International Journal of Research in Engineering and Science (IJRES)
ISSN (Online): 2320-9364, ISSN (Print): 2320-9356
www.ijres.org Volume 10 Issue 8 ǁ August 2022 ǁ PP. 59-70

Design and Analysis of a Set of Press Tools for Cable Bracket

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Abstract: *Press tools are manufactured to produce a specific component in large quantities with high accuracy on sheet metals. The press tool components are mainly used in automobiles and electrical appliances having high accuracy.This paper explains the press tool design of cable bracket in accordance with sheet metal calculations, modelling of double pitch punch progressive, cam bending tool as a set of press tools and analysis on cam bending tool in Solid Works 2020 software. The parts involved in the bending of a cable bracket in the cam bending tool are examined to assure design safety.*
Keywords: Double pitch punch progressive tool, cam bending tool, solid works 2020.

Date of Submission: 22-07-2022 Date of acceptance: 04-08-2022

I. INTRODUCTION

Press tools are unique tools to produce a required shape and size of component which are mainly of sheet metal withthe application of press. The basic operations of the press tool are piercing, blanking, bending, forming, trimming and notching. According to dies, press tools are classified into stage tool, progressive tool and compound tool.

In this paper, double pitch progressive tool and cam bending tools are designed for the production of cable bracket component. Where, a progressive tool operates a whole sequence of machining stages can be performed with a single press stroke with this tool. Bending is a forming operation which material is compressed between the male and female punches for obtaining required shape.

II. PROBLEM STATEMENT

The critical problems involved in this paper include, selecting the method of press tool design based on the size and shape of cable bracket, as it has a 60^0 bend profile on both ends. It is challenging to design using V-bending or edge bending as shown in figure 1. This complicates ejecting the cable bracket after the operation and to performing analysis according to calculated forces for producing cable bracket for the better quality of the cable bracket.

III. OBJECTIVES

a. Develop a cable bracket design model using Solid Works 2020 software.
b. Designing of press tools by analyzing the cable bracket drawing.
c. To model the tools with the use of sheet metal design principles.
d. Cam bending tool analysis for assuring design safety.

IV. COMPONENT DETAILS

The Cable bracket is manufactured from stainless steel (SS304) having 77.36 mm length, with a thickness of 1.5 mm, an inner bend angle of 60^o, and an inner radius of 1.5 mm on either ends. The 3D model of the cable bracket is shown in figure 1.

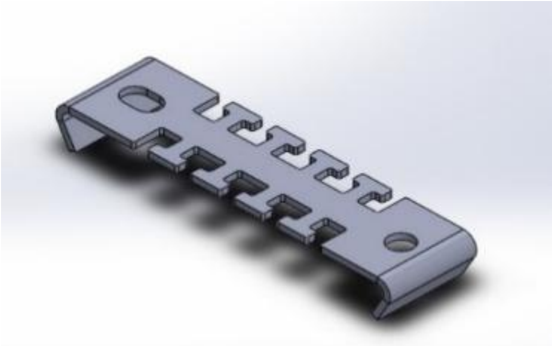

Figure 1 Cable bracket

Name	Cable Bracket
Material	Stainless steel / AISI 304
Thickness	1.5 mm
Ultimate Tensile Strength	620 MPa
Yield Strength	205 MPa
Conditions	Annealed, Hot finished
Production	5000 per month

Table 1 Cable bracket details

The cable bracket is engaged in the assembly of cable and wire accessories. The material chosen for the cable bracket is SS304, which is an excellent conductor of electricity and has a lower rate of corrosion. These cable bracket are designed for data rate connectivity.

V. TOOL DESIGN

5.1 STAGE 1: DOUBLE PITCH PROGRESSIVE TOOL

The cable bracket is carefully investigated in regards to its size, shape, and various cross sections. In this section, the design of a set of stage tools is done empirically, through experience, and through expertise as applied to cable bracket, and various design calculations are given below.

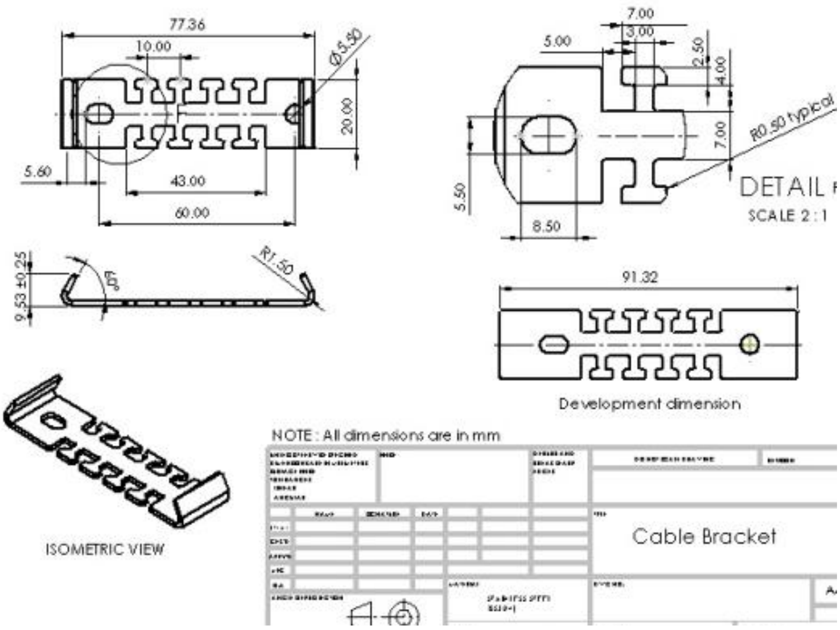

Figure 2 2D Drawing of Cable bracket

5.1.1 Development calculation:

The bend development formula is used to calculate the total length of the component before forming in order to minimize scrap consumption. The cable bracket development length calculation is shown in figure 3.

Bend Allowance $= \frac{\pi}{180} \times \theta \times (\mathrm{Kt} + \mathrm{IR})$..Eq 1

Where, ϴ → angle

t → thickness of sheet

Ir → internal radius

K → 0.33 when r < 2t

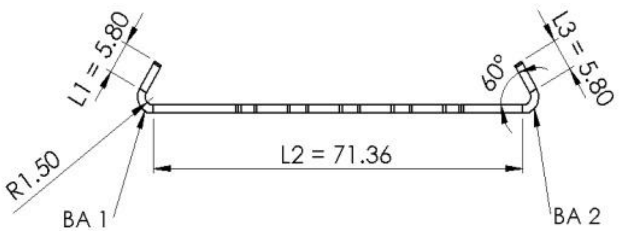

Figure 3 Bend Development

Development length = $L_1 + BA_1 + L_2 + BA_2 + L_3$

$$= 5.8 + \left(\frac{\pi}{180} \times 120 \times (0.33 \times 1.15) + 1.5\right) + 71.36 + \left(\frac{\pi}{180} \times 120 \times (0.33 \times 1.15) + 1.5\right) + 5.8$$

= 5.8 + 4.18 + 71.36 + 4.18 + 5.8

= 91.32 mm

5.1.2 Designing of strip layout

The strip design provides the material stock required to obtain the necessary component in the various operation stages. The strip layout's primary goal is to reduce scrap as much as possible while satisfying practical requirements. The formulas for reducing scrap are detailed further below.

i. Scrap Allowance$(A) = 1 \times t$..Eq 2

t is thickness of the sheet

= 1 × 1.5 mm **= 1.5 mm**

ii. Width of strip $(W) =$ Height of component $+ 2A$...Eq 3

= 91.32 + 2(1.5)

= 91.32 + 3

= 94.32 mm

iii. Pitch$(P) =$ Width of component $+ A$..Eq 4

= 20 + 1 (1.5)

= **21.5 mm**

"Since 1.5 mm is practically less allowance. So, addition of 1.5 mm is taken"

=21.5 mm + 1.5 mm **= 23 mm**

The burr direction should be down, corresponding to the bend. As a conclusion, we designed this wide strip layout shown below figure 4.

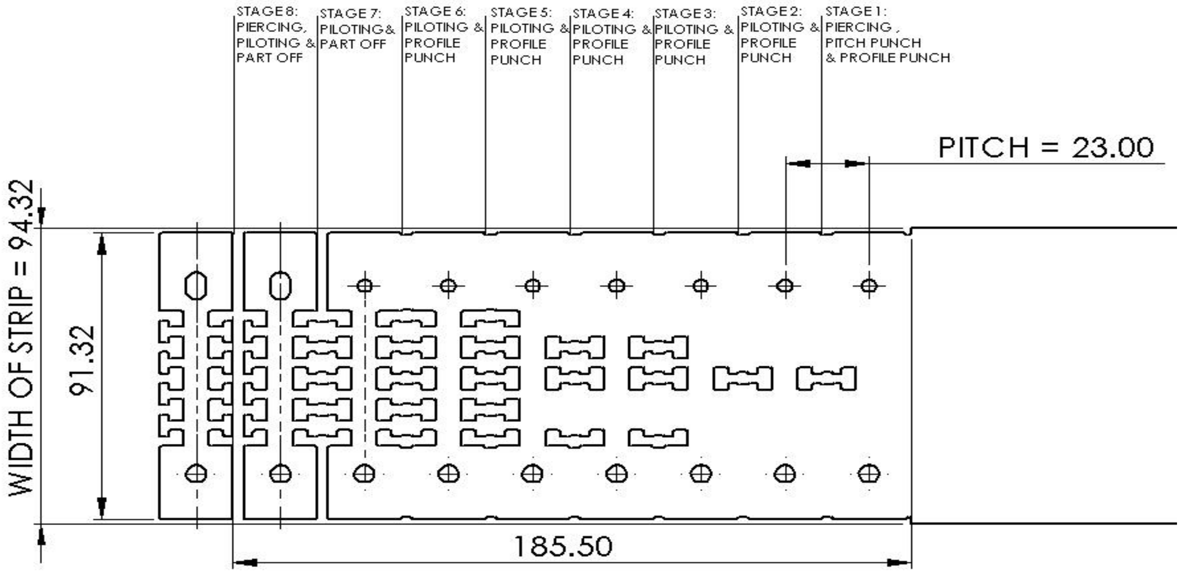

Figure 4Strip layout

iv. % of utlization $= \frac{\text{Area of blank} \times \text{No of rows} \times 100}{\text{Pitch} \times \text{width of strip}}$...Eq 5

$= \frac{1503.19 \times 1 \times 100}{23 \times 94.32}$

= **69.29%**

5.1.3 Shear force calculation

The shearing operation is required to get the desired size of the component. The component perimeter, shear stress of that material, and stock material thickness should all be determined before calculating the shearing operation. The shearing force of the cable bracket is explained in the subsequent calculation.

i. Shearing force of blank=Perimeter× Thickness of sheet×Shear Strength..........................Eq 6

= 400.33 × 1.5 × 53× 9.81

= **289618.73 N**

ii. Shearing force of oblong hole=Perimeter × Thickness of sheet×Shear Strength

= 23.29 × 1.5 × 53 × 9.81

= 18163.7 N

iii. Shearing force of Ø 5.5 mm hole=Perimeter×Thicknessof sheet×Shear Strength

= 17.29 × 1.5 × 53 × 9.81

= **13484.38 N**

Total shear force

= (289618.73 + 18163.7 + 13484.38)

= 321266.81 N

5.1.4. Stripping force

The stripping force applied after the operation is finished that enable the component to be ejected from the die. Basically, 10 to 20% of the calculated force is factored.

iv. Stripping Force=20%×Total shear force..Eq 7

= 0.2 × 321266.81

= **64253.36 N**

Total force = 321266.81 + 64253.36 =**39 T**

5.1.5. Press selection

After the calculation of the shearing force, the selection of press must be considered. So, the press should be 1.25 times of the required tonnage.

Press required tonnage

v. Press tonnage $= 1.25 \times$ Calculated Ton..Eq 8

= 1.25 × 39 T

Press tonnage = 49 T

5.1.6. Shearing Clearance calculation

The gap between the punch and the die cutting edges is referred as die clearance, the die clearance depends upon properties of the material to be sheared and the basic formula and the diagram of the cutting clearance is shown below figure 5.

Shearing Clearance $= 0.005 \times t \times \sqrt[2]{fs}$..Eq 9

$= 0.005 \times 1.5 \times \sqrt[2]{53}$

= 0.055mm

Clearance = 0.06 mm per side

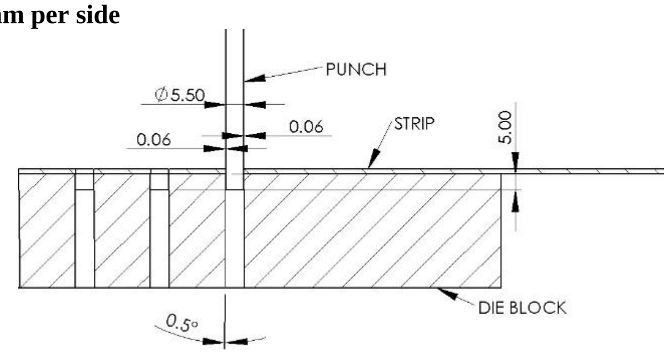

Figure 5 Die Clearance

5.1.7. Design of press tool elements

This is most important element of the tool. Die plate is cutting element of the tool which gives the blank size and shape. For determination of length, width and thickness the below formulae are used,

i. Thickness of Die block (T_D)= $\sqrt[3]{\text{Calculate Total Tonnage}}$..Eq 10

$= \sqrt[3]{39000}$

= **35 mm**

ii. Length of Die block=2T_D+Max Length of extreme edge of shearing aperture+2T_D....................Eq 11

= (2 X 35) + 185.5 + (2 X 35)

= 326 mm

iii. Widthof Die block $= 2T_D +$ Height of the component $+ 2T_D$...Eq 12

= (2 X 35) + 91.32 + (2 X 35)

= 232 mm

For practical purpose, a land of 3 to 6 mm is provided, here we have provided 5 mm to accommodate reasonable land without change in clearance between pinch and die as shown in figure 5.

For practical purposes, a drop of 0.25^0 to 0.5^0 is provided to enable scraps to fall freely. We have provided a $\mathbf{0.5^0}$ dropas shown in figure 5.

The set assembly consists of a stripper plate, a punch holder plate, and a punch back plate which have the same length and breadth but varies in thickness depending on the practical work need. The following formulas are being used to determine the overall dimensions of the stripper plate, punch holder plate, and punch back plate.

Element	Length	Width	Thickness
Punch holder plate	326 mm	232 mm	=0.75× T_D =0.75 × 35 = 25 mm
Punch back plate	326 mm	232 mm	=0.5 × T_D =0.5 × 35 = 15 mm
Stripper plate	326 mm	232 mm	=0.75×T_D =0.75 × 35 = 25 mm

Table 2 Dimensions of Moving half elements

The set assembly also consists of a bottom and top plate which have the same length and breadth but varies in thickness depending on the practical work need. The following formulas are being used to determine the overall dimensions of bottom and top plate. Here, the diameter of the guide bush taken for the die set is Ø45 mm.

iv. Thickness of Bottom Plate $= 1.5 \times T_D$...Eq 13

= 1.5 X 35 = **55 mm**

v. Thickness of Bottom Plate $= 1.25 \times T_D$...Eq 14

= 1.5 X 35 = **45 mm**

vi. Length of Top and Bottom plate $= 1⅛\ D_B +$ Length of die plate $+ 1⅛\ D_B$............................Eq 15

$$= \frac{9}{8} \times 45 + 326 + \frac{9}{8} \times 45$$

= 50+326+ 50

= 476mm

vii. Width of Top and Bottom plate $= 1⅛\ D_B +$ Width of die plate $+ 1⅛\ D_B$..............................Eq 16

$$= \frac{9}{8} \times 45 + 232 + \frac{9}{8} \times 45$$

= 50 + 232 + 50 +40

= 332 mm

5.1.8. Determination of Length of punch

The punches are the most significant part of the press tool. This provides the component's actual profile. To easily shear the strip, the length of the punch must be maintained.

Punch Length = Thickess of Punch Plate + Thickess of Stripper Plate + Thickess of strip + Land + 5mm...Eq 17

= 25 + 25 + 1.5 + 5 +5

= 61 mm

Fits in double pitch progressive tool:

- Punch holder and Punches =H7/k6
- Punch and Stripper = H7/k6
- Guide Pillar and Guide bush = H7/g6
- Dowel and plate = H7/m6
- Dowel holes = H7/m6

5.2 STAGE 2: CAM BENDING TOOL

The bending force is being used to estimate the cam bending tool. The bending force is calculated using the component drawing. We chose the edge bending formula since the cam punch is designed to bend the component at the edge.

Edge Bending Force $= \frac{0.33}{L} \times S_{tu} \times W \times t^2$..Eq 18

Where, S_{tu} → Ultimate tensile strength of SS304 = 620 N/mm^2

W → Strip width

t → thickness of sheet

L → bending length

$$= \frac{0.33 \times 620 \times 20 \times 1.5 \times 1.5}{6} = 1535\ N$$

For two sides, 1535 × 2 = **3070 N** or **0.31 T**

Fits in cam bending tool:

- Guide Pillar and Guide bush = H7/g6
- Dowel and plate = H7/m6
- Dowel holes = H7/m6
- Guide rails and Cam bending punch = H7/g6
- Pressure pad and Bottom plate = H7/g6
- Between the bending punches = H7/g6

VI. 3D ASSEMBLY OF A SET OF TOOLS

To ensure that every surface is geometrically accurate, the 2D drawing is turned into a 3D model using Solid Works software 2020 and the assembly of the both tools are done after each modelling of parts. The 3D model of the assembly of Double pitch progressive tool and cam bending tool is shown in figure 6 and 7.

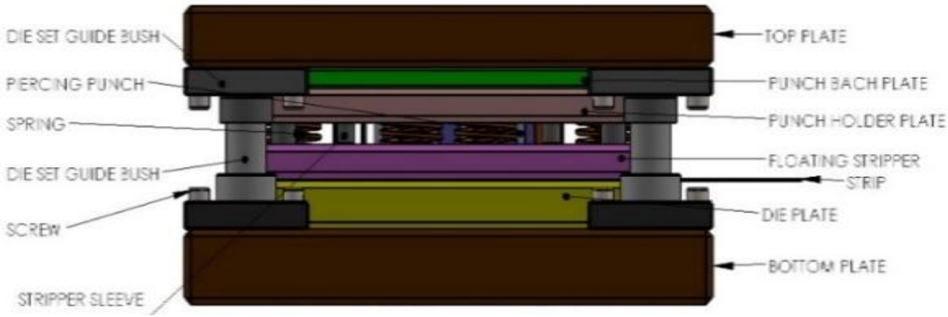

Figure 6 3D Model of Double pitch Progressive tool

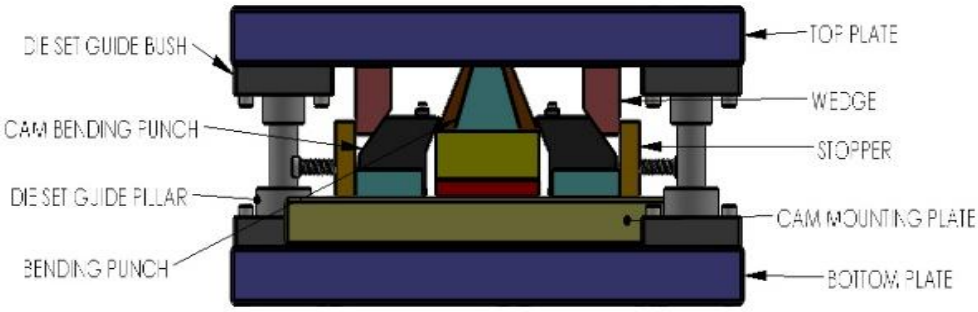

Figure 7 3D Model of Cam bending tool

VII. MATERIAL SELECTION

Press tools are typically produced from HCHCr steel alloys, that are high carbon steel alloys. But selection should be made depending on various aspects like as cost, strength, hardness, strain, and several other parameters. D2 or O1 materials are typically employed, with mild steel serving as a supporting plate.

D2 or High carbon high chromium steel is a cold work, high carbon, high chromium tool steel. D2 is a hardening alloy with such a high tool wear. It hardens on air cooling, providing in minimum distortion after heat treatment. It is used for long-run tooling applications that require high wear resistance, such as blanking or forming dies and thread rolling dies.

C	Si	Cr	Mo	V
1.5%	0.3%	12%	0.8%	0.9%

O1 or Oil hardened steel is an oil-hardening tool and die steel that may be used for a variety of applications like piercing and bending. Normal heat treatment care provides excellent hardening results and causes minor dimensional variations. O1 is abrasion resistant and tough enough for a wide range of tool and die applications.

C	Si	Cr	Mn	W
0.94%	0.3%	0.5%	1.2%	0.2%

Mild steel is used for the parts which are not involved in shearing operation but has high impact resistance. Excellent malleability with cold-forming capabilities. Heat treatment to enhance characteristics is not recommended.

ELEMEMT	MATERIAL	HARDNESS
Top Plate	Mild steel	-
Punch Back Plate	Oil Hardened steel / O1	48-50 HRC
Punch Holder Plate	Mild steel	-

Stripper Plate	Oil Hardened steel / O1	48-50 HRC
Die Plate	High Carbon High Chromium steel / D2	60-62 HRC
Guide Pillars & Bushes	Case Hardened Steel	54-60 HRC
Bottom Plate	Mild steel	-
Punches	High Carbon High Chromium steel / D2	60-62 HRC
Pilot punch	Oil Hardened steel / O1	48-50 HRC

Table 3 Materials in Double pitch Progressive tool

ELEMENT	MATERIAL	HARDNESS
`Top plate	Mild Steel	-
Pressure pad	Oil Hardened steel / O1	52-54 HRC
Cam mounting plate	Oil Hardened steel / O1	52-54 HRC
Bending punches	Oil Hardened steel / O1	52-54 HRC
Guide rail	Oil Hardened steel / O1	52-54 HRC
Wedge block	Oil Hardened steel / O1	52-54 HRC
Stopper	Oil Hardened steel / O1	52-54 HRC
Guide pillar and bush	Case Hardened Steel	56-60 HRC
Bottom plate	Mild Steel	-

Table 4 Materials in Cam bending tool

VIII. THEORETICAL ANALYSIS OF STRESS AND DISPLACEMENT

Theoretically, the component's stress value can be determined by dividing the component's force by its area. The condition of the component in the tool could be used to determine its displacement. The stress and displacement of the cam bending punch, bending punch, pressure pad, and wedge were calculated below.

a. Cam bending punch

$\text{Stress on Cam Bending Punch} = \frac{\text{Force}}{\text{Area}}$...Eq 19

Though Force =3070 N, we have two cam bending punches. As a result, for one cam bending punch is acting of load 1535 N. Area of cam bending punch is 59.3 mm ×20 mm.

$$= \frac{1535}{59.3 \times 20}$$

Stress on cam bending punch= 1.3 N/mm^2

The cam bending punch operates as a sliding support. As a conclusion, the displacement can be determined as

$\textbf{Displacement on Cam Bending Punch} = \frac{\mathbf{F \times L}}{\mathbf{A \times E}}$...Eq 20

Where, L = 42 mm

E = Young's Modulus = 2.1e5 N/mm^2

$$= \frac{\mathbf{1535 \times 42}}{\mathbf{59.3 \times 20 \times 2.1e5}}$$

Displacement on cam bending punch= 0.00026 mm

b. Bending punch

$$\text{Stress on Bending Punch} = \frac{\text{Force}}{\text{Area}}$$

Where, F = 3070 N

A = (40 × 20) × 2 + (49.7 × 20) = 2594 mm^2

$$= \frac{3070}{2594}$$

Stress on bending punch = 1.2 N/mm^2

The bending punch operates as a fixed at one end and load acts at other end. As a conclusion, the displacement can be determined as

$$\textbf{Displacement on Bending Punch} = \frac{\mathbf{F \times L}}{\mathbf{A \times E}}$$

Where, L = 35mm
E = Young's Modulus = 2.1e5 N/mm^2

$$= \frac{1535 \times 35}{2594 \times 2.1e5}$$

Displacement on bending punch = 0.0002 mm

c. Pressure pad

$$\text{Stress on Pressure Pad} = \frac{\text{Force}}{\text{Area}}$$

Where, F = 3070 N
A = 77.36 mm × 25 mm

$$= \frac{3070}{77.36 \times 25}$$

Stress on pressure pad = 1.6 N/mm^2

The pressure pad operates as a fixed supported and moves up and downwards. As a conclusion, the displacement can be determined as

$$\textbf{Displacement on Pressure Pad} = \frac{\mathbf{F \times L}}{\mathbf{A \times E}}$$

Where, L = 49.5 mm
E=Young's Modulus = 2.1e5 N/mm^2

$$= \frac{1535 \times 42}{77.36 \times 25 \times 2.1e5}$$

Displacement on pressure pad = 0.00037 mm

d. Wedge

$$\text{Stress on Wedge} = \frac{\text{Force}}{\text{Area}}$$

Though Force =3070 N, we have two wedge . As a result, for one wedge is 1535 N. Area of wedge is 20 mm × 27 mm.

$$= \frac{1535}{20 \times 27}$$

Stress on wedge = 2.8 N/mm^2

The wedge operates as a fixed at one end and load acts at other end. As a conclusion, the displacement can be determined as

$$\textbf{Displacement on Wedge} = \frac{\mathbf{F \times L}}{\mathbf{A \times E}}$$

Where, L = 36 mm
E = Young's Modulus = 2.1e5 N/mm^2

$$= \frac{1535 \times 42}{20 \times 27 \times 2.1e5}$$

Displacement on wedge = 0.0005 mm

IX. FEM ANALYSIS

In the FEM analysis, the parts subjected to load to bend the cable bracket include cam bending punch, bending punch, pressure pad and wedge. The analysis is performed to minimize practical problems and trails. Here, solid works software is used for FEM analysis.

The material used to the above material is oil hardened steel or O1. The material properties are shown below

Yield strength	1750 N/mm^2
Tensile strength	1930 N/mm^2
Elastic modulus:	210000 N/mm^2
Poisson's ratio	0.28
Mass density	7,610 kg/m^3
Shear modulus	79000 N/mm^2

Table 5 Physical properties of OHNS Material

Following the application of material, the previously stated parts are constrained by tool boundary conditions. After the parts are constrained under boundary conditions, the determined load is applied to it.

The mesh applied on the parts and it is stimulated. The properties of a solid mesh where the mesh information on the material are below.

Mesh type	Solid Mesh
Meshed Used:	Triangular mesh
Element Size	2 mm
Tolerance	0.1 mm
Mesh Quality	High

Table 6 Mesh properties on the parts

1. Cam bending punch

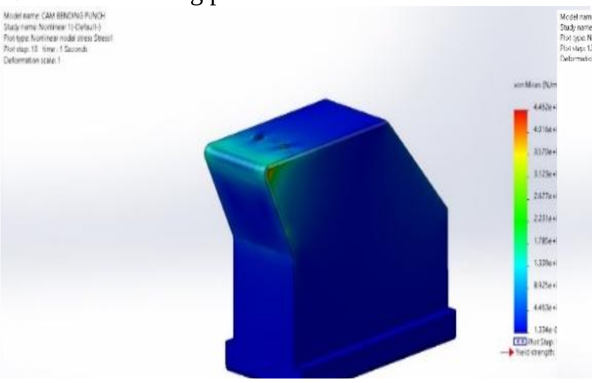

Figure 8 Stress results of Cam bending punch

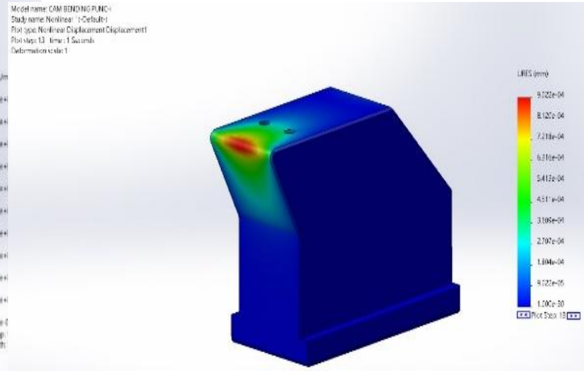

Figure 9 Displacement results of Cam bending

The maximum stress and maximum displacement on the cam bending punch is 40.46 N/mm^2 and 0.0009 mm respectively.

2. Bending punch

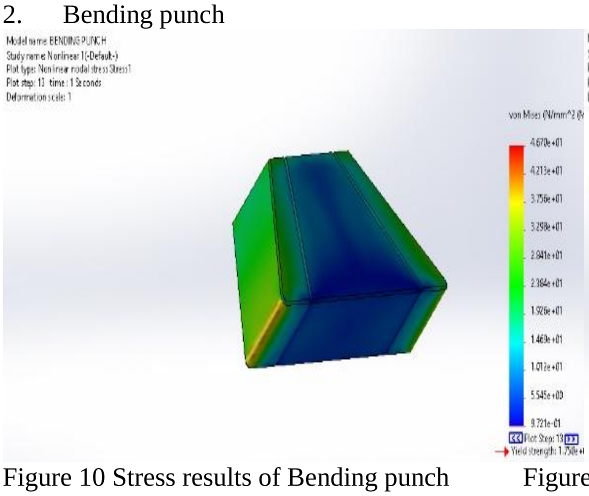

Figure 10 Stress results of Bending punch

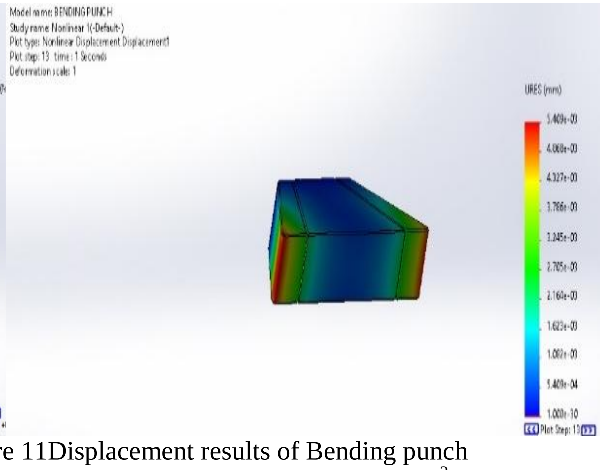

Figure 11Displacement results of Bending punch

The maximum stress and maximum displacement on the bending punch is 40.47 N/mm^2 and 0.005 mm respectively.

3. Pressure pad

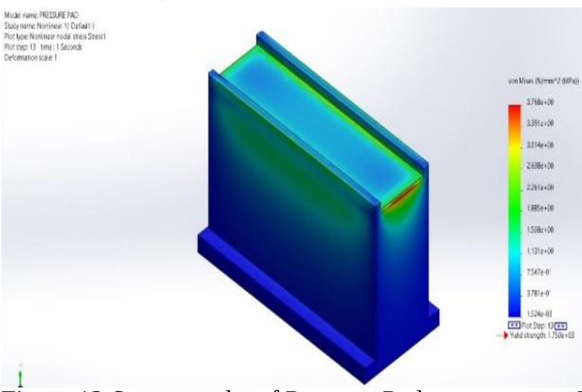

Figure 12 Stress results of Pressure Pad

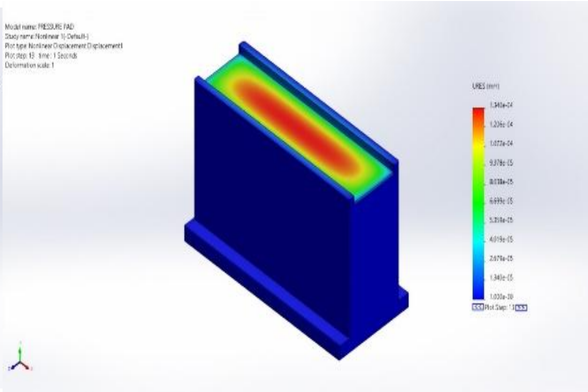

Figure 13 Displacement results of Pressure pad

The maximum stress and maximum displacement on the pressure pad is 3.76 N/mm^2 and 0.00013 mm respectively.

4. Wedge

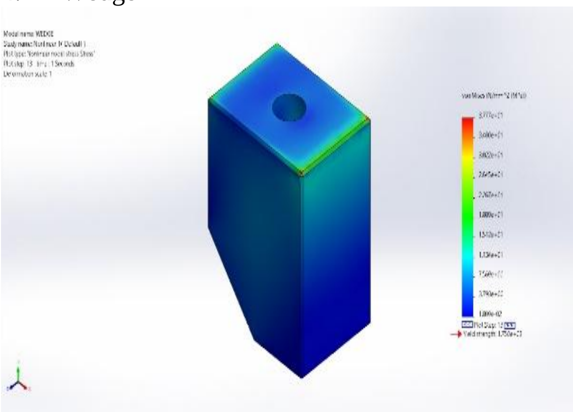

Figure 14 Stress results of Wedge

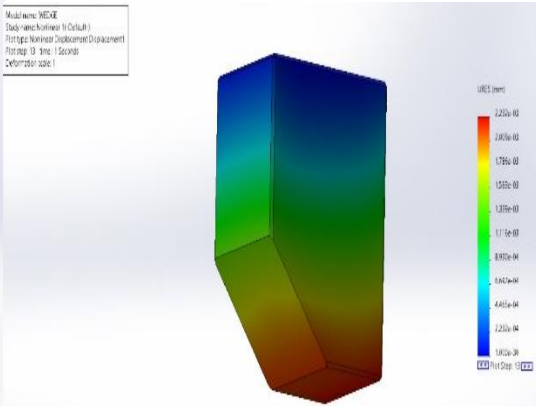

Figure 15 Displacement results of Wedge

The maximum stress and maximum displacement on the wedge is 30.77 N/mm^2 and 0.0022 mm respectively.

X. RESULTS AND DISCUSSION

	Description	Analysis results		Theoretical results	
		Stress (N/mm^2)	Displacement (mm)	Stress (N/mm^2)	Displacement (mm)
1	Cam bending punch	40.46	0.0009	1.3	0.00026
2	Bending punch	40.67	0.005	1.2	0.0002
3	Pressure pad	3.76	0.00013	1.6	0.0003
4	Wedge	30.77	0.0022	2.8	0.0005

Table 7 Comparison Between Solid works analysis and Theoretical results

We observed that,

a. The maximum stress by solid works analysis is little more compared to theoretical results.
b. The maximum displacement by solid works analysis are almost identical compared to theoretical results.
c. This will not affect the design of the above mentioned parts.

XI.CONCLUSION

The cable bracket's double pitch progressive tool and cam bending tool are developed following sheet metal die design principles. Solid works 2020 software is being used to model all of the parts and assemblies of the cable bracket's progressive tool and cam bending tool. Analysis on the cam bending punch, bending punch, pressure pad and wedge, where all are included in the bending of the cable bracket in solid works 2020 stimulation. On the above mentioned parts, the necessary materials, boundary conditions, and loads are applied. The observed results are as follows:

a. The stress values of the cam bending punch, bending punch, pressure pad, and wedge are found to be less than the yield stress of oil hardened steel.

b. It is observed that the deformation of the above mentioned components is well within the acceptable deformation.
c. The results from solid works stimulation and theoretical values are practically same.

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I am SUBRAHMANYA S doing M.Tech (Tool Engineering) in Government Tool Room and Training centre, Bengaluru. Currently done final semester project work on design and analysis of a set of press tool for cable bracket under the guidance of Dr. R NAGARAJA Ph.D, M.Tech in Tool Engineering.

PRINCIPAL/INTERNAL GUIDE

Dr. NAGARAJA R obtained his Bachelor's Degree in Mechanical Engineering from Mysore University, Chickmagalur in 1988. His M.Tech and Ph.D in tool Engineering was obtained from Visvevaraya Technological University, Belgaum. He started his career at Govt. Tool Room & Training Centre in the year 1992 and he has 31 years of experience in various departments such as Tool Design, Production, Process planning, Examination and various activities in training departments. He is presently working as Principal for PG/PD Studies in the same organization. Hisareas of interest are Tool design, Development, Validation and Natural Convection Heat transfer in Enclosures.