}) value, the width of the slot (B) and the diameter of the holes (d) to be drilled for the die fixing screw. Four M12 x 1.75 screws are going to be used for fitting purpose.

$$W_{PH} = B + 2\,M_{PH} + 3d = 30\text{mm} + 2\,(18\text{mm}) + 3(12\text{mm}) = 102\text{mm}$$

The punch holder will accommodate two combined lancing punches. So the area (86 mm x 102 mm) is for single punch station. There is 49mm gap between the die blocks which are designed to make successive blades at a distance of 85mm. considering this 49mm gap, the overall area of the punch holding plate will be 250 mm length with 102 mm width.

3.10 Punch Design

The press fitted shank should have to exceed the thickness of the punch holder at least with a length of 3 - 6mm. The punch head is usually made 3-10mm larger around the punch body perimeter [3]. The punch shank is made to be 39.5mm long which exceeds the punch holder plate thickness (18mm) with 21.5mm. The punch head is made with 6mm thickness and is chamfered with 1mm to remove burr edges. The blanking punch is made with 0.06mm clearance all around the cutting edge. This value has to be subtracted from the nominal blank dimension all around the profile to get the punch profile for blanking operation. The bending portion of the punch is given 1.5mm and 3mm, internal and external bend radius respectively as stated in section 3.4. The bending portion is given 4.5mm flat surface on the horizontal and 10mm on the vertical surface.

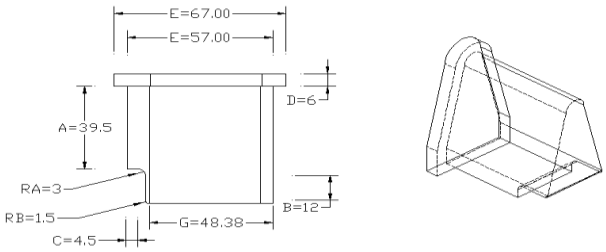


Figure 4 Punch Dimension

i. Overall Punch Length

The overall punch length depends on the shut height of the combined tool, thickness of the die bolsters, and thickness of back plates and thickness of the die block [4]. Hence, these values were calculated in section 3.6 and substituted in the following formula to find the overall punch length.

$$\text{Overall punch length} = (\text{Shut height}) - (\text{Die thickness}) - (\text{Sum of die bolster thickness}) - (\text{back plate thickness}) + 3$$

$$\text{Overall punch length} = 62\text{mm}$$

ii. Critical Bucking Force and Critical Length of the Punch

Basic length requirement for cutting punches made of steel should satisfy the following equation.

$$\frac{L}{\sqrt{I_{min}/A}} > k \quad \ldots\ldots\ldots\

$$P_{cr} = \frac{\pi^2 EI_{min}}{4L^2} \quad \ldots\ldots 16$$

$$= \frac{\pi^2 (2.1 \text{ x } 10^5 \text{ N/mm}^2)37009.68\text{mm}^4}{4(44\text{mm})^2} = 9.905 \text{ x } 10^6 \text{ N.}$$

Where, I_{min} = minimal moment of inertia = 37009.68mm^4 L = Unguided length of the punch

The critical buckling load is equal to $P_{cr} = 9.905 \text{ x } 10^6$ N, which shows the punch is safe from buckling for the applied force (63706.86 N). Further the critical pressure can combined to the safety factor in order to check for the critical length (L_{cr}).

$$P_{cr} = nP \quad \ldots\ldots 17$$

Where, n = safety factor, n = 2 − 3 for heat treated and 4 − 5 for non heat treated steels.

If a safety factor of 4 is considered to get L_{cr}, P_{cr} will be:

$$P_{cr} = nP = 4 \text{ x } 63706.86 = 254{,}827.44 \text{ N}$$

The maximum length that can be used if the applied load is equal to the critical load with safety factor of 4 will be,

$$L_{cr} = \sqrt{\frac{\pi^2 EI_{min}}{4CP}} = \sqrt{\frac{\pi^2 (2.1 \text{ x } 10^5)37009.68}{4(4 \text{ x } 63706.86)}} = 274.324 \text{ mm}$$

The punch is safe from buckling because the punch length lowers than 274.324 mm. But the length to be used also depends on the available shut height of the tool. So it is convenient to choose length that can fit the shut height. So the unguided length (distance between the punch face and bottom surface of the punch holder) of the punch is considered to be 44mm with overall length of 62mm.

iii. Compressive Strength and Deflection of the Punch

The maximum allowable compressive stress of the punch depends on the type of material used its hardness and other quantities. Tool grade steel, oil hardened and shock resistant will take up to 2070MPa before failure. So the compressive stress induced due to the applied load during operation is 68.42 MPa , which shows the tool is safe. The other parameter to be checked when punches are subjected to one sided operation, deflection due to bending force. This is observed on the punch due to the wipe bend to be performed. The punch will be checked for possible deflection as follows;

$$\delta_p = \frac{F_b L^3}{3EI} \quad \ldots\ldots 18$$

F_b = force of wipe bending = 2376N

I_{min} = minimal moment of inertia = 37009.68mm^4

E = modulous of elasticity = 210 GPa

L = Unguided length of the punch = 44mm

$$\delta_p = \frac{F_b L^3}{3EI} = \frac{2376 \text{ x } 44}{3(2.1 \text{ x } 10^5)37009.68} = 0.00868\text{mm}$$

Deflection of the punch is 0.00868 mm it shows that the deflection of the punch is within the allowable limit (< 0.025mm) [1, 2].

3.11 Design and Selection of Die Set Components

Ball bearing die sets assembly, with guide-post sliding inside a ball bearing which is contained in guide busing, is convenient for this combined tool. These classes of bearing also have advantage of easy assembly, satisfactory accuracy and can handle cutting and bending operations.

i. Design of Guide Posts and Guide Bushings

Guide posts are subjected to the bending force due to the weight of the top tool. So the deflection should be kept below 0.025mm. Rear pillars can be treated as cantilevered beams, to find the bending force 'W' using the following formula.

$$W_1 * A = W * B \quad \ldots\ldots 19$$

$$W = \frac{W_1 * A}{B} = \frac{4.5\text{kg} * 95\text{mm}}{85\text{mm}} = 4.275 \text{ kg}$$

Where, W_1= weight of top tool (4.5kg), W= weight causing bending,

A = distance between pillar and center of gravity of top tool (95mm)

B = distance between upper and lower bolster when tool is open (100mm)

By this weight the diameter of the pillar (D_p) can be estimated by the following formula;

$$D_p = \sqrt[4]{\frac{W*B^3}{15.3}} \quad \ldots\ldots 20$$

$$= \sqrt[4]{\frac{4.275 * (85)^3}{15.3}} = 20.03\text{mm}$$

The standard available ball bearing cages are provided to fit guide pillar diameter of 16, 20, 24, 25, 30 millimeters [1]. The calculated value is 20.03mm; it is selected standard dimension of 20mm for the diameter of the pillar which is available with standard lengths of 130, 140, 150, 160, 170, 180, 190 and 200.
Now the strength of pillar should be checked for crippling load failure by considering it as column with one end fixed and the other is free. For this types of loading Euler's formula can be used [7]. But before using Euler's formula, the part should be checked weather it is column or strut using slenderness ratio.

$$L/_K = \sqrt{\frac{C\pi^2 E}{\sigma_y}} \quad \ldots\ldots 21$$

$$= \sqrt{\frac{2(\pi^2 2.1 \text{ x } 10^5)}{330}} = 112$$

Where, σ_y = Yield strength of the material = 330N/mm^2
C = End fixity coefficient = 2 for one end fixed and the other free

The slenderness ratio is greater than 80, so Euler's buckling formula can be used, considering long column with one end fixed and the other free laterally, as follows to find crippling load (W_{cr}).

$$P_{cr} = \frac{\pi^2 EI}{4L^2}$$

$$= \frac{\pi^2 * 2.1 \text{ x}10^5 * 16{,}286.02}{4(130)^2}$$

$= 499{,}329.6$ N which is much gretter than the applied force (63706.86 N)hence it is safe

$$I \text{ (moment of inertia)} = \frac{\pi D^4}{64} = 16{,}286.02 \text{ mm}^4$$

L (Length of the column) (L = 2l) for one end fixed and the other is free

As explained above deflection caused by bending force due to the weight should be kept below 0.025mm. Considering as cantilevered beams, deflection can be found using the bending force due to the weight ($W_{bending}$) and free length (L = 130mm) as follows;

$$W_{bending} = W * 9.81 = 4.75\text{kg} * 9.81 = 46.5975\text{N}$$

$$\delta_{pillar} = \frac{W_{bending}\ L^3}{3EI} \quad \frac{46.5975\ (130)^3}{3(2.1 \text{ x}10^5 * 16{,}286.02)} = 0.0097\text{mm}$$

Since the deflection is in limit of 0.025mm, the guide pillar is safe.
The length (L), internal diameter (D_i) and external diameter (D_o) of the standard bush corresponding to 20mm ball type pillar will be 80mm, 28 mm and 47 mm respectively. The ball bearing will have length of 70mm with ball diameter of 4mm. Fit recommended between guide pillar and upper bolster (upper plate) is H7/p6, guide pillar and ball cage H7/g6 (precision location and sliding) and guide bush and Lower bolster (bottom plate) is H7/p6.

ii. Design of Upper and Lower Bolster Plates

The bolster is made to fit the tightening bolts on the press parallels, the area of the bolsters 270x250mm. Using 250mm width, 30mm distance between parallels and thickness of 28mm the bolster can be checked for allowable limit of deflection as follows by considering it as a simple supported beam subjected to uniformly distributed load.

$$d_{lower\ \ bolster} = \frac{5PL^3}{384EI} \quad \ldots\ldots 22$$

$$= \frac{5*129{,}447.76*(30)^3}{384*2.1 \text{ x } 10^5*457{,}333.3} = 0.00047\text{mm}$$

Where, $P =$ Press force (bending plus blanking) = 129,447.76N

$L =$ Distance between parallels = 30mm

$E =$ modulous of elasticity $= 2.1 \text{ x } 10^5 \text{ N/mm}^2$

$I =$ moment of inertia $= \frac{bt^3}{12} = 457{,}333.3\text{mm}^4$, 'b' is width and't' is thickness

This value of deflection shows that the lower bolster is safe to be 28mm thick.

The thickness of the upper bolster (22mm) is usually made 3/4th of the lower bolster; it is subjected to smaller load than the lower bolster. The upper bolster deflection can be found by considering it as simple supported beam subjected to concentrated force at the center.

$$d_{\text{up bolster}} = \frac{PL^3}{48EI} = \frac{129{,}447.76*(45)^3}{48*2.1 \text{ x } 10^5*221{,}833.3} = 0.0053\text{mm}$$

$L =$ Distance between successive bolts = 45mm, $I = \frac{bt^3}{12} = \frac{250*(22)^3}{12} = 221{,}833.3\text{mm}^4$

The value of deflection shows that the upper bolster is sufficient to be 22mm thick.

Table 6 Summary of die set materials and dimensions details

Die-set components	Material	Size
Upper bolster	M.s (St-42)	270 x 250 x 22 [L X W X t]
Lower bolster	M.s (St-42)	270 x 250 x 28 [L X W X t]
Guide bush	steel	Ø28 x Ø47 x 80 [D_i X D_o X L]
Ball cage	Aluminum alloy with steel balls	Ø20 x Ø32 x 80 [D_i X D_o X L]
Guide pillar	M.s (St-42)	Ø20 x 130 [D X L]

iii. Design of Dowel and Selection of Screw

The screws used for fastening the die and punch plate must withstand the stripping force generated during the operation. The root diameter of the screws is considered under direct tensile stress from the stripping load. The stripping force can be assumed 10% of the vertical force [4]. The design stress for socket head cup screws (SHCS) range from 80 MPa to 120MPa.

The combined tool has two combination dies which require 61404.2N vertical force for each punch. Using this vertical force the number of M8 SHCS screws can be found considering load capacity of 80 MPa.

$$\text{Stripping force} = 10\% \text{ of } 61404.2\text{N} = 6140.42\text{N}$$

The root area of metric screws can be found using the following formula.

$$A_r = 0.7854(D_B - 1.227P)^2 \quad \ldots\ldots\ldots\

$$A_{dwl} = \frac{\pi D_{Dwl}^2}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 24$$

$$D_{Dwl} = \sqrt{\frac{4A_{dwl}}{\pi}} = \sqrt{\frac{4*23.02\,mm^2}{\pi}} = 5.40\ mm \cong 6mm$$

So, two dowels of 6mm diameter can be used for this combined tool for each die block.
Screw and dowels for the upper half of the combined tool will have the same number of screws and dowels positioned in opposite direction to lower half.

3.12 Design of Stock Guide and Front Spacer

In this particular combined tool there are two combination dies side by side. Though, it will be convenient to use a channel type back guides in combination with front spacer and strip support to prevent the bulging of the thin stock. The width of the stock guide (distance between the back gage and front spacer) is made to be stock width plus 1/32 in. (0.794mm) when roll feed is used, stock width plus 1/16 in. (1.586mm) for hand. The thicknesses of both back gage and front spacer is usually made 1/8 in. (3.175mm) for strip thickness up to 1/16in. (1.568mm), for heavier strip it is made strip thickness plus 1/16 – 1/8 inch [3]. The support is relief to 3/8 in. (9.53mm) in to the supporting block (die block) to help starting of new stripes. The width and thickness of the strip are 66mm and 1.5mm respectively. So the width and thickness of the stock guide will be 67.568mm and 3.2mm respectively. The length of the component which has 6 blades is 554mm. considering the two die-blocks of 86mm length, the length of the back gage is made to be 371mm. Four socket flat head screws are used for both the back gage and front spacer. And for the stock support of 150mm length 100mm width, four flat head screws are provided to fix the stock support with the guiding element. A 67mm by 20mm steel block of 17.5mm thickness is provided to support the guiding unit against the lower bolster.

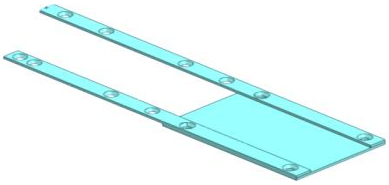


Figure 5 Guiding system and stock support 3D model

3.13 Design of Stock Locator

For this particular combined tool, there is a need of providing finger stop which can give a locating action at a distance of 18.5mm from the die block width edge. This distance is selected by considering a pitch of 85mm (distance between each blade of the sheet component). Since spring strippers are used, an inverted slot finger stop which is mounted in front of stock guide is going to be used. The dimensions of this finger stop are selected according to the available stock thickness, die-block contour and front spacer width of 67.568mm. The selected inverted type finger stop will have the following dimensions:

Table 7 Selected finger stop size

Stop number	A (mm)	B(mm)	C(mm)	D(mm)	E(mm)	F(mm)
1	3.175	6.35	16.67	37	1.58	46.23

The finger stop will be used with the aid of mounting block, shown in figure 6 (c), which is fixed on the lower bolster with two M5 hexagonal head bolts

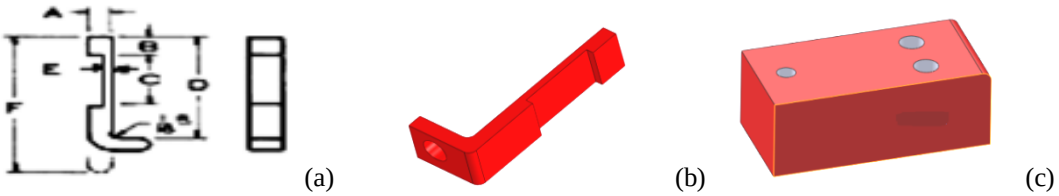


Figure 6 (a) Standard finger size used (b) 3D model of finger stop (c) Mounting block for finger stop

3.14 Shank Design

The major function of shank is to locate the combined tool on the machine ram. The ram has a seventh (H7) location hole with a transverse clamping screw. The shank location diameter should be easy running fit (f8) in the ram hole. The stripping load on the top tool tends to pull the shank out of the ram hole. So the shank is clamped by the clamping screw housed in the ram. Shanks are made of medium strength steels with a

hardness of 40-42 HRC, in which in this case DIN -C45 (AISI 1045) steel is used. Shank size is selected according to the press tonnage incorporated in the operation.

Since the combined tool stamps two blades in two stations located at equal distance from the combined tool center (ram center), it is recommended to use shanks with rectangular or circular flange type of shanks which are fixed on the bolster by screws. And the selected flange type shank should have fixing screws capable of handling the stripping force incorporated, which is 10% of 13ton (1.3ton = 12747.88N) M10 X 1.5 bolt can handle 4050N force safely, so four M10 X 1.5 screws will be used to fix the shank flange on the upper bolster. These types of shanks also have an advantage in distributing the load form press ram on the upper.

Center of pressure of the Combined Tool

The center of pressure can be found by center of gravity method, moment method or graphical method. The center of pressure will be computed using a gravity method, by taking the block edge as reference axis as follows:

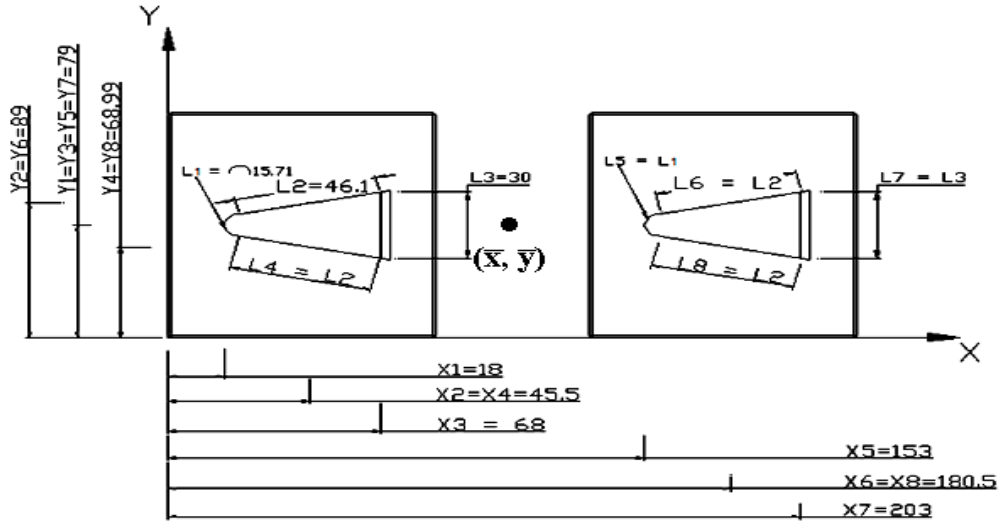


Figure 7 Center of pressure of the combined tool

Gravity method involves the following equations to find the center of pressure by substituting the length and corresponding center of each line in the blanking profile as shown on fig 7.

$$\bar{X} = \frac{L_1X_1+L_2X_2+\cdots+L_nX_n}{L_1+L_2+\cdots+L_n} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 25$$

$$= 114.76\text{mm}$$

$$\bar{Y} = \frac{L_1Y_1+L_2Y_2+\cdots+L_nY_n}{L_1+L_2+\cdots+L_n} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 26$$

$$= 79\text{mm}$$

So the ram center line (center of the shank) should coincide with centroid X= 114.76mm and Y = 79mm in reference to the die block edge.

The shank should be assembled in a position that the shank center line is the same as the position of ram center line.

Table 8 Standard shank dimensions for 13 tone force

D_1	D_3	D_4	D_5	D_6	l_1	l_2	l_3	t	t_1	r_1	Press tonnage
Ø25	**63**	**45**	**8.4**	**13.5**	**45**	**4**	**16**	**18**	**8.5**	**2.5**	**10-25**

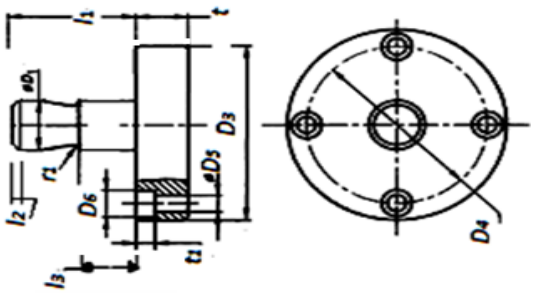


Figure 8 Standard flanged shank

IV. FEM ANALYSIS OF CRITICAL COMPONENTS :

In this section of the research, critical components will be validated by FEM analysis package (SolidWorks 2013). This helps to predict the real condition of tool components in the operation. This eliminates the traditional way of validating the design of components for specific application by manufacturing and testing on prototype, which is expensive and time consuming.

4.1. Lower Bolster

Table 9 Input data for lower bolster

Material type: St-42 structural steel, Load: simply supported beam loaded at the center with 129,447.76N Modules of elasticity: 210GPa, Poisson's ratio: 0.3	Yield strength: $\sigma_Y = 240$Mpa, Mass:13.9236 kg Volume:0.00178508 m^3, Density:7800 kg/m^3 Weight:136.451 N

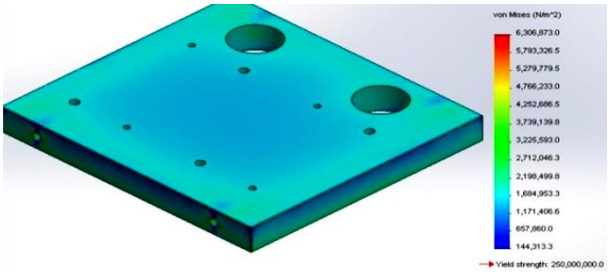


Figure 9 Lower bolster Von Mises stress

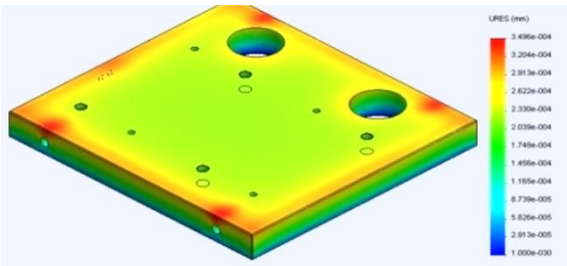


Figure 10: Lower bolster resultant displacement

Table 10 Summary of lower bolster analysis result.

Parameter	Min	Max
Stress (Von Misses Stress) [MPa]	0.169981	6.83651
Displacement (Resultant Displacement) [mm]	0	0.000347868

4.2. Upper Bolster

Table 11 Input data for upper bolster

Material type: St-42 structural steel (hardened mild steel). Modules of elasticity: 210GPa, Poisson's ratio: 0.3, Load: simply supported beam loaded at center: 129,447.76N	Yield-strength: $\sigma_Y = 240$MPa, Mass:10.7855 kg, Volume: 0.00138276 m3, Density:7800 kg/m3, Weight:105.698 N

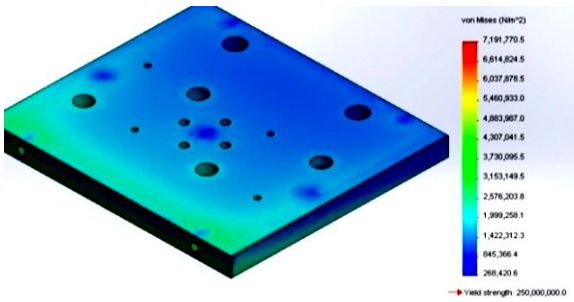


Figure 11 Upper bolster Von Mises stress

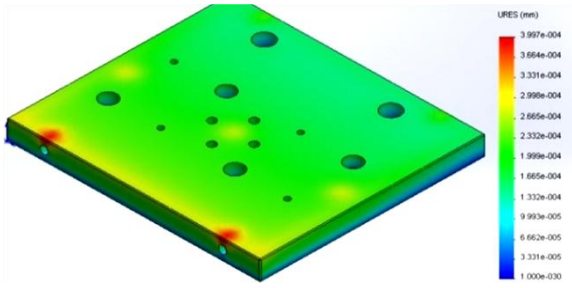


Figure 12 Upper bolster resultant displacement

Table 12 Summary of upper bolster analysis result

Parameter	Min	Max
Stress (Von Misses Stress) [MPa]	0. 268421	7.19177
Displacement (Resultant Displacement) [mm]	0	0.000399712

4.3. Stripper Plate

Table 13 Input data for stripper plate

Material type: St-42 structural steel, Load: uniformly distributed load due to stripping force on the stripping area 750lb (3633.15N), Modules of elasticity: 210 GPa, Poisson's ratio: 0.3	Yield strength: $\sigma_Y = 240$MPa, Mass: 1.42558 kg Volume: 0.000182766 m^3, Density:7800 kg/m^3 Weight: 13.9707 N

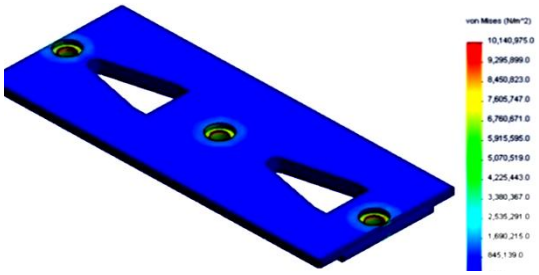


Figure 13 Stripper plate Von Mises stress.

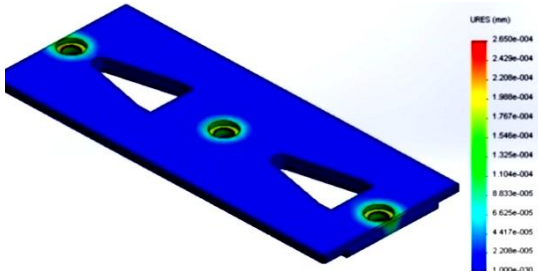


Figure 14 Stripper plate resultant displacement.

Table 14 Summary of stripper plate analysis result

Parameter	Min	Max
Stress (Von Misses Stress) [MPa]	0.000063	10.141
Displacement (Resultant Displacement) [mm]	0	0.000265005

4.4. Die-Block

Table 15 Input data for die block

Material type: D2 tool steel Load: 1. Uniformly distributed force 61404.2N vertical load. 2. Shear force from the horizontal force of 2302N 3. Shearing force due to bending operation of 2376N	Modules of elasticity: 210 GPa, Poisson's ratio: 0.394, Yield strength: $\sigma_Y = 2200$MPa,Mass:1.93177 kg, Volume:0.00025418 m^3, Density:7600 kg/m^3

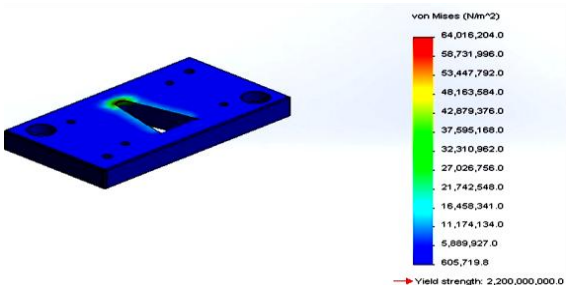


Figure 15 Die-block Von Mises stress

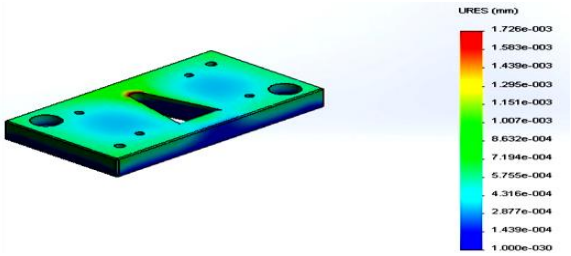


Figure 16 Die block resultant displacement

Table 16 Summary of die block analysis result

Parameter	Min	Max
Stress (Von Misses Stress) [MPa]	0.605720	64.0162
Displacement (Resultant Displacement) [mm]	0	0.00172645

4.5. Punch

Table 17 Input data for punch

Material type: D2 tool steel Load: 1. Uniformly distributed force 61404.2N vertical load. 2. Shear force from the horizontal force of 2302N.	Yield-strength: $\sigma_Y = 2200$MPa**,** Mass:0.546281 kg, Volume: 7.18791x10^{-5} m^3, Density:7600 kg/m^3

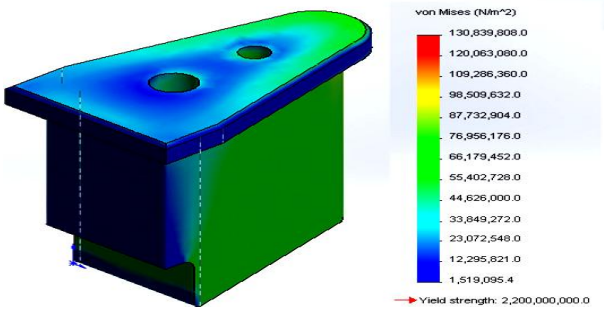


Figure 17 Punch Von Mises stress

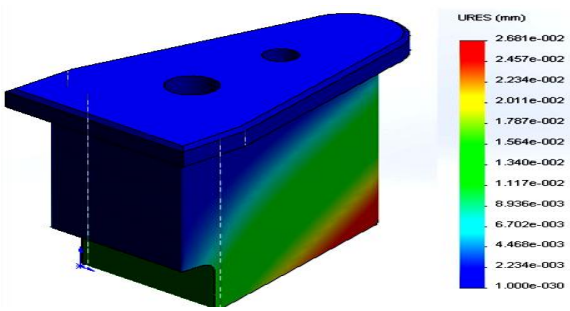


Figure 18 Punch resultant displacement

Table 18 Summary of punch analysis result

Parameter	Min	Max
Stress (Von Misses Stress) [MPa]	1.5191	130.84
Displacement (Resultant Displacement) [mm]	0	0.0026809

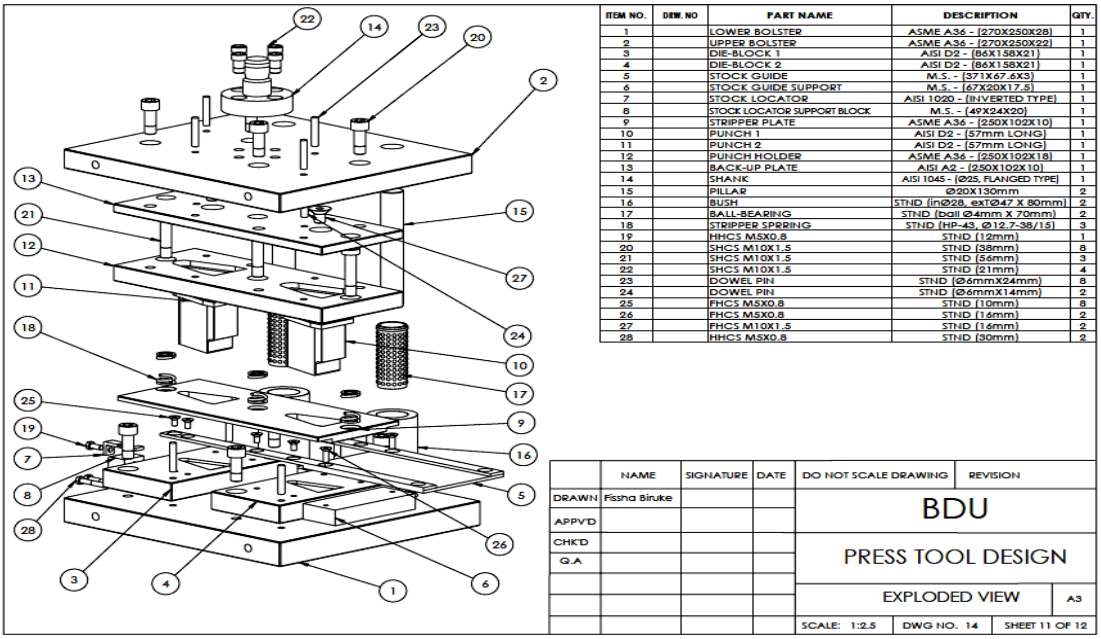


Figure 19 Exploded view of the press tool

V. RESULT AND DISCUSSION

The die block is made of D2 (HCHCr) tool steel, which is selected from the available candidate materials. An optimum die clearance of 0.05625mm is considered for the die block. Since the cutting operation is blanking, the die clearance is considered only on the die block. Angular clearance of 2° with 3mm land is provided for the die block. The area of the die block is empirically found to be 86 x158mm with 21mm thickness. With this area the die block is found to be subjected to a stress of 47.38MPa, which is within the allowable stress limit of the material.

The stripper utilized is spring loaded pipe. The thickness of the stripper plate is 10mm, and is made of structural steel (St-42). Three springs of 38mm length, this can provide 151kg stripping force, are selected from the standard spring series. The stripper assembly is equipped with the appropriate stripper bolts. The bolts utilized have total length of 54.5mm, with 12mm thread length and M10x1.5 bolts.

The combined press tool is equipped with 10mm backup plate made of A2 tool steel, which should be hardened in the vicinity of 40-50HRC. The backup plate is provided with one M10x1.5 screw hole and one 6mm diameter dowel holes to fulfill the punch holding requirement. Also 18mm thick punch holder plate made of St- 42 steel is provided to support the punches.

The punch is made of D2 tool steel. The punch length is found to be 62mm. the punch have a d/t ratio of 14.03, which is much greater than 1.10 (minimum d/t ratio requirement). The punch is considered as one end fixed and the other free column to check for buckling failure. The critical buckling load is found to be $9.905x10^6$N, while the applied load is 63706.86N. This shows that the punch is will not fail due to buckling failure. The compressive stress on the punch is found to be 68.42MPa, while the allowable stress is 160MPa. The deflection on the punch is found to be 0.00537mm, which is in the limit of allowable deflection of 0.025mm.

Die set with ball baring assembly is selected for the combined tool for the advantages of easy assembly, satisfactory accuracy, ability to handle both cutting and bending operations. The die set is equipped with 20mm diameter pillar having length of 130mm, which is made from St-42 steel. The pillar has a slenderness ratio of 112.02 (which is greater than 80), so it is considered as a long column with one end fixed and the other free to apply Euler's buckling formula. By applying Euler's equation, the pillar is found to buckle if a load of 499,329.6N is applied. So the pillar is safe to handle a load of 63706.8N. The pillar is checked for bending deflection due to the weight of the top tool, and found to deflect 0.0097mm which is below the allowable defection of 0.025mm. The lower bolster is made of hardened mild steel (St-42), to have the advantage of preventing the occurrence of cracking. The lower bolster is 28mm thick; with this thickness it is

found to deflect 0.00047mm, which is below the allowable deflection on die sets. The upper bolster will be 22mm thick; with this thickness it is deflected to 0.0053mm due to the applied central concentrated press force, which is also in the allowable deflection limit.

The results of the FEM analysis will be summarized and compared with the analytical results on the table 19; it can be clearly seen during the operation of the machine tool the allowable working stresses and deflections are within the acceptable limit.

Table 19 Comparison between analytical and FEM results

No.	Die Component	Yield strength [MPa]	Analytical calculation results		FEM results	
			Stress [MPa]	Deflection [mm]	Stress [MPa]	Deflection [mm]
1	Lower bolster	250	3.32	0.00047	6.836	0.00034
2	Upper bolster	250	9.73	0.00530	7.192	0.00040
3	Stripper plate	250	4.75	0.00054	10.14	0.00027
4	Die block	2200	47.38	0.00470	64.02	0.00170
5	Punch	2200	68.42	0.00537	130.8	0.00268

VI. CONCLUSION

Design of combined press tool for rice thresher blade made of sheet metal component has been developed by following the fundamental die design principles. The press tonnage required for the operation is below the capacity of the machine which exists in the company. Moreover the geometrical compatibility of the hydraulic press and the designed combined press tool is excellent. The analytical results and the finite element result looks sound. The limiting parameters (stress and deflection) in the design process all in the allowable limit this will make the design to meet the functional requirements. And the analytical analysis outcomes were modeled with 2D and 3D space with SolidWorks 2013 and assembled with their respective positions. This was followed by analysis in Logopress3 and structural analysis in SolidWorks simulation 2013. Manufacturing the thresher blade using the die tool which is designed will result in higher production rate by decreasing the production lead time from 3 days to 1 day; this will decrease the cost of manufacturing comparing to the previous cost of the product. In the existing condition using the round bar as a blade the welding process is used which cause structural instability and increment in weight causing difficult in portability. But in the improved state the blades are made of stamped sheets which have better structural stability. This improved state also result in a 10kg mass reduction and better portability. Also use of this combined tool to manufacture the sheet component to replace the round bar will enhance the efficiency of the rice thresher by improving the threshing capacity from 46kg/hr to 80kg/hr. Therefore newly developed combined press tool is recommended in order to have an improved productivity, improved efficiency, better flexibility, more economical manufacturing process with lower cost of the product.

REFERENCE

[1]. Ivana Suchy , *Hand Book of Die Design* , 2nd edition McGraw-Hill, 1998.
[2]. Gashaw Desie , Yonas Mitiku, *Progressive Die Design for Self Explosive Reactive Armor Holder (Case Study at Bishoftu Motorization Industry-Ethiopia),* the International Journal of Engineering and Science (IJES),Volume 3, Issue 3 Pp 75-86, 2014
[3]. J R Paquin; R E Crowley, *Die design fundamentals*, New York, N.Y: Industrial Press, 1987.
[4]. Prakash H. Joshi, *Press Tools Design and Construction*, Surya Kiran, 19, K.G.Marg,, New Delhi
[5]. G.R. Nagpal, *Tool Engineering and design*, ©2002 khanna Publishers
[6]. Kumar Shailendra, *An Intelligent System for Selection of Materials for Press Tool Components*, Journal of Engineering Research and Studies E-ISSN0976-7916, National Institute of Technology, Surat, India
[7]. Tagel Markos, *Redesigning a Combination Die and Developing a Manufacturing Plan for a Coffee Sieve,* thesis work submitted to Bahir Dar University, August 17, 2012.