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Research Paper

DESIGN OF A DUAL OPERATING MODE SHEET FOLDING MACHINE

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A sheet folding machine that can be operated through hydraulics by two hydraulic cylinders or manually (with the cylinders disengaged) was designed. The design need emanated from the strained national electrical grid system that has recently seen industrialists and households in Zimbabwe experiencing major power cuts. The machine enables manufacturers to schedule heavier jobs during periods when power supply is up and lighter jobs during power cut periods hence run their workshops throughout the daily production shifts. The two hydraulic cylinders can be disengaged from the machine's folding beam so that manual operation can be done through a manual clamping lever system. The folding force at full capacity is 294.6 KN (29.46 Ton), total bending length of 1.8 m and working height of 1 m. The folding force decreases significantly in manual operating mode to 500 N, considering that on average an operate can manually exert that force. A student version of Simulation X 3.5 was used to simulate the hydraulic operation of the machine.

Keywords: Sheet metal folding, Folding machine, Sheet metal bending, Press brake

INTRODUCTION

Sheet metal bending and folding accounts for the production of a wide range of consumer durable goods. The demand of goods that fully or at least partly comprise of bent sheet metal parts promises to remain high. Typical products made from sheet metal folding include enclosures, electric boxes, casings for electrical and electronic gadgets, trays, lids, troughs, air ducts, and chimneys. The paper will articulate the design of the components of the folding machine which comprise of the folding beam, the clamping beam, the hydraulic system, selection of a pump to power the hydraulic system, design of the hydraulic cylinder connections such that they can be easily disengaged from hydraulic to manual mode. Currently the bulk of available sheet metal folding machines are operated manually whilst few hydraulic operated ones

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exist. The operation of hydraulic folding machines is greatly affected by the power cuts hence the need for a machine that operates both on hydraulic and manual mode. Designing a dual operation mode machine gives the manufacturers more flexibility on limited resources as compared to buying two machines with different operating modes so that the other can be shelved when there is no power supply. To deal with this problem, this article focuses on the design of dual-mode sheet metal folding machine that has the ability to be disengaged from hydraulic mode to manual mode in the shortest period of time. With the growth of Small and Medium scale Enterprises (SMEs) worldwide, the folding machine can be used to produce a wide range of products in small factories for local market as well as foreign markets with low power consumption since the operators can choose to switch into the manual mode even during periods when there is power supply.

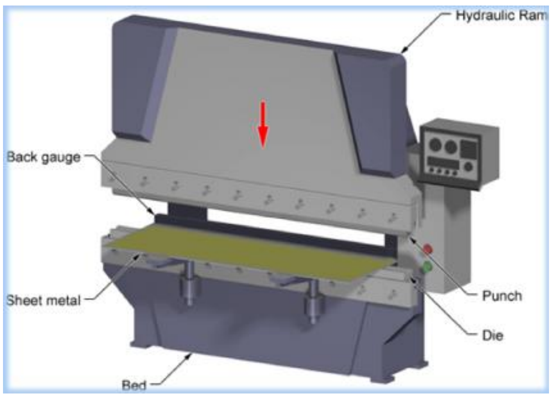
CLASSIFICATION OF SHEET METAL BENDING PROCESSES

There are various sheet metal processing operations, for instance, laser cutting and bending, punching, deep drawing and redrawing, bending, incremental forming, shearing and blanking, stretch forming, rubber hydroforming, spinning and explosive forming (Groover, 2010). Bending along a straight line is the most common of all sheet forming processes; it can be done in various ways such as forming along the complete bend in a die, or by wiping, folding or flanging in special machines, or sliding the sheet over a radius in a die (Marciniak, 2002). The terms folding and bending are loosely used in the sheet-metal industry and largely interchangeable in common parlance, to be precise, the term 'folding' refers to sharp corners with a minimum bend radius and the term 'bending' refers to deflections of relatively large corner radii. Folding and bending involve the deformation of material along a straight line in two dimensions only (Timings, 2008).

Bending by Press Brakes

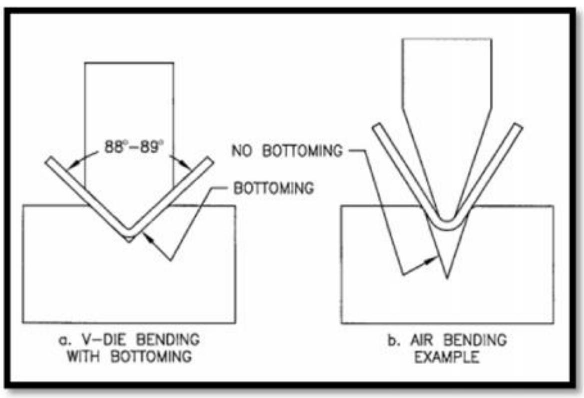
Bending is a metal forming process in which a force is applied to a piece of sheet metal causing bending of it to an angle and forming the desired shape (Manar, 2013). The process is typically performed on a machine called a press brake which can be manually or automatically operated. To bend sheet metal, 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) (Simons, 2006). The opposite configuration is also possible. Bending produces a V-shape, U-shape, or channel shape along a straight axis in ductile materials, most commonly sheet metal. Commonly used equipment include box and pan brakes, brake presses, and other specialized machine presses. A typical press brake is illustrated in Figure 1.

Figure 1: Press Brake



V-die bending can be used in two different ways: for "air bending" or for "bottoming". In air bending the punch stops a certain distance above the bottom. The die set can be used for bending at any angle larger than 85°. The bottoming die bends the sheet metal at an angle of the die, which may be 90° or any other angle. Both types of V-die bending allow for over bending, which means that the bends under 90° can be produced. Figure 2 shows V-bending die sets.

Figure 2: V-Die Bending, Air Bending, and Bottoming



Source: Suchy (2006)

Sheet Metal Folding Machines

Sheet metal folding process is performed on sheet metal folding machine. The machine consists of a clamping beam that holds down the sheet metal work, and a folding beam that performs the folding operation. The clamping beam consists of a detachable clamping blade and the folding beam of a hard and removable segment, this allows for damaged segments to be replaced. Another feature on the folding machine is a back gauge that allows for highest repeatability of work.

Two types of sheet metal folding machines are available, one that is hydraulic powered and one that operates manually. Manual folding machines however have some disadvantages in that they are not conducive to higher production rates, quality or repeatability; however they are suitable for low scheduling light workloads. Hydraulic powered sheet metal folding machines counteract the disadvantages of manually operated sheet metal folding machines, but they have a limitation in that they are affected by power cuts.

Hence the design on a folding machine that operates both on hydraulic and manual mode will minimize the impact of power cuts on the production rate and also at the same time improving on quality work through scheduling of light workloads during periods of power cuts and scheduling of other demanding workloads during periods in which electricity is available.

Hydraulic Power System

Sheet metal working machines can be classified according to energy supply. Five categories can be identified as follows;

Mechanical: Where the work force is supplied by some mechanical means such as a cam or lever.

Hydraulic: These utilize the pressure of water or other fluid media.

Steam: They use pressurized steam.

Electromagnetic: Uses electromagnetic force.

Hydraulic power system was chosen for the folding machine because of the following advantages over other methods of power transmission (Dawei, 2008):

- Simpler design – In most cases, a few pre-engineered components will replace complicated mechanical linkages.
- Flexibility – Hydraulic components can be located with considerable flexibility. Pipes and hoses instead of mechanical elements virtually eliminate location problems.
- Smoothness – Hydraulic systems are smooth and quiet in operation. Vibration is kept to a minimum.
- Control – Control of a wide range of speed and forces is easily possible.
- Cost – High efficiency with minimum friction loss keeps the cost of a power transmission at a minimum.
- Overload protection – Automatic valves guard the system against a breakdown from overloading.

The main disadvantage of a hydraulic system is maintaining the precision parts when they are exposed to bad climates and dirty atmospheres, hence protection against rust, corrosion, dirt, oil deterioration, and other adverse environmental conditions is very important. Disposal of the hydraulic fluid is also a threat to the environment.

DESIGN OF THE FOLDING MACHINE COMPONENTS

Detailed design calculations for sizing the folding machine components are carried out in this section. The initial conditions for basing the design on are given in Table 1.

Maximum Folding Force

The force required to perform folding depends on the strength, thickness, and length of the sheet metal (Groover, 2010). The maximum

Table 1: Initial Conditions

Parameter	Value
Maximum bending length	1800 mm
Maximum bending thickness	2 mm
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

folding force can be estimated by means of the following equation:

$$F = \frac{K_{bf}(TS)wt^2}{D} \quad ...(1)$$

where,

$K_{bf} = 0.33$

$TS = 248$ MPa

$w = 1\,800$ mm

$t = 2$ mm; and

$D = 2$ mm.

Therefore;

$$F = \frac{0.33 \times (248) \times 1800 \times 2^2}{2}$$

$F = 294.6$ kN

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 as shown in Figure 3.

Figure 3: Clamping Beam and the Clamping Mechanism

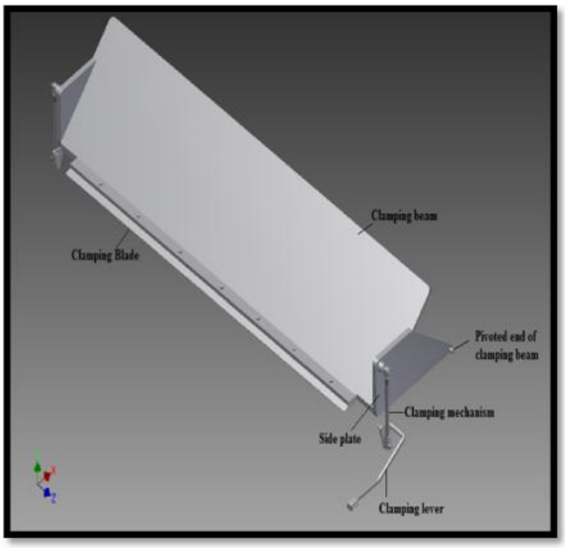
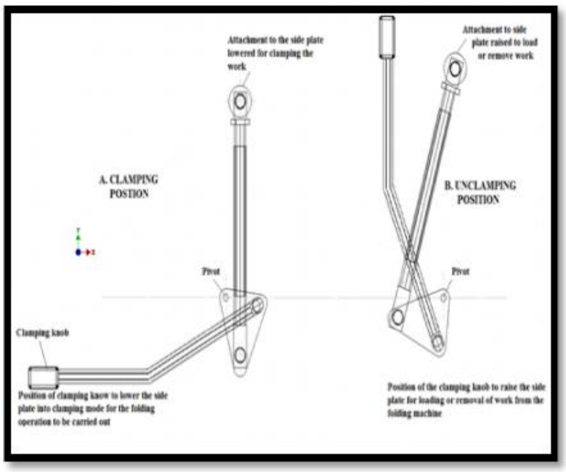


Figure 4: Operation of the Clamping Mechanism



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 must resist the clamping force they are exposed to. Operation of the clamping mechanism is illustrated in Figure 4.

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:

$$\pm_a = 0.75 \text{ x } \textit{proof strength} \quad \text{...(2)}$$

$$\pm_a = 0.75 \text{ x } 65000 \text{ psi}$$

$$\pm_a = 48759 \text{ psi}$$

The force 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:

$$A_t = \frac{Load}{\pm_a} \quad \text{...(3)}$$

$$A_t = \frac{16550\ lb}{48750\ lb/in^2}$$

$$\therefore A_t = 0.339\ in^2$$

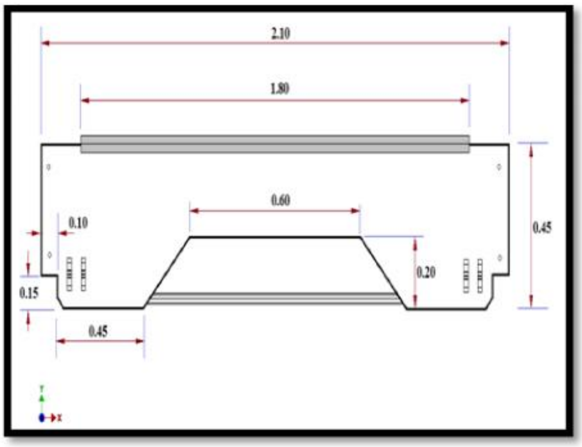
Tensile stress area of 0.339 in^2 requires a diameter of 7/8 inches, which is equivalent to 22.22 mm.

Hence the diameter of the clamping mechanism column should be 22.22 mm with a course thread of 9 threads per inch.

Design of the Folding Beam

Figure 5 shows the front view for the folding beam.

Figure 5: 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* ...(4)

Surface area, *A*, of folding beam:

$$\textit{Surface area} = 2.1\,(0.45) - \frac{1}{2}(0.20)\,(1+0.6) - 2(0.15)$$

$$\therefore \textit{Surface area} = 0.755\ m^2$$

Using machine steel of density 77 kN/m^3, if the thickness of the folding beam is t, therefore the weight of the folding beam is:

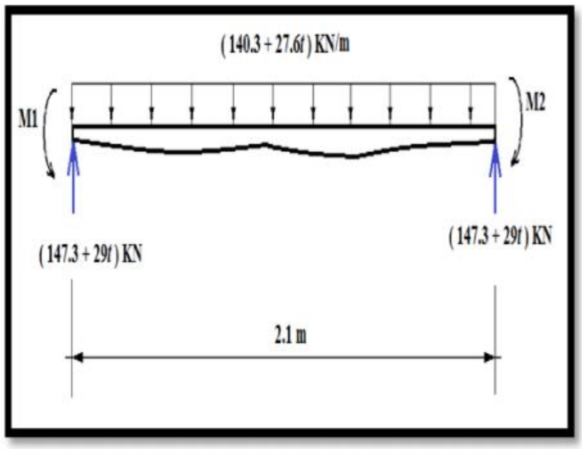
W = *Surface area* x *density* x *thickness*

$W = 0.755\ m^2 \times 77\ kN/m^3 \times t$

$\therefore W = 58.135\ t\ kN$

The beam is supported on its two ends and the 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 6 represents the loading on the beam.

Figure 6: Loading on the Folding Beam



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

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

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

$$\therefore \textit{Force acting per unit length} = (140.3 + 27.6\ t)\ kN/m$$

Reaction force at the beam supports:

$$\textit{Reaction at each support} = \frac{\textit{Total force acting on the beam}}{2}$$

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

$$\therefore \textit{Reaction at each support} = (147.3 + 29\ t)\ kN$$

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;

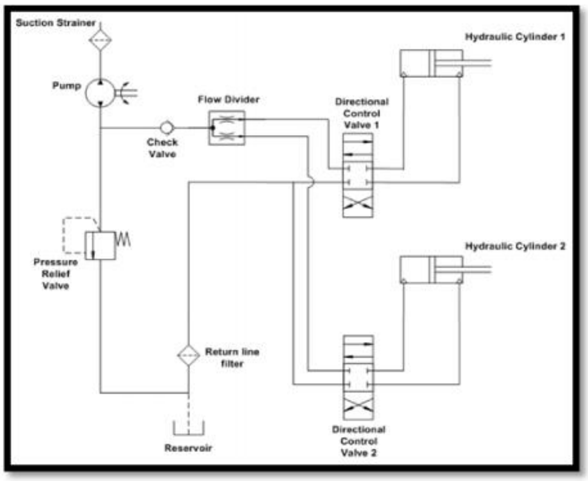
$t = 0.015$ or $t = -0.015$

Therefore the thickness of the folding beam is 15 mm.

Design of the Hydraulic System

Figure 7 shows the design of the hydraulic circuit system of the metal folding machine.

Figure 7: Circuit Diagram of Hydraulic System

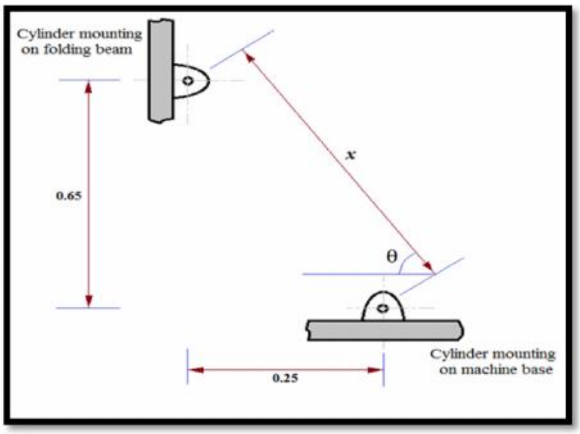


The folding force is evenly distributed between the two hydraulic cylinders.

$$\therefore \text{Load on each hydraulic cylinder} = 147.3\ kN$$

The axial load on each hydraulic cylinder can be determined using the angle of inclination of the cylinders. The angle of inclination, θ, is determined using Figure 8.

Figure 8: Illustration for Determining the Angle of Inclination of the Cylinders



$$length,\ x = \sqrt{0.65^2 + 0.25^2}$$

$$\therefore x = 0.69\ m$$

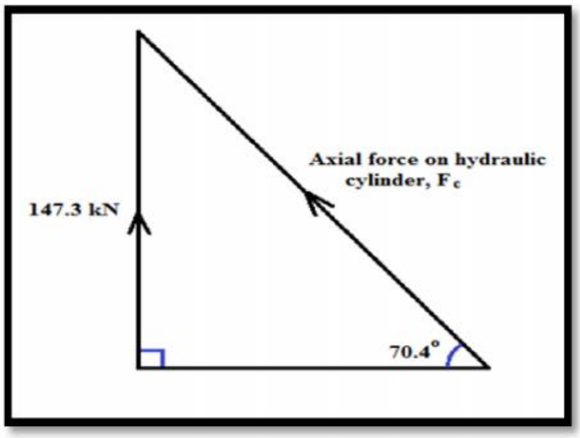
$$\frac{\sin\theta}{0.65} = \frac{\sin 90^o}{x}$$

$$\theta = \sin^{-1}\left(\frac{0.65\sin 90^o}{0.69}\right)$$

$$\theta = 70.4^o$$

The axial force, F_c, on the hydraulic cylinder required to generate the folding force is calculated with reference to Figure 9.

Figure 9: Axial Force on the Hydraulic Cylinder



$$\frac{F_c}{\sin 90^o} = \frac{147.3\ kN}{\sin 70.4^o}$$

$$F_c = 156.4\ kN$$

The load will be evenly distributed between the two hydraulic cylinders hence the force on each cylinder is:

$$\text{Force on each cylinder} = \frac{156.4\ kN}{2}$$

$$\text{Force on each cylinder} = 78.2\ kN$$

For the hydraulic system an operating pressure of 150 bars is selected. The effective piston area, A_{eff}, is calculated below using Equation 5.

$$A_{eff} = \frac{Axial\ force\ on\ each\ hydraulic\ cylinder}{Operating\ pressure} \qquad ...(5)$$

$$A_{eff} = \frac{78.2\ kN}{150 \times 10^5}$$

$$\therefore A_{eff} = 5.21 \times 10^{-3} m^2$$

The effective piston area is used to calculate the hydraulic cylinder bore diameter, D, as follows:

$$A_{eff} = \frac{\text{®} D^2}{4} \qquad ...(6)$$

$$D = \sqrt{\frac{4 \times 5.21 \times 10^{-3}}{\text{®}}}$$

$$\therefore D = 0.081\ m$$

Using preferred cylinder sizes and piston diameters, we determine that the appropriate piston diameter, d, is 56 mm.

The annulus area, A_a, is calculated using Equation 7.

$$A_a = A_{eff} - \frac{\text{®} d^2}{4} \qquad ...(7)$$

$$A_a = 5.21 \times 10^{-3} - \frac{\text{®} \times (56 \times 10^{-3})^2}{4}$$

$$A_a = 2.75 \times 10^{-3} m^2$$

Using the annulus area, the retracting force, F_r, is calculated using Equation 8.

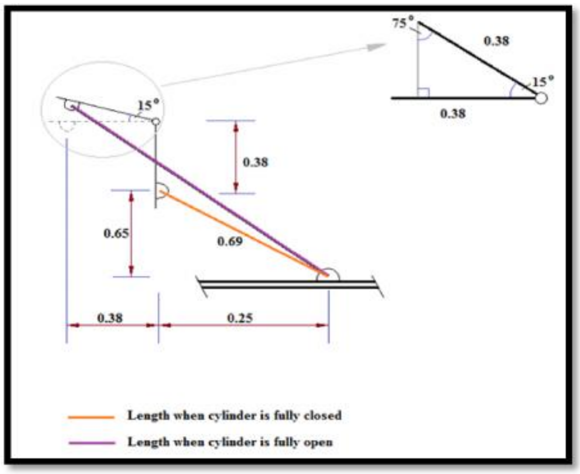
$$F_r = P A_a \qquad ...(8)$$

$$F_r = 150 \times 10^5 \times 2.75 \times 10^{-3}$$

$$\therefore F_r = 41.25\ kN$$

The stroke of the hydraulic cylinders is determined with reference to Figure 10.

Figure 10: Illustration for Determining the Stroke of the Hydraulic Cylinder



When the cylinder is fully open:

The total horizontal length is = 0.25 + 0.38 sin 75

The total vertical length is = 0.65 + 0.38 + 0.38 sin 15

Therefore the length of the fully open cylinder, l_o, is:

$$l_o = \sqrt{(0.25 + 0.38 \sin 75)^2 + (0.65 + 0.38 + 0.38 \sin 15)^2}$$

l_o = 1.28 *m*

The stroke of the cylinder is calculated as follows:

Stroke = Fully open length – fully closed length ...(9)

Stroke = 1.28 *m* – 0.69 *m*

Stroke = 0.59 *m*

Using the stroke obtained above the velocity, v_e, of the hydraulic cylinder during extension of a period of 4 seconds can be determined as follows:

$$V_e = \frac{Stroke}{Cylinder\ \ extension\ \ time} \quad ...(10)$$

$$V_e = \frac{0.59\ m}{4\ s}$$

$$V_e = 0.148\ m/s$$

The flow rate, Q_e, during extension is calculated as:

$$Q_e = A_{eff} \text{ x } velocity\ during\ extension \quad ...(11)$$

$$Q_e = 5.21 \text{ x } 10^{-3}\text{m}^2 \text{ x } 0.148\ m/s$$

$$Q_e = 7.71 \text{ x } 10^{-4} m^3/s$$

$$Q_e = 0.77\ litres/s$$

$$Q_e = \frac{0.77 \times 60}{3.785}\ GPM$$

$$Q_e = 12.21\ GPM$$

The same flow rate is used to retract the cylinder; hence the retraction velocity, v_r, is determined as follows:

$$V_r = \frac{Q_e}{A_a} \quad ...(12)$$

$$V_r = \frac{7.71 \times 10^{-4}}{2.75 \times 10^{-3}}$$

$$V_r = 0.28\ m/s$$

Therefore the retraction time, t_r, is

$$t_r = \frac{Stroke}{v_r} \quad ...(13)$$

$$t_r = \frac{0.59\ m}{0.28\ m/s}$$

$$t_r = 2.1\ s$$

Hydraulic Pump Sizing

The hydraulic pump supplies the pressurized fluid for cylinder actuation. A vane pump is to be used to power up the hydraulic system. The following calculations will be useful in sizing the desired pumping unit.

Taking a volumetric efficiency, n_v, of 90% and speed rating of 1200 rpm, we can obtain the theoretical pump delivery.

$$\eta_v = \frac{Actual\ \ pump\ \ delivery}{Theoratical\ \ pump\ \ delivery} \quad ...(14)$$

$$Theoretical\ \ pump\ delivery = \frac{Actual\ pump\ delivery}{\eta_v}$$

$$Theoratical\ \ pump\ \ delivery = \frac{7.71 \times 10^{-4} m^3/s}{0.9}$$

$$Theoratical\ pump\ delivery = 8.57 \text{ x } 10^{-4}\ m^3/s$$

$$Theoratical\ pump\ delivery = 51.4\ litres/min$$

The pump displacement per revolution, D_p, is calculated as follows:

$$D_p = \frac{Theoratical\ \ pump\ \ delivery \times 60}{Pump\ \ speed\ \ rating} \quad ...(15)$$

$$D_p = \frac{8.57 \times 10^{-4} m^3/s \times 60}{1200\ rpm}$$

$$D_p = 4.29 \text{ x } 10^{-5} m^3$$

From the calculations above a hydraulic vane pump with a rating of 1200 rpm, a displacement of 4.29×10^{-5} m^3 and a flow capacity of 51.4 litres/min (13.58 GPM) is selected.

Reservoir Design

The most suitable hydraulic tank volume is two to three times the pump capacity. Selecting a factor of 2.5, the capacity of the hydraulic reservoir is calculated as follows:

$$Reservoir\ volume = 2.5 \text{ x } pump\ capacity \quad ...(16)$$

Reservoir volume = 2.5 x 51.4 *litres*

Reservoir volume = 128.5 *litres* = 0.1285 m^3

A reservoir should be high and narrow rather than shallow and broad. Therefore, a rectangular prism shaped reservoir with a length, *l*, of 0.8 *m* and width, *w*, of 0.4 *m* is chosen.

The height, *h*, of the reservoir is calculated as follows:

$$h = \frac{Reservoir\ volume}{l \times w} \quad ...(17)$$

$$h = \frac{0.1285\ m^3}{0.8\ m \times 0.4\ m}$$

$$h = 0.401\ m$$

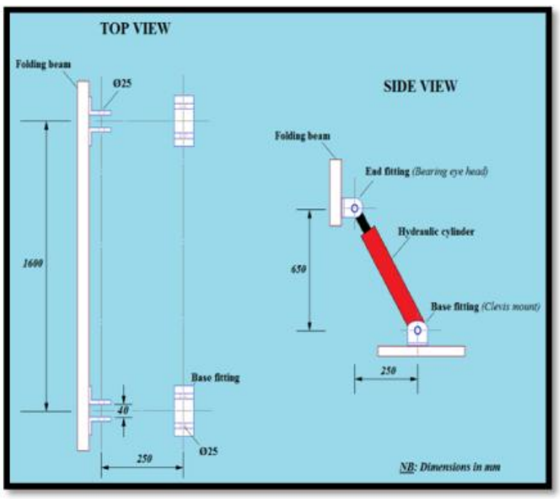
The height tank will have to be greater than 0.401 m because there must be a clearance volume above the oil, therefore the new height of the tank chosen will be 0.45 m.

Cylinder Connections

The attachment of the hydraulic cylinders to the folding beam has to be detachable to enable switching of the machine from hydraulic to manual mode. Figure 11 shows the details of the cylinder connections on the machine.

At the base fitting a clevis mounting is used that provides a single pivot mounting point on the cylinder. At the end fitting a bearing eye head is used to mount the cylinder onto the folding beam. The cylinder will be detached on the end fitting to disengage manual mode. A detachable connection using an M24 bolt with a shank length of 60mm will be used to connect the cylinder onto the folding beam.

Figure 11: Connection Details of the Hydraulic Cylinders



HYDRAULIC SYSTEM SIMULATION RESULTS

To simulate the operation of the hydraulic cylinder, a package called 'Simulation X 3.5' was used. The simulation give a mathematical imitation of the operation of each hydraulic cylinder over the period from fully closed to fully open which takes 4 seconds.

Figure 12 represents the simulation results of the piston stroke over the complete cycle time; a maximum stroke of 590 mm is attained.

Figure 12: Piston Stroke

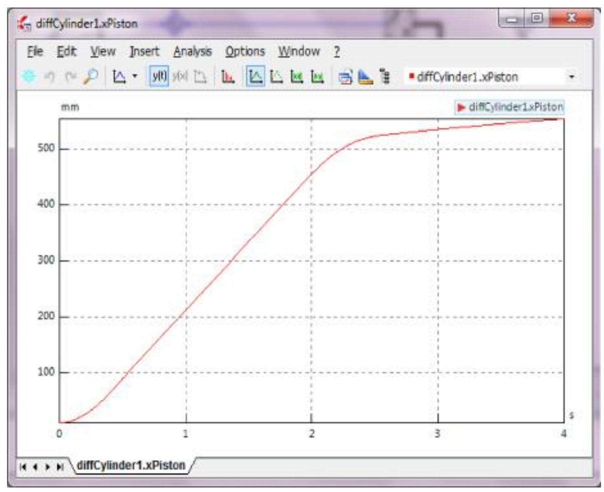


Figure 13 displays the flow rate from the pump to the directional control valve, from the calculations a flow rate of 51.4 l/min was obtained.

Figure 13: Pump Flow Rate to Directional Control Valve

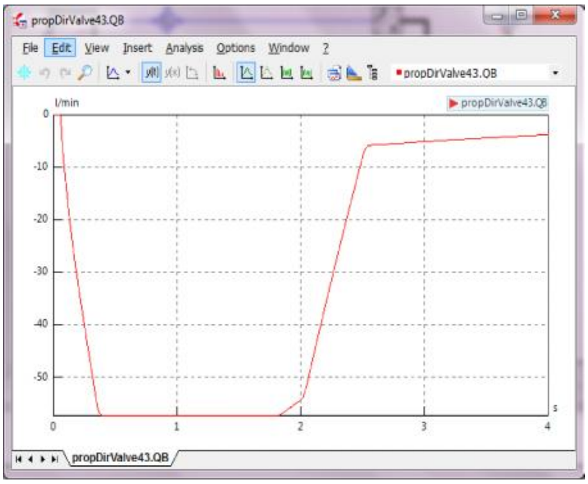


Figure 14 shows the pressure supplied by the pump of 150 bars during the 4 seconds.

Figure 14: Pressure Supply

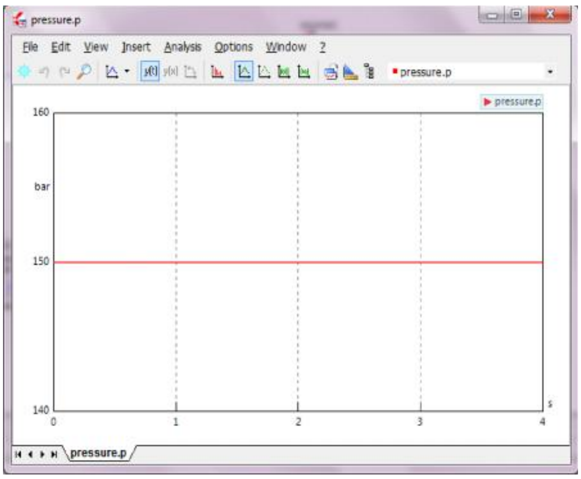
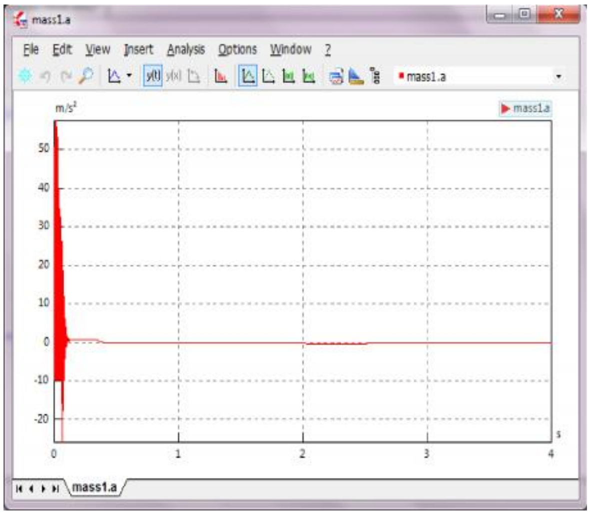


Figure 15 shows the acceleration on the load, in the first split seconds there are high accelerations due to shock as the load is starting is move. The acceleration then

Figure 15: Acceleration of the Load



decreases to zero to indicate constant velocity of the folding beam.

COMPLETE FOLDING MACHINE MODEL

A complete model of the designed dual operating mode sheet folding machine is shown in Appendix A. The model shows the location of the clamping mechanism (on one end of the machine) and a protractor gauge is located on the side frame for angle measurement during the bending operation.

CONCLUSION

The article has demonstrated an engineering procedure for design and validation by means of simulation, of a dual operating mode sheet metal folding machine. A non-commercial hydraulic simulation package (Simulation X 3.5) was used for simulation of the hydraulic system for the machine. A constant operating pressure of 150 bars will be maintained during machine operation whilst the folding bar shows a rapid acceleration, within a fraction of a second, towards the specimen sheet metal before it reaches a steady approach during the actual bending operation. This action prevents whipping of the specimen sheet metal and also gives good quality bending of sheet metal parts. A full scale version of the sheet metal folding machine can be manufactured to benefit the small and medium enterprises since the machine can be scheduled to operate manually and through hydraulic actuation.

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APPENDIX A

Dual Operating Mode Sheet Metal Folding Machine