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Development of a Motorized Pipe Bending Machine

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ABSTRACT

This is an improved design of the manually operated pipe bending machine commonly used by local metal construction workers. The design ensures a smooth operation during the bending process during which the shape or length of the pipe is bent to a required angle. The bending force is provided by means of a gear assembly, powered by a 2 Horse Power electric motor. It is desired that the pipe material permanently yields under the applied force without breaking since upon removal of the force the material is not expected to recover its original shape (plastic deformation). Thus yield stress of the range of pipe thicknesses was duly considered in the design. The bending mechanism is achieved by use of a worm and flywheel gear assembly, a protractor scale for recording the angle of bend, a detachable handle, bend dies of different sizes (between 18mm and 28mm), bearings and flexible couplings. The machine can be used in rural areas where there is no power supply. In this case, the bending fork is disengaged from the flywheel and the detachable handle is connected to manually turn the fork. The test result shows a smooth pipe bending operation up to a pipe thickness of 2mm. A square worm gave a relatively better result with the flywheel.

Keywords: *Pipe bending, motorized, angle of bend, bend dies, gear assembly, manually-operated.*

1. INTRODUCTION

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. 1.0)

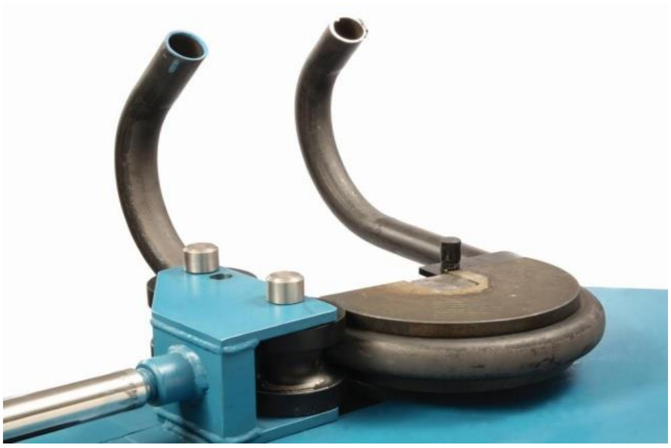

Fig. 1.0 Pipe Bending Process

In some pipe bending process and in order to prevent the pipe from collapsing, a mandrel is put inside the pipe itself.

Generally pipe bending machines can be:

- pneumatically powered;
- manually powered;
- hydraulically driven;
- electrically (servomotor) powered.

There is the Semi-automatic or automatic operation version which requires minimal need for the presence of an operator. The semi-automatic pipe bending machine (especially the hydraulic types) seems to be most preferred due to its low running cost, flexibility and less operator interface required. The Computer Numerical Control (CNC) bending machines have been developed to circumvent the problems associated with other pipe bending machines. The modern computer technology, linked with servo-mechanical control offers an excellent method for controlling 3 – bending axes (Khurmi and Gupta, 2006). In this arrangement, the operator hands the machine apart and actuates the start button. The machine bends the part. The operator removes the part and actuates the return sequence to repeat the operation. The CNC machine gives maximum accuracy and repeatability high degree of control, versatility and low labour input. It however requires initial high capital expenditure, operator computer familiarity (high skilled labour), water, air, and relatively high electrical power which make the operating cost high. Special application bending machines are normally custom-built designed for bending specific parts, especially parts that cannot be bent using the conventional bending equipment owing to size, shape or configuration (Miller, 2008).

The situation in most developing countries is however quite different. Craftsmen still use local means in their cold metal bending processes as locally made designs depend much on human effort. Further, the available local designs have no means of specifying the angle of bend and thus, pipe bending is a combination of both human effort (sometimes aided by lever principle) and guess work. First attempt was made by the author to develop a locally suitable pipe bending machine in 2007. The machine which was mechanically operated however showed some shortcomings. This necessitated the improved design which not only takes care of the identified shortcomings but also seeks to eliminate human effort. The motorized rapid pipe bending machine is developed to suit local conditions and minimize cost.

2. PRELIMINERY DESIGN

Figure 2.1 shows the orthographic view of the machine.

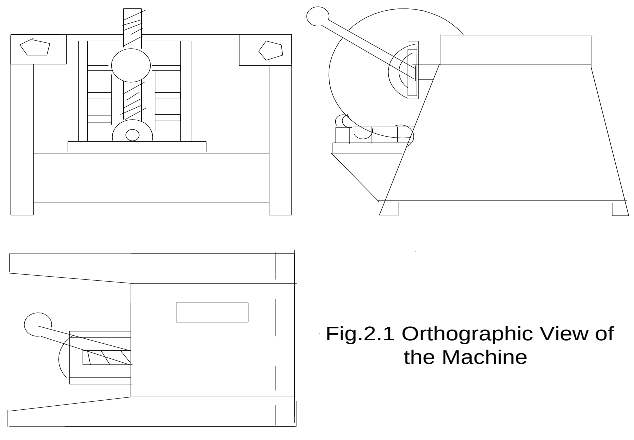

Fig.2.1 Orthographic View of the Machine

2.1 Description

The major machine components such as the electric motor, the warm gear, flywheel, and drive shaft are encased, leaving out a space that holds an extra length of the pipe in position. This guide is made adjustable in order to handle different sizes of bell.The bend fork is bolted to the flywheel. For manual bending, the bolts to the flywheel are loosened and an effort arm fixed to the bend fork is used to effect bending of the pipe. A protractor scale is attached beside the bell to indicate the angle of bend. After each operation, the bend fork is returned to position by the action of an automatic 2-way switch.

2.2 Principle of Operation

The machine is powered by an electric motor which supplies power to the spur gear speed reduction mechanism. The reduced output speed is transmitted to the warm and wheel gear supported on two bearings. The rotating flywheel gradually moves the pipe to the required direction and angle of bend, or at most a U-shape. With the help of the protractor scale, the angle of bend is recorded. In rural areas where there is no power supply, the connection of the bend fork with the flywheel is disengaged and a detachable handle is connected to the fork in order to carry out the operation manually.

3. DESIGN ANALYSIS

3.1 Effort Required to Bend the Pipe

The machine is considered as a lever with effort arm inclined at angle θ (Fig.3.1).

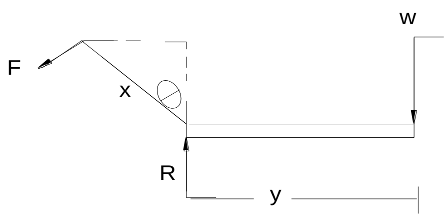

Fig.3.1 Lever

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.

Horizontal component, $F_h = F \sin\theta$ and Vertical component,

$F_v = F \cos\theta$

Taking moment at the support reaction, R;

$Fx \cos\theta = Wy$; $F = Wy / x \cos\theta$; where F = Effort required to bend the pipe

Let $\theta = 30^o$ and $x = 5y$ (i.e. depending on the length of the effort arm)

$F = 23.3 \times 10^3 . y / 5y \cos 30 = 5.5$ kN; where W = average bending force of pipe, 23.3×10^3 N

3.2 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

3.3 Speed Reduction (Spur Gear Design)

Minimum number of teeth on the pinion;

$T_p = 2A_w / G\sqrt{\ 1 + 1/G\ (1/G + 2) \sin^2\theta - 1}$ (Shigley and Mischke, 1989)

Where G = Gear ration / Velocity ratio; and θ = pressure angle, 20

A_w = Fraction by which the standard addendum is multiplied, 1m for θ =20

$T_p = 2 \times 1 / 2 \sqrt{1 + ½ (1/2 + 2) \sin^2 20 -1} = 14\ 2$

Thus, we choose T_p = 18 from standard table (Shigley and Mischke, 1989)

Number of teeth on the gear, $T_g = 2T_p = 2 \times 18 = 36$

But centre distance between the gears, $L = D_g/2 + D_p/2$

Where D_g = Diameter of gear, and D_p = Diameter of pinion

$D_g / D_p = 2$; $D_g = 2D^p$; $L = 2D_p / 2 + D_p / 2 = 3/2 (D_p) = 1.5 D_p$

$64 = 1.5D_p$; $D_p = 43$

$D_p = m\ T_p$; where m is the module

$m = D_p / T_p = 43/18 = 2.4$; Use standard value, m = 2.5

Pitch Circle Diameter of gear, $D_g = 2D_p = 2 \times 43 = 86$ mm

3.4 Face Width of the Pinion and the Gear

Pitch line velocity, $V = \pi D_p N_p / 60 = \pi \times 0.043 \times 1410 / 60$

= 3.17 m/s

For medium load shock condition and between 8~10 hours of service per day (Khurmi and Gupta, 2004);

Service Factor, C_s = 1.54 and 2.369 for non-enclosed gears.

Tangential Tooth Load, $W_T = C_s (P/V) = 2.369 \times 1500/3.20 = 1110.5$N

Velocity Factor, $C_v = 4.5 / 4.5 + V = 4.5 / 4.5 + 3.20 = 0.584$

Since the pinion and the gear are of same material, the pinion is weaker. For 20^o involute teeth;

Lewis Form Factor, $Y_p = 0.154 – (0.912 / T_p) = 0.154 – (0.912 / 18) = 0.1033$

Thus, design tangential tooth load; $W_T = \delta W_p \times C_v \times b \times \pi \times m \times Y_p$

Where δW_p is the safe stress of the pinion, 140 MPa and b is the face width of both pinion and gear.

$W_T = 140 \times 0.584 \times b \times \pi \times 2.5 \times 0.1033$; 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

3.5 Power Transmitted

$P = W_T \times V = 1110.50 \times 3.2 = 355$kW

Check for Static and Dynamic Loading

Flexible endurance limit for steel, δ_s = 252 (Khurmi and Gupta, 2004)

Static load or endurance strength, $W_s = \delta s \times b \times \pi \times m \times y$

$W_s = 252 \times 23.8 \times \pi \times 2.5 \times 0.1033 = 4865.9N$

Power that can be transmitted due to static loading is; $P_s = 4865.9 \times 2.9 = 14.3$ kW

Since P_s (14.3 kW) is greater than P (1.5 kW), the design is safe from the standpoint of static loading.

Also Dynamic Load, $W_D = W_T + [2/V (bc + W_T) / 2/V \sqrt{bc} + W_T$

But from table (Khurmi and Gupta, 2004), C = 228, and tooth error, e = 0.02

$W_D = 1110.5 + [2/3.2 (23.8 \times 228 + 1110.5) / (2/3.2 \sqrt{23.8 \times 228 + 1110.5}) = 3918.29N$

Power that can be transmitted from this dynamic load, $P_D = W_D \times V = 3918.29 \times 3.2 = 12.5$ kW

Since P_D (12.5kW) is greater than P (1.5kW), the design is safe from the standpoint of dynamic loading.

3.6 Design of Pinion Shaft

Load acting between the tooth surface; $W_N = W_T / \cos\theta = 1110.5 / \cos 20 = 1181.8N$

Weight of pinion, $W_p = 0.00118 \times T_p \times bm^2 = 0.00118 \times 18 \times 23.8 \times 2.5^2 = 3.16$ N

Resultant load acting on the pinion;

$W_R = \sqrt{W_N^2 + W_P^2 + 2W_NW_P\text{Cos}\,\theta} = \sqrt{1181.8^2 + 3.16^2 + 2(1181.8 \times 3.16 \times \text{Cos } 20)} = 1183.29$ N

Bending Moment due to this resultant load;

$M_B = W_R \times D_p/2 = 1183.29 \times 43/2 = 25440.74$ N-mm

Twisting Moment on pinion;

$M_T = W_T \times D_p/2 = 1110.5 \times 43/2 = 23875.8$N-mm

Equivalent Moment, $M_E = \sqrt{M_B^2 + M_T^2} = \sqrt{25440.74^2 + 23875.8^2} = 34889.6$ N-mm

But equivalent twisting moment is given by;

$T_E = (\pi / 16) \times 40 \times D_p^3$; $D_p^3 = 34889.6 \times 16 / \pi \times 40 = 4440.5$; $D_P = 16.5$ mm

This shows that with $D_p = 43$ mm, the design is quite safe.

Diameter of pinion hub = $1.8 D_p = 1.8 \times 43 = 77$ mm

Length of hub = $1.25 D_p = 1.25 \times 43 = 54$ mm

Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 10 mm).

3.7 Design of Gear Shaft

Normal load acting on the gear, $W_N = 1181.8N$

Weight of gear, $W_g = 0.00118\ T_g bm^2 = 0.00118 \times 36 \times 23.75 \times 2\ 5^2 = 6.31$ N

Resultant load acting on the gear; $W_R = \sqrt{W_N^2 + W_g^2 + 2W_NW_g \text{Cos}\,\theta}$

$W_R = \sqrt{1181.8^2 + 6.31^2 + 2(1181.8 \times 6.31 \times \text{Cos } 20} = 1189.73N$

Bending moment due to resultant load, $M_B = 1189.73 \times 86/2 = 51158.39$ N-mm

Twisting moment, $M_T = 1110.5 \times 86/2 = 47751.5$ N-mm

Equivalent moment, $M_E = \sqrt{M_B^2 + M_T^2} = \sqrt{51158.39^2 + 47751.5^2} = 69981.3$ N-mm

But equivalent twisting moment, $T_E = (\pi/16) \times 40 \times D_g^3 = (\pi/16) \times 40 \times D_g^3$

$D_g^3 = 69981.3 \times 16 / \pi \times 40 = 8906.7$; D_g (minimum value) = 21mm

This shows that with $D_g = 86$ mm, the design is quite safe.

Diameter of gear hub = $1.8 D_g = 1.8 \times 86 = 154.8$ mm

Length of gear hub = $1.25 D_g = 1.25 \times 86 = 107.5$ mm

Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 12 mm)

3.8 Design of Worm Gear

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

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

$10.16 \times 147.65 = T_s \times 73.83$; $T_s = 10.16 \times 147.65 / 73.83 = 20.37$ N-mm

Hence, transmitted power, $P = T_sV = T_s \times 2 \times \pi \times N/60 = 20.37 \times 2 \times \pi \times 705 / 60 = 1.5$ kW

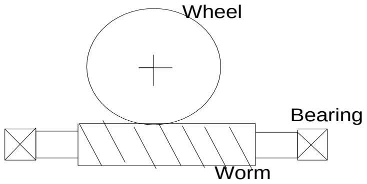

Fig.3.2 Worm Gear

The minimum value of X/LN will correspond to;

$Cot^3\ \Phi = V.R = 20$ Where; X = Lead = 200 mm, L_N = Normal load, and V.R. = Velocity ratio

$Cot\ \Phi = 2.71$; $\Phi = 20.22^o$

$X / L_N = ½ (1/\sin \Phi + V.R/\cos \Phi) = ½ [(1/\sin 20.22) + (20/\cos 20.22)] = 3.85$

$L_N = 200 / 3.85 = 51.92$ N

Axial load, $L_A = L_N / \cos \Phi = 51.92 / \cos 20.22 = 55.33$ N

For a V.R. of 20, the number of starts or threads on the worm,

$n = T_w = 2$ (Allens et al, 1980)

Thus, axial pitch of the thread on the worm;

$P_a = L_A/2 = 55.33 / 2 = 27.67$

Module, $m = P_a/\pi = 27.67/\pi = 8.8$ (Take standard module = 8).

Thus, axial pitch of the threads on the worm,

$P_A = \pi m = \pi \times 8 = 25.14$

Axial lead of the threads on the worm,

$L_a = P_a\ n = 25.14 \times 2 = 50.28$ mm

Normal lead of the threads on the worm, $L_n = L_a \cos \Phi = 50.28 \times \cos 20.22 = 47.2$ mm

But centre distance $= L_N/2\pi\ (1/\sin \Phi + 1/\cos \Phi) = 47.2/2\pi\ (24.21) = 181.73$ mm

Let D_W = Pitch circle diameter of the worm; then

$Tan\ \Phi = L / \pi D_W$

$D_W = L_a / \pi \tan \Phi = 50.28 / \pi \tan 20.22 = 43.44$ mm

Since V.R. is 20 and the worm has double threads; Number of teeth on the worm gear $T_g = 20 \times 2 = 40$

Face length of the worm (i.e. length of the threaded portion);

$L_W = P_A\ (4.5 + 0.02T_W) = 25.14\ (4.5 + 0.02 \times 2) = 114.09$ mm

(this is normally increased by 25 ~ 30 for the feed mark) (Khurmi and Gupta, 2004);

Thus $L_W = 144$ mm

Depth of tooth, $h = 0.686P_A = 0.686 \times 25.14 = 17.24$ mm

Addendum, $a = 0.313P_A = 0.313 \times 25.14 = 7.86$ mm

Outside diameter of worm,

$D_{OW} = D_W + 2a = 43.44 + 2 \times 7.86 = 59.42$ mm

Circle pitch diameter of worm gear,

$D_g = m\ T_g = 8 \times 40 = 320$ mm

Outside diameter of worm gear,

$D_{og} = D_g + 1.0135P_A = 320 + 1.0135 \times 25.14 = 345.5$ mm

Face width,

$b = 2.38\ P_A + 6.5$ mm $= 2.38 \times 25.14 + 6.5 = 66.31$ mm

3.8.1 Design of Worm Shaft

Torque acting on the worm gear shaft,

$T_g = P \times 60 / 2\pi N_g$

Considering 30% overload; $T_g = 1.3 \times 1500 \times 60 / 2 \times \pi \times 35.25 = 528.26$ N-m

Torque acting on the worm shaft,

$T_W = Tg / V.R. \times \eta$ where η = efficiency of worm gear

But $\eta = Tan\ \Phi / Tan\ (\Phi + \alpha)$ where α = angle of friction

Rubbing velocity, $Vr = \pi D_W N_W / \cos \Phi = \pi \times 0.0434 \times 705 / \cos 20.22 = 102.53$m/mm

Coefficient of friction, $\mu = 0.275 / Vr^{0.25} = 0.275 / (102.53)^{0.25}$ $= 0.086$

Thus, angle of friction, $\alpha = \tan^{-1} \mu = \tan^{-1} 0.086 = 4.92$

Efficiency of worm gear, $\eta = \tan 20.22 / \tan (20.22 + 4.92) = 0.78\%$

Thus, torque acting on the worm shaft,

$T_W = 528.26 \times 10^3 / 20 \times 0.78 = 33.86 \times 10^3$ N-m

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

$W_T = 2 T_W / D_W = 2 \times 33.86 \times 10^3 / 43.44 = 1559.06$ N

Axial load acting on the worm,

W_A = Tangential load on the worm gear

$W_A = 2T_g / D_g = 2 \times 528.26 / 320 = 3301.63$ N

Radial or separating force on the worm gear,

$W_R = W_A \tan \Phi = 3301.65 \times \tan 20.22 = 1216.07$ N

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

Then, bending moment due to W_R in the vertical plane = $W_R X/4 = 1216.07 \times 40 / 4 = 12160.7$N-m

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

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

Bending Moment in the horizontal plane, $M_H = W_T \times 40/4 = 1559.06 \times 40/4 = 15590.6$ N-m

Resultant Bending Moment acting on the worm shaft,

$\mathbf{M_R} = \sqrt{M_H^2 + M_V^2}$

$\mathbf{M_R} = \sqrt{276291.1^2 + 15590.6^2} = 276730.63$ N-m

Equivalent twisting moment on the worm shaft,

$\mathbf{M_{ET}} = \sqrt{T_W^2 + M_R^2}$

$\mathbf{M_{ET}} = \sqrt{(33.86 \times 10^3)^2 + (276730.63)^2} = 278794.44$ N-m

But $M_{ET} = \pi/16 \times \tau \times D_W^2$

where D_W = diameter of worm shaft; τ = allowable shear stress, 40 N/mm^2

$D_W = 32.86$ mm (Use D_W = 36mm).

4. PERFORMANCE TEST

Three different sizes of bell (lower bend dies) were used for the test. The following were considered;

1. quality of bend
2. angle of bend

Series of tests were carried out using galvanized and steel pipes for the different sizes of bells. The test results are presented in figures 4.1 -to - 4.6.

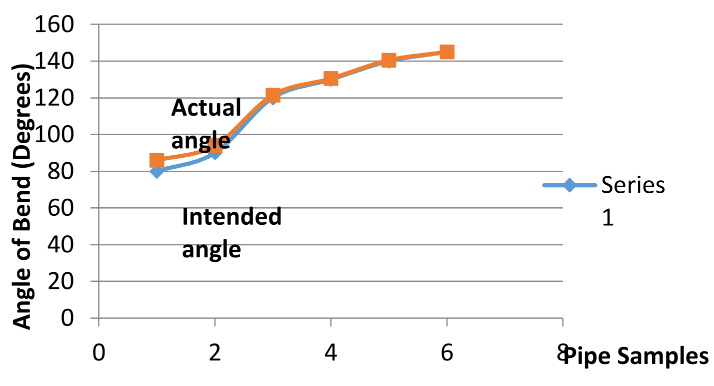

Fig.4.1 Result for 1.5mm Pipe Thickness Manual Bent Using 18mm Bell

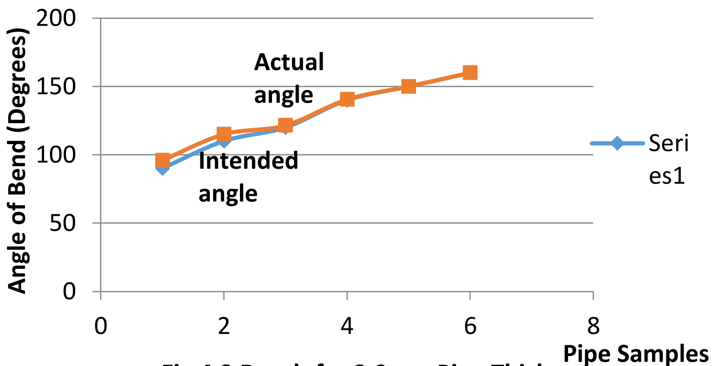

Fig.4.2 Result for 2.0mm Pipe Thickness Manually Bent Using 22mm Bell

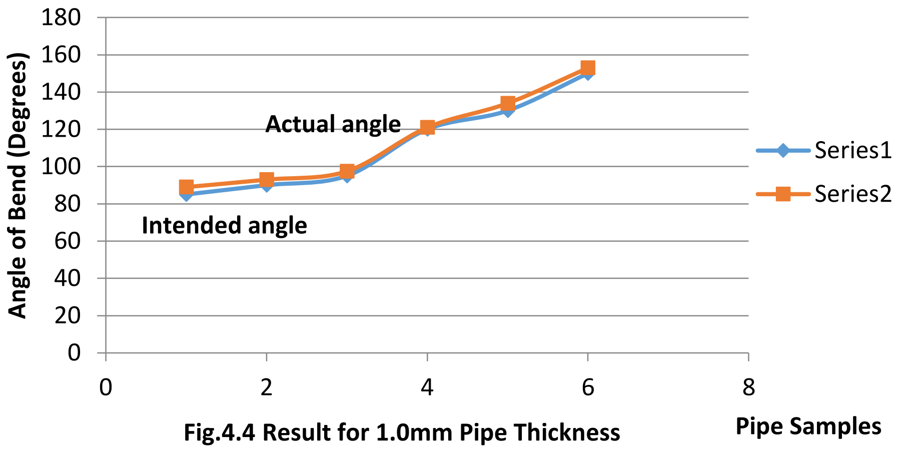

Fig.4.4 Result for 1.0mm Pipe Thickness Electrically Bent Using 18mm Bell

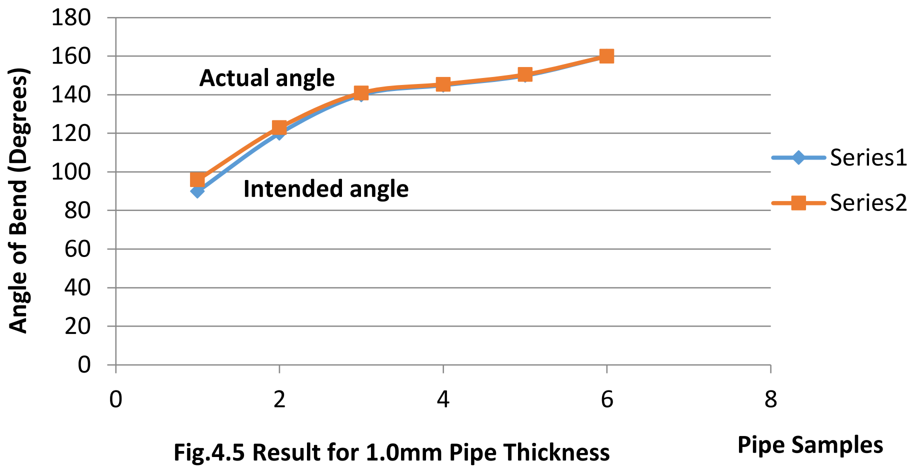

Fig.4.5 Result for 1.0mm Pipe Thickness Electrically Bent Using 22mm Bell

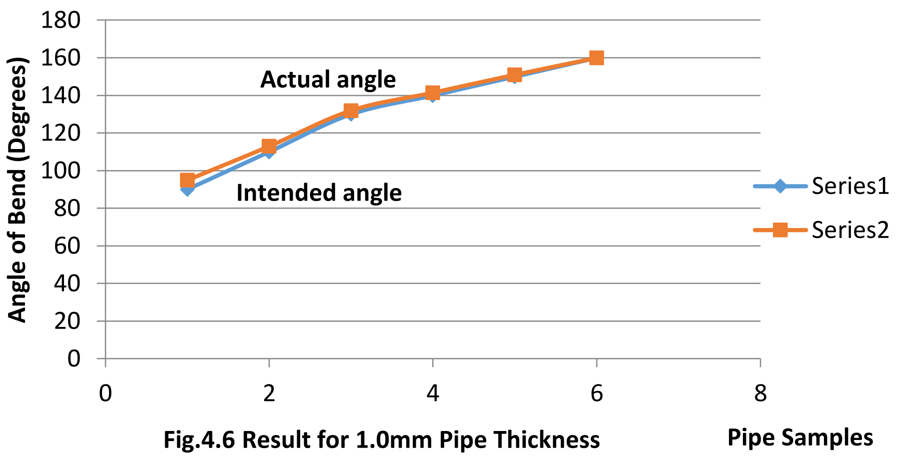

Fig.4.6 Result for 1.0mm Pipe Thickness Electrically Bent Using 28mm Bell

5. DISCUSSION

Observations were made as follows:

- Up to 2 mm pipe thickness could be bent manually depending on the operator's physical strength. Up to 1 mm pipe thickness could be electrically bent most conveniently, but operates sluggishly above 1 mm.

- The slight deviation from intended angle of bend was due to spring back action of the pipe, which obviously reduces with decrease in angle of bend.

- Wrinkles and bulging of the pipe during bending were noticed in pipes of lesser thickness.

- With a semi-circular bell, angles between 80^o to 180^o were obtainable, below which the bent pipe was observed to follow the bell's contour to give a U or C shape. To obtain a lower bend angle, a different bell specially made for 45^o is used.

6. CONCLUSION

The machine is powered by a 2HP motor and pipe bending can be achieved in both ways (either in upward or downward direction). Electrical bending is achievable with the aid of a worm and wheel gear assembly. The provision for manual bending makes it possible to use the machine in rural areas were there may be no power supply. For pipes of lesser thickness, a mandrel should be introduced in order to prevent collapse.

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