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Design and Analysis of Multipurpose Machine for the Productivity of Sheet Metal Process

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Strictly as per the compliance and regulations of:

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Alie Wube Dametew

Abstract- The global drive toward intermediate technology and sustainable development motivated the development of multipurpose machines for small-scale metal and ,metal product manufacturers are critical task. Improving the productivity of manufacturing process are significant tasks in all manufacturing engineers and manufacturing industries. Though this comparative advantage were achieved by the proper design, analysis and manufacturing of production machines. Consequently, the objective of this paper was to design and analyzed a multipurpose sheet metal machine to improve the production of sheet metal process and enhance competitiveness of the sector. Since, using mathematical methods this paper presents the design and development of multipurpose metal machine for small-scale enterprises, includes of three rollers; bending component, pipe bender, grooving/beading, twisting parts are rigid from base and machine frame of the machine. During the analysis sheet metal type, sheet metal thickness, forced induced in the machine and production capacity was considered as a parameters in this study However, the design and analysis result indicates that, there were significant improvement is observed in the productivity performance of using multipurpose sheet metal machine in terms of cost, production efficiency, production time, production capacity. The study also attempts to analyses the deflection, material types, the effects of stresses, wear-resistance, ultimate strength of the machine, thermal effect natural frequencies under subjected loads using mathematical analysis were done. In the conclusion, the study investigates that productivity analysis of sheet metal summery and conclusion of the work is also employed.

Keywords: multi-purpose, design and analysis, sheet metal forming, productivity, economic analysis.

I. Introduction and Background

Manufacturing is the process of converting raw material into semi or finished goods. There are many manufacturing process are found in production process. Depending on the manufacturing process industries use numerous machines to convert raw materials in to products. Since to increase the productivity and profit of the industries, manufacturing machines/equipment should be properly designed and manufactured. However the methodology of rapid machine design attempts to shorten design-to-manufacture time of production equipment by using advanced engineering tools such as Computer Aided Design systems (CAD), mathematical and Finite Element Analysis (FEA) during the conceptual design phase. Since identifying and to apply the best design concepts, overall development time can be shortened. However, in this paper the new approach to conceptual design can be applied at any phase during the concept generation, whether it is the design as a whole or a component in particular. As components are already part of the machine assembly, changes in their design are automatically updated in the whole skeleton. However, currently, number of sheet metal machines have been designed and fabricate to manufacture different sheet metal products. Main while most machines are performs single operations [P. S. Thakare 20012] and limited production performances. Those machines have their own disadvantages, till sheet metal machines have limited functions which could cause the productivity, the efficiency, versatility and competitiveness the sectors. however the main challenges that are seen on single purpose machines were higher machine cost, reduce sheet metal process productivity, poor production time and non-versatile. As the result multipurpose sheet metal machine need to be designed and fabricated in order to reduce machine cost and increase sheet metal process productivity and improve competitiveness of the sectors.

Since, the main objective of the study is to design analysis and fabricate multipurpose sheet metal machine so as to improve the productivity and competitiveness of sheet metal manufacturing sectors. The design and analysis process were includes select proper materials for production of multipurpose sheet metal machine, mathematical design, analysis and fabrication of multipurpose sheet metal machine, and productivity and economic analysis were considered.

a) Research Methodology

The study was conduct through literature review of research articles, books, magazines, manuals and electronic sources which are discuss related to design, manufacturing and mechanics of sheet metal production process. The analysis was done with mathematical methods and the analysis consider attempts to analyzed, investigate optimum capacity multipurpose sheet metal machine, so as to improve the productivity and comparativeness of sheet metal production sectors. The study attempts to analyses the deflection, material types, the effects of stresses, wear-resistance, ultimate strength of the machine, thermal

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effect natural frequencies under subjected loads using mathematical analysis were done. Since, the design, manufacturing productivity, economic analysis and evaluation of the project was done using the selected empirical analysis. in addition detail and assembly drawing of the machine is done using Solid work or Catiya Software). Finally the conclusion and recommendation of the study was done.

II. Literature Reviews

a) Introduction

Sheet metal fabrication plays an important role in the metal manufacturing world (Cloutier, 2000). Since, sheet metal process is metal formed into thin and flat pieces. It is one of the fundamental forms used in metalworking, and can be cut and bend into a variety of different shapes. Thicknesses can vary significantly, although extremely thin pieces of sheet metal would be considered to be foil or leaf, and pieces thicker than ¼ inch or a centimeter can be considered plate. [Xiaoyun Liao, 2007]. There are many special purposes machines used in this industry to make products. However, the proper selection of the machines depends upon the type of the work under-taken by the particular industry. Since, there are many examples of sheet metal work, which can be seen in our everyday lives. Although, the metals generally used for sheet metal work include black iron sheet, copper sheet, tin plate, aluminum plate, stainless sheet and brass sheet. on the other hand sheet metal is used and applicable in the production of materials ranging from tools, to hinges, automobiles, airplane wings, medical tables, roof for building etc. In terms of process type sheet metal fabrication ranges from deep drawing, stamping, forming, and hydro forming, to high-energy-rate forming (HERF) to create desired shapes (Cloutier, 2000). However, sheet metal is usually produced in sheet thickness less than 6mm by reducing the thickness of a long work piece by compressive forces applied through a set of rolls. This process is known as rolling [Xiaoyun Liao, 2007]. Due to the versatility, sheet metal process is a metal forming process which is spread throughout the world [Michael Lindgren 2009]. Since sheet metal forming is one of the most important semi finished and finished products used in the steel industry, and sheet metal forming technology is therefore an important engineering discipline within the area of mechanical engineering. Consequently, currently, sheet metal forming products have numerous applications, for example in manufacturing industries, buildings, airplanes and the automotive sector as well as infrastructure and domestic appliances, [Anas Muzamil 2009]. Since sheet metal process is a highly productive process and its use increases every year, [P. Groche 2006]. Although, the main feature of sheet metal is its ability to be formed and shaped by a variety of processes. Thus, each process does something different to the metal giving it a different shape or size [Xiaoyun Liao,2007]. The following are the common type of sheet metal process that performed using different type of sheet metal machines.

b) Shape Rolling

Shape rolling of sheet metal is the bending continually of the piece along a linear axis. This causes alteration of the original form of the sheet as it passes through a pathway of series of rollers. Such work tool as shape rolling machine is found to be very useful in manufacturing processes for used parts in various industries like inner and outer panels and stiffeners in automotive and agricultural industries, small metal workshops to roll round and conical profiles for stoves, cylinders (flue pipe, water pipes), basic machine elements with curved surfaces, buckets, bins, gear box cover, mud guards, drinkers and feeders for poultry, feed mixers etc. The machine rolls sheet metal up to 1.5 mm thick and 1m wide and rolls complete cylinders down to 75 mm diameter with 55 mm diameter rollers. Since considering high cost of tools and products in sheet rolling processes, detection and controlling factors for producing precise product are important. In most processes, geometry and configuration of rolling components could be obtained from the geometry of product at the end of loading. Therefore elastic recovery (known as spring back) formed part of the unloading process, and it is the most important factor in deviation of final products from desired geometry.

Also spring back is influenced by a combination of various process parameters such as tool shape and dimension, contact friction condition, material properties, thickness, were investigated and considered [Alie Wube Dametew & Tafesse Gebresenbet, 2016]. Since in this study the design and evaluation of a shape rolling machine with simple crank mechanism, higher roller diameter were analysis.

Cutting/Shearing: Sheet metal can be done in various ways from hand tools called tin snips up to very large powered shears. With the advances in technology, sheet metal cutting has turned to computers for precise cutting and shearing.

Punching Process: Punching is the process of using a machine to press a shape through a sheet of metal and into a die cutter to create that shape in the metal. These machines are single purpose type use, manual type, hydraulic, pneumatic, or electrical power to press the shape with enough force to cut the metal. [K. Abdel-Malek,2008]. Since punching can be better understood as pressing the material against a die with a huge force, this force pushes the material into the die shape and sheers of excess material. [K. Abdel-Malek, 2008]. Improved and effective punching machine is designed in these projects.

Bending Process: In engineering mechanics, bending (also known as flexure) characterizes the behavior of a slender structural element subjected to a lateral load. Since, a structural element subjected to bending is known as a beam. Although, closet rod sagging under the weight of clothes on clothes hangers is an example of a beam experiencing bending. [K. Abdel-Malek,2008].However, bending produces reactive forces inside a beam as the beam attempts to accommodate the flexural load; in the case of the beam, the material at the top of the beam is being compressed while the material at the bottom is being stretched [Xiaoyun Liao, 2007] .

Type of sheet bending

- *Edge bending* is also known as flanging in which one edge of the sheet is bent to 90 while the other end is restrained by the material itself and by the force of blank-holder and pad. The flange length can be easily changed and the bend angle can be controlled by the stroke position of the punch[Alie Wube Dametew & Tafesse Gebresenbet,2016].

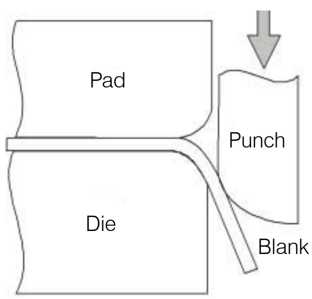

Figure 1: Edge bending

- *V-Bending* In V-bending, the clearance between punch and die is constant (equal to the thickness of sheet blank). It is used widely. The thickness of the sheet ranges from approximately 0.5 mm to 25 mm [Alie Wube Dametew & Tafesse Gebresenbet,2016].

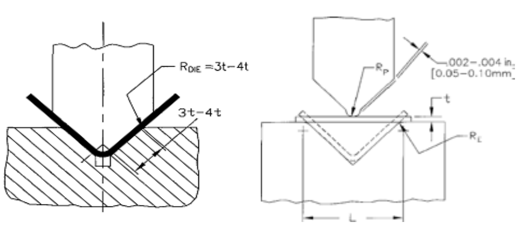

Figure 2: V – Bending

- *U - Bending* U-die bending is performed when two parallel bending axes are produced in the same operation. A backing pad is used to force the sheet contacting with the punch bottom. It requires about 30% of the bending force for the pad to press the sheet contacting the punch.

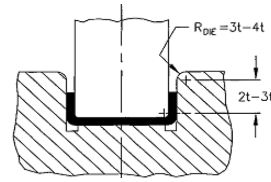

Figure 3: U – Bending

- *Coining* is a bending process in which the punch and the work piece bottom on the die and compressive stress is applied to the bending region to increase the amount of plastic deformation. This reduces the amount of spring-back. The inner radius of the work piece should be up to 0.75 of the material thickness. [Ivana Suchy, 2006 Hand book of die design]

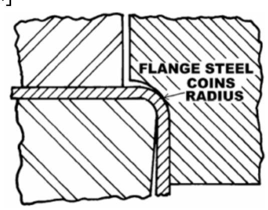

Figure 4: Coining bend

In this study comparative study of single and multipurpose sheet metal machine has been done. From the results, it is cleared that, productivity of three in one sheet metal machine is higher than that of single purpose. The result also shows that total cost for manufacturing metal sheets using three in one machine is 10.43% lower than single purpose machine. Although, the status of sheet Metal forming industry in Ethiopia is not going as it is expected, because of high cost of machine, problem of skill man power, [MOST,2013]. But ,in this time of globalization, the production of sheet forming process use single purpose machines which requires more money, time and labor for atomization of the process. All the machines for using in sheet metal forming process still a single purpose which results low productivity. In order to increase the productivity of sheet metal process, versatility of sheet forming machine is important issue for engineers. However this project deals with the design, analysis and fabrication of five in one sheet metal machine (multipurpose sheet metal machine) is done. Five in one sheet metal machines are perform number of operations within a single machine that is rolling, bending, cutting/ shearing, punching beading and circular shear operations are done within a single Machine.

III. Design and Analysis

a) Rolling Design considerations

i. *General design principles*

Following basic rolling operation on a sheet metal, components can be rolled to give it a definite shape. Bending of parts depends upon material

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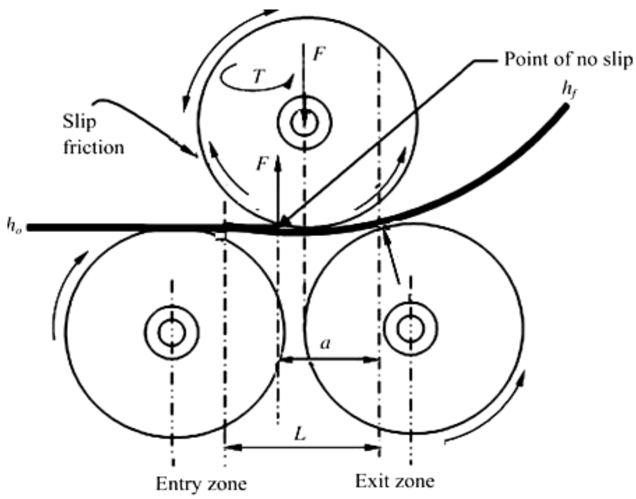
properties at the location of the roll bend. To achieve bending, the work material must be subjected to two major forces; frictional force which causes a no-slip action when metal and roller came in contact and a bending force acting against the forward speed and the torque applied to move the material (Figure).



e 5: Shape rolling mechanism

where, a = distance from exit zone to the no-slip point (assume $a = L/2$);

F = force applied to rollers; T = torque applied to rollers; L = roll gap; r = radius of rollers; μ = frictional force 0.4 Nm^{-1}; ho, hf = thickness of the sheet before and after time t.

At least two rollers were involved in flat rolling depending on the thickness and properties of material while three or multiple roller system is required in shape rolling. A work material under bending load is subjected to some form of residual stress and deformation as it bends. Since, materials at the outer bend radius undergo tensile plastic deformation while the material at the inner bend radius undergoes compressive plastic deformation. However, at least two rollers were involved in flat rolling depending on the thickness and properties of material while three or multiple roller system is required in shape rolling. A work material under bending load is subjected to some form of residual stress and deformation as it bends. Materials at the outer bend radius undergo tensile plastic deformation while the material at the inner bend radius undergoes compressive plastic deformation.

$$Lb = \theta(r + \kappa T) \quad (1)$$

where, Lb = bend allowance; θ = bend angle; r = bend radius to neutral axis; k = constant for material, for $r<2T$; $k=0.33$; for $r>2T$, $k=0.5$, T=thickness of material.

The strain on the outermost fibers of the bend is evaluated by Equation (2) given by Jack (2003):

$$\varepsilon = \frac{1}{\frac{2r}{T} + 1}$$

Maximum bending force is calculated by Equation (3) given below (Jack, 2003): $\frac{\sigma yield * LT2}{W} = \frac{\sigma UTS * LT2}{W}$, where, P = maximum bending load; k = constant for particular die from 0.3 to 0.7; σ_{yield} = yield stress for material; σ_{UTS} = ultimate tensile stress for the material; L = length of bend (along bend axis); w = distance between reaction supports When the rollers are in contact with the load, there is a frictional force existing, and an applied force, F and a slip between rollers and the load, which is not constant over the entire surface area of contact (Wagoner and Li,2007). An assumption of no reduction in size of material thickness during rolling makes, the thickness uniform i.e. and for small diameter bending, $a = L$. For large diameter bending $L>a$. Thus the maximum force is given by Hugh (2003) as , $\mu 2 * r = hf - ho = Maximum\ draft$ where, μ = frictional force 0.4 Nm-1; ho, hf = thickness of the sheet before and after time t. Analytical solutions of bending process have been presented by several researchers (Dongjuan *et al.,* 2007; Kim *et al.,* 2007; Wagoner and Li, 2007); however, for inverse analysis of spring back in free bending process, a state of plain strain and negligible shear deformation is assumed.

Since, considering the two strain components; the elastic strain (εe) and plastic strain (εp), the total axial strain (εx) can be written as, $\varepsilon x = \varepsilon e + \varepsilon p = \frac{(1-v2)\sigma x}{E} + \varepsilon p$

where, εx = total axial strain;εe = elastic strain; εp = plastic strain; E = Young's modulus, V=Poisson's ratio. Required bending moment (M) can be calculated as, $M = \int_A \sigma xydA$, where, A = area of shaft; σx = axial stress; y = radial arm in mm. Axial strain (εx) can be obtained as

$$= \frac{2}{t}\int_{y_c}^{t/2} \varepsilon_p\, dy + \frac{24y}{t^3}\int_{y_c}^{t/2} \varepsilon_p\, ydy + \frac{12M(1-V^2)y}{Ebt^3}$$

Assume axial stress (σx = I):can be obtained as

$$\sigma_x = \frac{2E}{(1-V^2)t}\int_{y_c}^{t/2} \varepsilon_p\, dy + \frac{24y}{t^3}\int_{y_c}^{t/2} \varepsilon_p\, ydy + \frac{12My}{bt^3} - \frac{E\varepsilon_p}{(1-V^2)}$$

Where, b and t are width and thickness of the sheet respectively; εe = elastic strain; ε_p = plastic strain; E = Young's modulus and V= Poisson's ratio. Bend radius after spring back can be written as

$$\rho' = \frac{1}{\frac{1}{\rho} - \frac{12M(1-V^2)}{bt^3E}}$$

Where, ρ and ρ' are bending radius before and after spring back respectively. Knowing the thickness and width of sheet plate and considering material's behavior, analysis of V-bending for various bending angle and radius become possible. The power required to roll the material is given by

$$P = force \times velocity = (Lwy_{ave}) \times (2\pi rn)$$

where, P = power in watts required to roll the sheet and n = speed in rmin-1. The spring back effect in bending is compensated by the following Equation (11) (Jack, 2003).

$$\frac{\sigma_{before}}{\sigma_{after}} = 4\frac{(r_{before}\,\sigma_{yield})^{3}}{ET} - 3\frac{r_{before}\,\sigma_{yield}}{ET} + 1$$

where, σ_{before} = σ_{after} = 1 for flat sheet; € = Bending strain.

b) Results and discussion of roller design

The result table below indicates the maximum bend radius obtainable as an approximate radius of roller for a typical material length of 500-630 mm is 2.5 mm. The largest measured diameter of complete cylinder the machine can handle (roll) is 184 mm using the given diameter of rollers. At reduced roller aperture of 2.5 mm, the material folded over and the radius of cylinder reduced. This result showed a remarkable improvement over Rob's report of 75 mm diameter.

Table 1: Design results of aperture, roll gap and the bend radius

Diameter of the Roller in mm	Material Length in mm	Roller aperture/before	Roller aperture/after bending in mm	Roll gap in mm	Bending radius in mm	Product shape
76	500-630	10mm	8	2	123	Semi-circle
		8mm	5.5	2.5	100	Full circle
		7.0mm	2.5	3	55	Double folded circle

The maximum bend radius when the aperture is closed is approximately the circumferential distance round the roller. The spring back effect on the rolled plate is noticeable as the rolling aperture decreases the bend radius considerably decrease. This effect can be increased by stretching out the material as it rolls or increasing the aperture width. Alternatively, variable roller diameters could be used when handling materials of varying length. Other dimensions can be obtained by varying the width of aperture until the bend radius approaches a straight line depending on the length of work material. The maximum width of material that can be handled with significant bend radius is,123 mm and the length is 500-630 mm.

Besides, the maximum width (machine capacity) of strip of material the machine could handle is 1,050 mm for tinplates and mild steel with thickness not exceeding 2 mm. Although, when the machine was used to roll mild steel plate of 2.5 mm thickness, the bearing blocks showed evidence of possible failure. The wooden bearing blocks could not support the bending stress exerted by the materials with higher thickness than 2 mm and as a result the machine usage can only be restricted to light gauge metal work and thus find use in tinsmith and welding workshops. Slip friction in rolling was eliminated between the rollers and the load by keeping the contact surfaces smooth and free of lubricants and dirt. Since, the average number of operators required to operate the rollers at a giver operation is shown in Table below with their average weights. Material thicker beyond 2 mm sheet thickness requires two operators while 3 mm thickness material could not be conveniently roll due to the required bending force and the strength of the material used in construction. The bearing capacity of wood cannot support such material thickness. The analysis result shows that, the average percentage acceptance of the machine by the artisan is 70.59% (n=24) indicating that the technology is acceptable. Fifteen percent of the welders and 40% of tinsmiths were well acquainted with the functions of the machine. A total of 73.53% (n=26) of the respondents are not acquainted (technology awareness) with the shape rolling technology while 26.47% (n=9) have a fair knowledge of the use of such machine in metal rolling.

Table 2: Table of performance tests Result

Material Type	Material Thickness in mm	No, operations	AV.Wet operator kg	Max width of work in mm	Length of the material
Aluminum sheet	3	1	65	1050	1200.00
	1.5	1	65	1050	1200.00
Galvanized sheet	1	1	71	525	800.00
	2	2	69.8	525	800.00
Milled steel sheet metal	1	1	65	1050	1200.00
	2	2	72	500	600.00
	3	2	72	300	400.00

c) *Design of pipe Bending Machine*

Companies considering the purchase of tube bending equipment are faced with an extensive set of alternatives. It is important that a potential purchaser research his needs and the various equipment available. The manufacturer who gives special attention to details will gain a significant edge over his competition. In this discussion we will investigate some of the more popular options available for tube bending, their benefits, limitations, cost and applications. Although, the design of pipe bending machine has undergone many changes, development and improvements over a period of time. Pipe bending requires mechanical force which acts on the pipe either directly or indirectly. This was done manually with the operator providing the effort required for bending the pipe. The major setback was the energy, time and effort expended in accomplishing the task. This means that the quality of bend would depend on the strength and skill of the operator. Though relatively cheaper, manual pipe bending falls short of dimensional accuracy and uniformity. Many versions of pipe bending machine have been developed aimed at eliminating human effort (www.paramount-roll.com). In one arrangement, the mechanical force required for bending is provided by a hydraulic ram powered by combustible fuel in an internal combustion engine, or by electricity. In this case, the hydraulic pump which pumps hydraulic into the ram is powered by an electric motor. By early 80’s, the development of mechanized pipe bending machine came into existence In general, the bending process uses mechanical force to push the pipe against a die: this way, the pipe is forced to get conformed to the shape of the die. In many cases, the end of the pipe is rolled and rotated around the die, while the pipe itself is firmly held in place (Fig.)

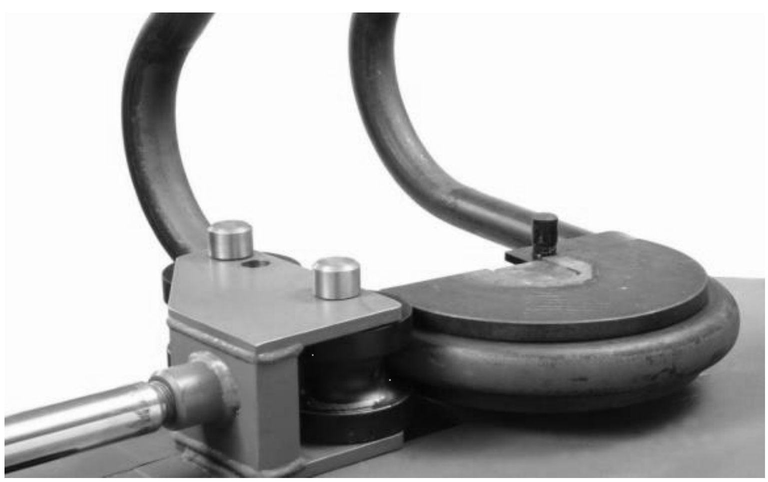


Figure 6: Pipe Bending Process

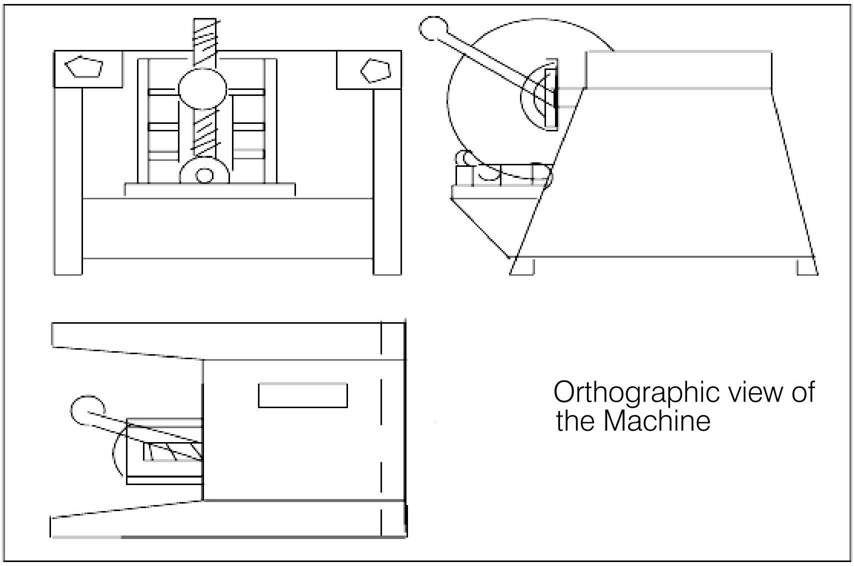


Figure 7: Orthographic View of the Machine

i. *Effort Required to Bend the Pipe*

The machine is considered as a lever with effort arm inclined, the load, W is provided by the pipe resistance to bend. The effort arm, x has both horizontal and vertical components, with the vertical component representing the active force. Since horizontal component, Fh = F sin θ and Vertical component, Fv = F cos θ Taking moment at the support reaction, R; , Fx cos θ = Wy; F = Wy / x cos θ; where F = Effort required to bend the pipe, Let θ = 30o and x = 5y (i.e. depending on the length of the effort arm), F = 23.3 x 103. y / 5y cos 30 = 5.5 kN; where W = average bending force of pipe, 23.3 x 103 N

ii. *Power Requirement*

A gradual application of effort will bend the pipe quite smoothly. This means that very small velocity will be required. An available motor capacity standard is therefore selected and reduced to appropriate speed output.

Choosing a motor of 1.5 kW; Power (P) = Force (F) x Velocity (V) ; Thus, V = P / F = 1500 / 5500 = 0.273 m/s

d) Speed Reduction (Spur Gear Design)

i. *Minimum number of teeth on the pinion*

Tp = 2Aw / G√ 1 + 1/G (1/G + 2) sin2θ – 1 (Shigley and Mischke, 1989), Where G = Gear ration / Velocity ratio; and θ = pressure angle, 20 , Aw = Fraction by which the standard addendum is multiplied, 1m for θ =20

Tp = 2 x 1 / 2 √ 1 + ½ (1/2 + 2) sin220 -1 = 14 2, Thus, we choose Tp = 18 from standard table (Shigley and Mischke, 1989), Number of teeth on the gear, Tg = 2Tp = 2 x 18 = 36

But centre distance between the gears, L = Dg/2 + Dp/2, Where Dg = Diameter of gear, and Dp = Diameter of pinion Dg / Dp = 2; Dg = 2Dp; L = 2Dp / 2 + Dp / 2 = 3/2 (Dp) = 1.5 Dp 64 = 1.5Dp; Dp = 43, Dp = m Tp ; where m is the module m = Dp / Tp = 43/18 = 2.4; Use standard value, m = 2.5

Pitch Circle Diameter of gear, Dg = 2Dp = 2 x 43 = 86 mm

Face Width of the Pinion and the Gear: _Pitch line velocity, V =πDpNp / 60 = π x 0.043 x 1410 / 60 = 3.17 m/s

For medium load shock condition and between 8~10 hours of service per day (Khurmi and Gupta, 2004); Service Factor, Cs = 1.54 and 2.369 for non-enclosed gears.

Tangential Tooth Load, WT = Cs (P/V) = 2.369 x 1500/3.20 = 1110.5N, Velocity Factor, Cv = 4.5 / 4.5 + V = 4.5 / 4.5 + 3.20 = 0.584 , Since the pinion and the gear are of same material, the pinion is weaker. For 20o involutes teeth;

Lewis Form Factor, Yp = 0.154 – (0.912 / Tp) = 0.154 – (0.912 / 18) = 0.1033 ,Thus, design tangential tooth load; WT = δWp x Cv x b x π x m x Yp, Where δWp is the safe stress of the pinion, 140 MPa and b is the face width of both pinion and gear. , WT = 140 x 0.584 x b xπ x 2.5 x 0.1033; b = 17mm, But minimum face width is taken as (9.54 ~ 12.5)m ; , Thus, let minimum face width, b = 9.54 x 2.5 = 24 mm

Power Transmitted: P = WT x V = 1110.50 x 3.2 = 355kW

e) Check for Static and Dynamic Loading

Flexible endurance limit for steel, δs = 252 (Khurmi and Gupta, 2004), Static load or endurance strength, Ws = δs x b x π x m x y ,Ws = 252 x 23.8 x π x 2.5 x 0.1033 = 4865.9N , Power that can be transmitted due to static loading is; Ps = 4865.9 x 2.9 = 14.3 kW, Since Ps (14.3 kW) is greater than P (1.5 kW), the design

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is safe from the standpoint of static loading. Also Dynamic Load, WD = WT + [2/V (bc + WT) / 2/V √bc + WT.But from table (Khurmi and Gupta, 2004), C = 228, and tooth error, e = 0.02 , WD = 1110.5 + [2/3.2 (23.8 x 228 + 1110.5) / (2/3.2√23.8 x 228 + 1110.5) = 3918.29N, Power that can be transmitted from this dynamic load, PD = WD x V = 3918.29 x 3.2 = 12.5 kW , Since PD (12.5kW) is greater than P (1.5kW), the design is safe from the standpoint of dynamic loading.

✓ *Design of Pinion Shaft of the pipe bender*

Load acting between the tooth surface; WN = WT / cos θ = 1110.5 / cos 20 = 1181.8N, Weight of pinion, Wp = 0.00118 x Tp x bm2 = 0.00118 x 18 x 23.8 x 2.52 = 3.16 N.

f) Resultant load acting on the pinion during bending

WR =√WN2 + WP2 + 2WNWPCos θ = √1181.82 + 3.162 + 2(1181.8 x 3.16 x Cos 20) = 1183.29 N

Bending Moment due to this resultant load;
MB = WR x Dp/2 = 1183.29 x 43/2 = 25440.74 N-mm

✓ *Twisting Moment on pinion;*

MT = WT x Dp /2 = 1110.5 x 43/2 = 23875.8N-mm
Equivalent Moment, ME = √MB2 + MT2 = √25440.742 + 23875.82 = 34889.6 N-mm
But equivalent twisting moment is given by;, TE = (π / 16) x 40 x Dp3 ; Dp3 = 34889.6 x 16 /π x 40 = 4440.5 ; DP = 16.5 mm , This shows that with Dp = 43 mm, the design is quite safe.
Diameter of pinion hub = 1.8 Dp = 1.8 x 43 = 77 mm
Length of hub = 1.25 Dp = 1.25 x 43 = 54 mm , Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 10 mm).

✓ *Design of Gear Shaft*

Normal load acting on the gear, WN = 1181.8N
Weight of gear, Wg = 0.00118 Tgbm2 = 0.00118 x 36 x 23.75 x 2 52 = 6.31 N
Resultant load acting on the gear; WR =√WN2 + Wg2 + 2WNWg Cos θ
WR = √1181.82 + 6.312 + 2(1181.8 x 6.31 x Cos 20 = 1189.73N
Bending moment due to resultant load, MB = 1189.73 x 86/2 = 51158.39 N-mm
Twisting moment, MT = 1110.5 x 86/2 = 47751.5 N-mm
Equivalent moment, ME = √MB2 + MT2 = √ 51158.392 + 47751.52 = 69981.3 N-mm
But equivalent twisting moment, TE = (π/16) x 40 x Dg3 = (π/16) x 40 x Dg3
Dg3 = 69981.3 x 16 / π x 40 = 8906.7; Dg (minimum value) = 21mm
This shows that with Dg = 86 mm, the design is quite safe.
Diameter of gear hub = 1.8 Dg = 1.8 x 86 = 154.8 mm
Length of gear hub = 1.25 Dg = 1.25 x 86 = 107.5 mm
Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 12 mm).

✓ Design of Worm Gear

The output of the gear is transmitted to the worm (Fig.3.2), such that Ng = Nw = 705 rev/min

Motor torque x speed of motor = Torque on gear shaft x speed of gear

10.16 x 147.65 = Ts x 73.83; Ts = 10.16 x 147.65 / 73.83 = 20.37 N-mm

Hence, transmitted power, P = TsV = Ts x 2 xπ x N/60 = 20.37 x 2 x π x 705 / 60 = 1.5 kW.

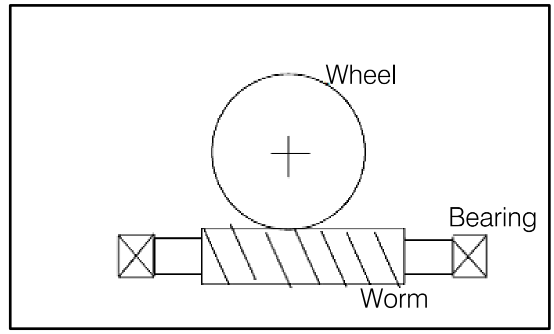

Figure 8: Worm Gear

- ✓ The minimum value of X/LN will correspond to;
- ✓ Cot^3 Φ = V.R = 20 Where; X = Lead = 200 mm, LN = Normal load, and V.R. = Velocity ratio
- ✓ Cot Φ = 2.71; Φ = 20.22o
- ✓ X / LN = ½ (1/sin Φ + V.R/cos Φ) = ½ [(1/sin 20.22) + (20/cos 20.22)] = 3.85
- ✓ LN = 200 / 3.85 = 51.92 N
- ✓ Axial load, LA = LN / cos Φ = 51.92 / cos 20.22 = 55.33 N
- ✓ For a V.R. of 20, the number of starts or threads on the worm, n = Tw = 2 (Allens et al, 1980)
- ✓ Thus, axial pitch of the thread on the worm;
- ✓ Pa = LA/2 = 55.33 / 2 = 27.67
- ✓ Module, m = Pa/π = 27.67/π = 8.8 (Take standard module = 8).
- ✓ Thus, axial pitch of the threads on the worm, PA = πm = π x 8 = 25.14
- ✓ Axial lead of the threads on the worm,
- ✓ La = Pa n = 25.14 x 2 = 50.28 mm
- ✓ Normal lead of the threads on the worm, Ln = La cos Φ = 50.28 x cos 20.22 = 47.2 mm.
- ✓ But centre distance = LN/2π (1/sin Φ + 1/cos Φ) = 47.2/2π (24.21) = 181.73 mm
- ✓ Let DW = Pitch circle diameter of the worm; then
- ✓ Tan Φ = L / πDW
- ✓ DW = La /π tan Φ = 50.28 /π tan 20.22 = 43.44 mm
- ✓ Since V.R. is 20 and the worm has double threads; Number of teeth on the worm gear Tg = 20 x 2 =40
- ✓ Face length of the worm (i.e. length of the threaded portion);
- ✓ LW = PA (4.5 + 0.02TW) = 25.14 (4.5 + 0.02 x 2) = 114.09 mm (this is normally increased by 25 ~ 30 for the feed mark) (Khurmi and Gupta, 2004);
- ✓ Thus LW = 144 mm

- ✓ Depth of tooth, h = 0.686PA = 0.686 x 25.14 = 17.24 mm
- ✓ Addendum, a = 0.313PA = 0.313 x 25.14 = 7.86 mm
- ✓ Outside diameter of worm,
- ✓ DOW = DW + 2a = 43.44 + 2 x 7.86 = 59.42 mm

g) Circle pitch diameter of worm gear

Dg = m Tg = 8 x 40 = 320 mm

Outside diameter of worm gear,

Dog = Dg + 1.0135PA = 320 + 1.0135 x 25.14 = 345.5 mm

Face width,

b = 2.38 PA + 6.5 mm = 2.38 x 25.14 + 6.5 = 66.31 mm

✓ *Design of Worm Shaft*

Torque acting on the worm gear shaft,

Tg = P x 60 / 2πNg

Considering 30% overload; Tg = 1.3 x 1500 x 60 / 2 xπ x 35.25 = 528.26 N-m

Torque acting on the worm shaft,

TW = Tg / V.R. x η where η = efficiency of worm gear

But η = Tan Φ / Tan (Φ + α) where α = angle of friction

Rubbing velocity, Vr =πDWNW / cos Φ = π x 0.0434 x 705 / cos 20.22 = 102.53m/mm.

Coefficient of friction, μ = 0.275 / Vr 0.25 = 0.275 / (102.53) 0.25 = 0.086

Thus, angle of friction, α = tan -1 μ = tan -1 0.086 = 4.92

Efficiency of worm gear, η = tan 20.22 / tan (20.22 + 4.92) = 0.78%

Thus, torque acting on the worm shaft, TW = 528.26 x 103 / 20 x 0.78 = 33.86 x 103 N-m

Tangential load on the worm, WT = Axial load acting on the worm gear

WT = 2 TW / DW = 2 x 33.86 x 103 / 43.44 = 1559.06 N

❖ *Axial load acting on the worm,*

WA = Tangential load on the worm gear

WA = 2Tg / Dg = 2 x 528.26 / 320 = 3301.63 N

Radial or separating force on the worm gear,

WR = WA tan Φ = 3301.65 x tan 20.22 = 1216.07 N

If distance between worm shaft bearing and worm gear, X = 400mm,

Then, bending moment due to WR in the vertical plane = WRX/4 = 1216.07 x 40 / 4 = 12160.7N-m

Bending moment due to axial force in the vertical plane = 3301.63 x 320/4 = 264130.4 N-m

Total Bending Moment in the vertical plane, MV = 12160.7 + 264130.4 = 276291.1 N-m.

Bending Moment in the horizontal plane, MH = WT x 40/4 = 1559.06 x 40/4 = 15590.6 N-m.

h) Resultant Bending Moment acting on the worm shaft, during bending process are

- ✓ MR = $\sqrt{MH^2 + MV^2}$
- ✓ MR = √276291.12 + 15590.62 = 276730.63 N-m
- ✓ Equivalent twisting moment on the worm shaft,
- ✓ MET = √ TW2 + MR2
- ✓ MET = √ (33.86 x 103)2 + (276730.63)2 = 278794.44 N-m
- ✓ But MET = π/16 x τ x DW2
- ✓ where DW = diameter of worm shaft; τ = allowable shear stress, 40 N/mm2
- ✓ DW = 32.86 mm (Use DW = 36mm).

Since, from the analysis we observed that, the pipe bending can be achieved in both ways (either in upward or downward direction) up to 4.25 mm pipe thickness could be bent manually depending on the operator's physical strength. During the bending process, the slight deviation from intended angle of bend was occurred due to spring back action of the pipe, which obviously reduces with decrease in angle of bend. However, wrinkles and bulging of the pipe during bending were noticed in pipes of lesser thickness. in this machine with a semi-circular bell, angles between 80° to 180° were obtainable, below which the bent pipe was observed to follow the bell's contour to give a U or C shape. Since, to obtain a lower bend angle, a different bell specially made for 45° is used. For pipes of lesser thickness, a mandrel should be introduced in order to prevent collapse. The provision for manual bending makes it possible to use the machine in rural areas were there may be no power supply.

i) Bending Design

Bending is a metal forming process in which a force is applied to a piece of sheet metal causing bending of it to an angle and forming the desired shape (Manar, 2013). While, the operation is typically performed on a machine called a press brake which can be manually or automatically operated. Though, to bend sheet metal, a bottom tool (die) is mounted on a lower, stationary beam (bed) and a top tool (punch) is mounted on a moving upper beam (ram). Since the design and analysis of this study is considered this points properly, to achieve optimum bending products. However the bending design consists and considers the f parameters given in the table below.

Table 3: Initial Conditions for Design

Maximum bending length	1800 mm
Maximum bending thickness	2mm
Tensile strength of sheet metal Mild steel	248 MPa
Clearance between folding beam and clamping beam	2 mm
Maximum folding angle	1050
Frame material	Structural Steel
Folding beam and clamping beam material	Machine Steel

j) Maximum Folding Force

The force required to perform folding depends on the strength, thickness, and length of the sheet metal (Groover, 2010). However, the maximum folding force can be estimated by means of the following equation: $F = \frac{Kbf\,(TS)wt2}{D}$,where, *Kbf* = 0.33,*TS* = 248 MPa, *w* = 1 800 mm, *t* = 2 mm; and *D* = 2 mm. Therefore; $F = 0.33 * 248 * 1800 * 2 * \frac{2}{2} = 294.6KN$

k) Clamping Beam Design

The clamping beam exerts a force that holds down the sheet metal onto the folding bed. The hold down force when performing folding operation is 50% the required folding force. Since it is applied across two ends of the machine. Therefore the clamping force is given by: Clamping force = 0.5 x folding force, Clamping force = 0.5 x 294.6 kN, Clamping force = 147.3 kN.

The clamping beam is designed such that it is welded onto side plates that are connected to a clamping mechanism. The clamping mechanisms are located on both sides of the clamping beam, but the clamping knob is only located on one end. The adjusting screws on the clamping mechanism of bending machine must resist the clamping force they are exposed to. Since, the load is shared equally on either side of the clamping mechanism, therefore is equal to half the clamping force that is 73.65 kN. Allowable stress levels to 75% of proof strength are to be used in the clamping mechanism bolts. The selected material for the clamping mechanism according to the Society of Automotive Engineers (SAE), is grade 4 with no head marking and proof strength of 65 ksi. Then the allowable stress is also given as σ_a =0.75*proof strength, Since σ_a=0.75*65000psi=48759psi.

In addition, the force applied on each side of the clamping mechanism is 73.65 kN = 16.55 klb

Therefore the required tensile area to which the force should act is:

$At = \frac{Load}{\sigma a} = \frac{16550\,lb}{\frac{48750\,lb}{in2}}$, Since, A_t+=0.339in^2. Thus, tensile stress area of 0.339 in2 requires a diameter of 7/8 inches, which is equivalent to 22.22 mm. Hence the diameter of the clamping mechanism column should be 22.22 mm with a course thread of 9 threads per inch.

l) Design of the Folding Beam

Figure shows the front view for the folding beam.

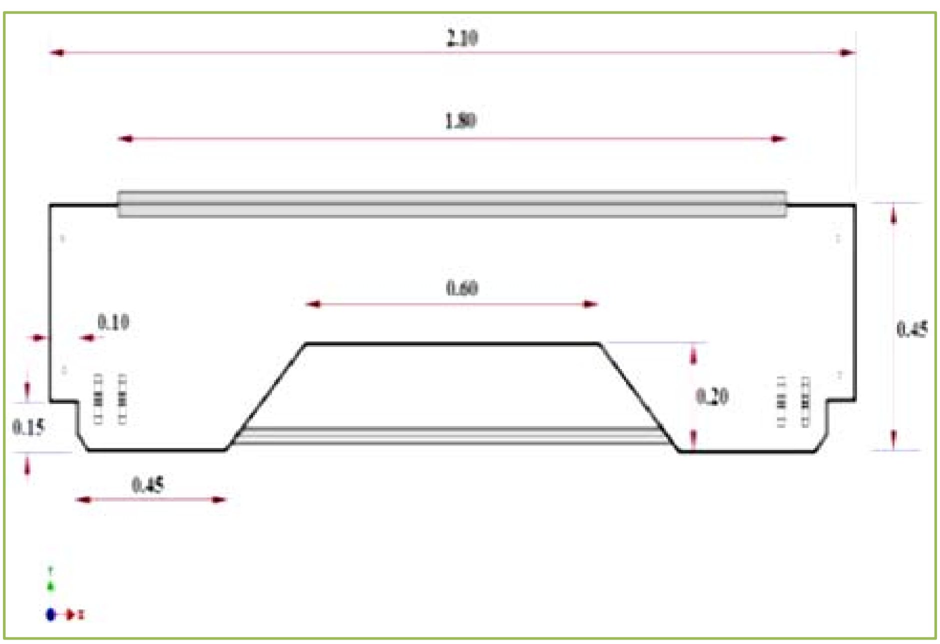

Figure 9: Folding Beam Front View

The weight of the folding beam is calculated using the formula below. Weight of folding beam = surface area x density x thickness Surface area, A, of folding beam: Surface area = 2.1 (0.45), $-0.5(0.2)(1+0.6)-2(0.15)$

Therefore *surface area*=0.755 m^2, Using machine steel of density 77 kN/m^3, if the thickness of the folding beam is therefore, the weight of the folding beam is: *W* = *Surface area* x *density* x *thickness*

W = 0.755 m2 x 77 kN/m^3 x *t*, ∴*W* =58.135 *t kN*, The beam is supported on its two ends and other force (including its weight) acting on the beam is the maximum required folding force of 294.6 kN acting uniformly across the length of the beam. Figure below also represents the loading induced on the beam during bending process.

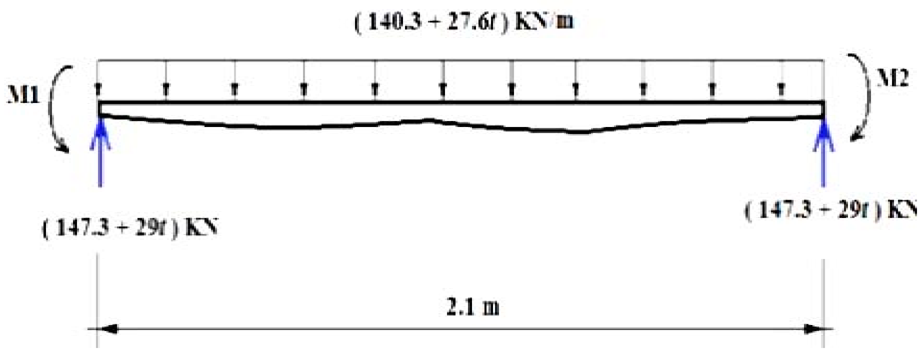

Figure 10: Loading on the Folding Beam

Total force acting on the beam = (294.6 +58.135t) kN.

$$Force\ acting\ per\ unit\ length = \frac{Total\ force\ action\ on\ thebeam}{Length\ of\ the\ beam}$$

$$Reaction\ at\ each\ support = \frac{Total\ force\ action\ on\ thebeam}{2}$$

$$Reation\ at\ each\ support = \frac{(294.6+58.135t)kN}{2},$$

Therefore reaction at each support =(147.3 +29 t)kN.

Taking moments and resolving forces at determined points along the folding beam and factoring a safety factor of n = 3, and an allowable stress of 350 Mpa, t,is found to have the following value; t = 0.015 or t = -0.015. Therefore the thickness of the folding beam is 15 mm.

IV. Productivity Analysis

Increase in productivity is the key factor for prosperity at all levels. It is the relationship between the result obtained and the factors employed to achieve the result. Productivity is the relationship between outputs to input. It is an indication of an enterprise capability. In case of the defined machine, the output highly depends upon the working skill of the employed persons. If he is having long experience of working over the machine then definitely his rate of making pipes would be higher than the rate of person who is new to machine. As per the definition of productivity, we have a simple relation for it and it is given by

$$Force\ acting\ per\ unit\ length = \frac{(294.6+58.135)kN}{2.1}$$

Thus, the force acting per unit length (140.3 +27.6 t)kN /m Reaction force at the beam supports also calculated as,

$$\text{Productivity} = \frac{\text{Output}}{\text{input}}$$

To directly compare the productivity in terms of capital required, we take the ratio of output in terms pipe Productivity=[(M anf. cost of material RS/ pieces x No. of material smanf, /day)][(Expenditure/material x No. of mant/day]

a) Productivity of Manually Operated Machine

Since time required for manufacturing the one pipe is 40 minutes, total number of pipes manufactured in a day is 12. Using above equation, Productivity of Manually operated machine is calculated as

$Productivity = 217 * \frac{12}{58.38*12} = 3.71$------For Manually operated machine

b) Productivity of Power Operated Machine

Since time required for manufacturing the one pipe is 10 minutes, total number of pipes manufactured in a day is 48. Thus, Productivity of Power operated machine is calculated as

$productivity = 217 * \frac{48}{26.07*48} = 83.46$-------------

For Power operated machine. Since our multipurpose machine is power operated , then it is productive. in a single operation , the productivity is double for manual operated machine, but our machine multipurpose, within this single machine performed more than four type of sheet forming process. Since the productivity is four times a single power operated machine. thus this multipurpose power operated sheet metal machine is efficient, effective and more extremely product type of machines.

V. Conclusion and Recommendation

a) *Conclusion*

In this study the design and analysis of multipurpose sheet metal machine is done using mathematical methods. During the analysis and design process different parameters were considered for improving the productivity and performance of sheet metal forming process. As well comparative study of manually operated multipurpose machine and power operated sheet bending machine has been done. the multipurpose machines are very efficient in rolling, bending, grooving and beading of metallic components. The machine is cost effective based on the materials of production and simplicity of the design of component parts. Operational mode meets the level of technical knowhow of the artisans. Also, productivity of both the machines has been calculated. From the results, it is cleared that, productivity of power operated sheet bending machine is higher than manual type. As a result a multipurpose power operated machine is much, much efficient, effective and productive on compared to manual operated machine. Since the design and analysis of this study was meet the objectives properly. Since, sheet metal manufacturing industries improve their financial performance and productivity by using these type of multipurpose machines rather than single purpose machine.

b) *Recommendation*

This multi type machines have numerous advantages including, optimum machine cost, high production capacity and productive, efficient and effective and contributes the comparativeness of the sectors. Since, sheet metal manufactures could be use such type of machines to improve their performance and productivity. Further recommend that, researchers further study to add the function of such type of machines, hence this used as a base for the researchers.

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