RESEARCH PAPER ON DESIGN, FABRICATION AND FUNCTIONAL ANALYSIS OF HEAVY-DUTY PIPE BENDING MACHINE USING HYDRAULIC POWER PACK

Praveen Kumar KJ[1], Rajesh D[2], Yashwanth Babu N[3]

[1,2,3] *Department of Mechanical Engineering, Sri Sairam Institute of Technology, Chennai, 600044, India.*

***

Abstract: Light duty machines readily available in the market are capable of bending up to 4" 100NB sch. 40 pipes. In Nuclear industries, more than 100 NB and 40 sch. pipes are required to be bent to longer radius. Hence, to meet this need for this paper a reliable, mobile and low-cost prototype pipe bending machine is designed, the detailed part drawings were made and the industrial machine components were fabricated, assembled, functionally tested by introducing design innovation and improvisation to the existing light duty pipe bending machine driven by a hydraulic power pack to bend pipe that needs larger bending radius and have higher wall thickness. The investigation was performed on 4" NB sch. 160 pipe of ASTM A 333 grade 6 (of Bending radius = 500mm) pipe. The paper also presents the research work on partitioning of finite mesh elements with static nodal analysis. It is expected that this study might be helpful for further related works.

Index Terms - Bending radius, Wall thickness, Ovality, angular bent, mesh elements.

I. INTRODUCTION

Fluids are transported through the pipe in the industries. To change the direction of fluid flow, pipe bends are used. Short pipe bends are normally manufactured by casting or forging. Casted and forged pipe bends are joined by means of welding, leading to heterogeneous microstructure and resulting in invariant chemical composition is inevitable. This heterogeneous piping system leads to pressure drop and erosion. So, this heterogeneity in composition and microstructure reduce the service life of the piping system. So alternatively, long radius pipe bends are made by cold forming using the same pipe material, in case of conventional pipe bending machine. So, this long bends reduces the number of weld joints and also reduces friction flow, pressure drop and erosion into the pipe bends. The standard and market available pipe bending machine has a capacity of up to 4" 100NB Schedule 40 (Outer Diameter 114.30 mm x 6.02 mm wall thickness) on carbon steel material (UTS 420Mpa). However, higher size wall thickness pipe bends are required for nuclear industries to meet the design life of more than 60 years.

Due to lack of machinery readily available in the market and need of heavy-duty pipe bending machine for higher thickness pipe, we are designing and fabricating a heavy-duty pipe bending machine to meet the industrial requirement.

II. DESIGN OF HEAVY-DUTY PIPE BENDING MACHINE

The available light duty pipe bending machine in market was studied in detail and similar conceptual heavy duty pipe bending machine is designed. A 50 Ton capacity hydraulic power pack is used for bending of heavy-duty pipes. The following are major components of the machine.

A. TOP PLATE

Material of construction is IS 2026 Grade

Chemical composition

ELEMENT	C%	Mn%	S%	P%	Si%	C.E.%
CONTENT	0.22	1.5	0.045	0.045	0.04	0.41

Mechanical properties

PROPERTIES	UTS(MPa)	Y.S(MPa)
METRIC	410	250

Allowable working stress = $\frac{\text{Ultimatetensilestrength}}{\text{factorofsafty}}$

$= \frac{410}{8}$ = 51.25 N/mm2

Allowable working stress $= \frac{Force}{Area} = \frac{Force/2}{L*t} = \frac{50000*9.81/2}{(355-70)*t}$ The minimum allowable thickness for the top plate is 16.79 mm.

B. HINGE PLATE

Material of construction is 17-4 PH Steel hardened at

900°C.

Chemical composition

ELEMENT	C%	Mn%	S%	P%	Si%	Cr%
CONTENT	0.07	1.0	0.030	0.040	1	15-17.5

Mechanical properties

PROPERTIES	UTS(MPa)	Y.S(MPa)
METRIC	1448	1379

Considering 4 shear zones

Number of Shear = 4

Shear stress = UTS/2 = 1448/2 = 724 N/mm2

Allowable shear stress = $\frac{\text{Shear stress}}{\text{FOS}} = \frac{724}{8}$

= 90.5 N/mm2

Shear force = $\frac{Force/2}{Number of shear} = \frac{490500/2}{4}$

= 61312.5N

Allowable shear stress = $\frac{61312.5}{\frac{\pi}{4} * d * d}$

90.5 = $\frac{61312.5}{\frac{\pi}{4} * d * d}$

d^2 = 862.60 d = $\sqrt{862.60}$d = 29.37 mm

The minimum allowable hinge pin diameter is 29.37 mm.

C. PUSHING ROD

Material of Construction is Cast iron.

Allowable stress = Ultimate tensile strength/Factor of Safety

Allowable stress = 720/8 = 90 N/mm2

Allowable stress = Force/Area = 490500/500*t

t = 490500/500*90 t = 11 mm

The minimum allowable thickness for pushing die is 11 mm.

D. GUIDE PIN

Material of construction is 17-4 PH Steel hardened at 900°C.

1. **By Shear method:** Number of Shear = 2

   Shear stress = UTS/2 = 1448/2 = 724 N/mm2

   Allowable shear stress = $\frac{\text{Shear stress}}{\text{FOS}}$ = 724/8 = 90.5 N/mm2

   Shear force = F/4/No of shear = 122625/2 = 61312.5 N

   Allowable shear stress = 61312.5/((π/4) *d2)

   90.5 = 61312.5/((π/4) * d2)

   d2=61312.5/((π/4) *90.5) d2 = 862.6 d = $\sqrt{862.60}$

   d = 29.37mm

The minimum allowable guide pin diameter by shear method is 29.37mm.

2. By Bending Method

$\sigma = \frac{\text{Bending moment (M)}}{\text{Section modulus (Z)}}$ =M/Z= 1448/8 = 181 N/mm2

Allowable stress = $\frac{(W \times L/4)}{(\frac{\pi}{32}) \times d^3}$

$181 = \frac{\frac{\frac{F}{2} \times \frac{L}{2}}{4}}{(\frac{\pi}{32}) \times d^3}$ $d^3 = \frac{\frac{\frac{490500}{2} \times \frac{390}{2}}{4}}{(\frac{\pi}{32}) \times 181}$

d^3 = 672829.72 d = 87.62mm

The minimum allowable diameter by bending method is 87.62mm.

E. GROVE IN PLATE

Material of construction is IS 2026 Grade B.

Shear stress = ultimate tensile strength /2

= 410/2 = 205 N/mm2

Allowable shear stress = 205/8 = 25.625 N/mm2

Shear force = (F/2) / (1) = 245250/1 = 245250 N

Allowable shear stress = shear force /shear area

25.625 = 245250 / (L*W)

W = 26.95mm

The minimum allowable groove width is 26.95 mm.

F. DESIGN OF BOLT

Considering High tensile bolts p 10.9

For p 10.9 UTS = 1000N/mm2

Yield strength = 900 N/mm2

Shear stress = $\frac{\text{ultimate tensile strength}}{2}$ $\frac{1000}{2}$= 500 N/mm2

Allowable shear stress = $\frac{500}{8}$ = 62.5 N/mm2

Shear force = $\frac{F/2}{1}$ = 245250N

Shear stress = $\frac{\text{shear force}}{\text{shear area}}$ 62.5 = $\frac{245250}{\frac{\pi}{4} * d * d * n}$

Assuming n=5, d2 = $\frac{4996.18}{5}$ d2 = 999.23 d = 31.6 mm

Assuming n= 6, d2 = $\frac{4996.19}{6}$ d2 = 832.69

d = 28.85 mm. So, M30 bolt of P10.9 is selected.

G. BACK PLATE

Material of construction is IS 2026 Grade B.

Allowable working stress σ = $\frac{UTS}{FOS}$ = $\frac{480}{8}$ = 51.25 N/mm2

Deflection δ = $\frac{WL^3}{48EI}$ = $\frac{WL^3 \times 12}{48 \times 200 \times 10^3 \times b^3 d}$

1= $\frac{490500 \times 352^3 \times 12}{48 \times 200 \times 10^3 \times b^3 \times 352}$ b^3 = $\frac{7.292 \times 10^{11}}{9600000}$ b= 42.35 mm. The minimum allowable breadth of back plate is 42.35 mm.

III. MODELLING AND STRESS ANAYLSIS

A. 3D-Model of various components before assembly

1. Top plates

Fig-1 Top plates

Flat plates of uniform cross section provided with a series of holes of different diameter to insert pins through supporting dies.

2. Supporting Dies

Fig-2 Supporting Dies

Pair of dies of cylindrical cross section with elliptically inscribed grooves on their periphery to rest and move the pipe of various diameters.

3. Guide pins

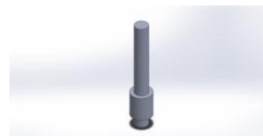

Fig-3 Guide pins

Pair of die pins resembling a long rod with square turning at its bottom end, is inserted through the supporting dies to provide physical support to the dies and hence the pipe.

4. Top Hinged plate

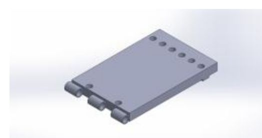

Fig-4 Top Hinged plate

Square plate of distinctive cross section welded with hinge joint at one of its end and other end has an envelope to rest smoothly into the slot engraved on top of the back plate. It is also provided with a series of bolt to keep them intact.

5. Back plate

Fig-5 Back plate

Square plate marks the back end of the machine whose top and bottom sides are inscribed with rectangular slots to place the top and bottom plate diplomatically.

B. Innovation in the design in contrast to the conventional pipe bending machine

1. Multiple grooves on the Supporting dies to accompany pipe of different OD

Fig-6 (a)Die with single groove (b)Die with multiple grooves of different sizes

In conventional design the supporting die pair is required for each size of the pipe like 25NB, 50NB, 75NB and 100NB (1” NB, 2” NB, 3” NB and 4” NB). To reduce the number of supporting die pairs, innovative design is adopted. The support die is divided into 4 equal segments each 90 degrees apart on the outer periphery. The pipe resting grooves for each size of the pipe is designed on each segments of the supporting die as seen in the figure.

2. Introduction of slots to reduce the number of bolts used

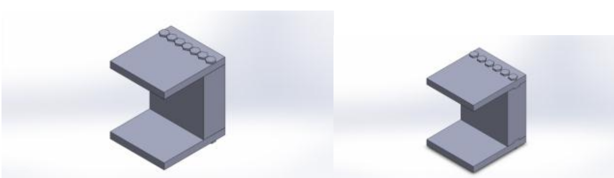

Fig-7 Number of bolts (a)Before slot (b)After slot

By introducing slots on top and bottom of the back plate, the number of bolts used to accompany the top and bottom plates is significantly reduced because of self-locating characteristics of the plates which causes sharing of the shear load partially.

No. of Bolt before introducing slot=7

No. of Bolts after introducing slot=6

3. Introduction of collars to reduce thickness of Die pins

Fig-8 (a) Supporting Die (b) Collar for Die

From the above performed design calculation we have obtained a very large thickness value for die pin which obviously would raise the problem in practicality, as we are demanded to drill a larger hole on the flat plates which will reduce the mechanical strength of it. To avoid the foresaid problem, we have opted to use an innovative approach of using collar which provides merely equal amount of stability without being getting into the action of drilling larger holes.

4. Introduction of Rib on top of front top plate before Hinge to enhance the strength of the joint

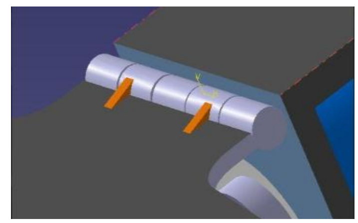

Fig-9 Ribs

The overall mechanical strength of the top plate is enhanced by the addition of a Rib before the hinge without adding much complexity to the machine.

C. Simulation results of stress analysis without and with slot

1. Back plate without and with slots for static

Displacement

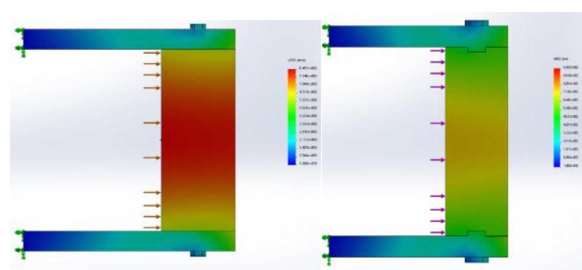

Fig-10 Static displacement analysis

In case of plate without slot, the load concentration is intense at the center of the plate in contrast to plate with slot because by introducing slots on top and bottom of the back plate, the shear load is partially shared in the slot and hence the number of bolts required is significantly reduced.

2. **Back plate without and with slots for static strain analysis**

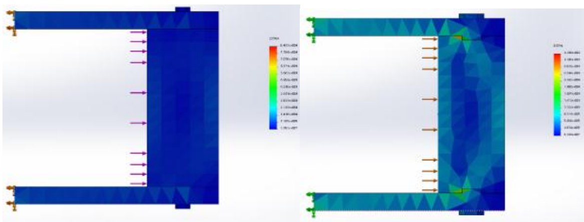

Fig-11 Static strain analysis

As the stimulated result clearly shows that the slot has acquired considerable load from the back plate and hence reducing the load concentration at the center of the plate.

IV. ASSEMBLY OF THE MACHINE AND FUNCTIONAL TESTING

1. Top plate

Fig-12 Picture of Fabricated plate

2. Push Die

Fig-13 Picture of Fabricated push Die

3. Supporting die with and without grooves

Fig-14 Picture of Grooves

4. Collar, Supporting die and Guide pin

Fig-15 Picture of Collar and Guide pin

5. Top Plate with Rib

Fig-16 Picture of Rib

6. Spacer Push Rod

Fig-17 Picture of push rod

7. Back Plate

Fig-18 Picture of Back plate

8. Hydraulic Power Pack

Fig-19 Picture of hydraulic power pack

9. Assembled Machine

Fig-20 Picture of HDPBM

10. Assembled Machine with top plate open

Fig-21 HDPBM ready to be loaded

V. CONCLUSIONS

This research work helped in

1. Designing a Heavy-Duty Pipe Bending machine and various stresses were analyzed and hence the conservative design was arrived. The detailed part drawings were made; the individual machine components were fabricated, assembled and functionally tested.

2. 4" NB sch. 160 pipe of ASTM A 333 grade 6 of (Bending radius = 500mm) was bent successfully using the fabricated heavy duty pipe bending machine. The bent pipe was also subjected to dimensional inspection including measuring of angular bent, bend radius, wall thickness and ovality. It is found that the dimensions are within the acceptable limits.

A. Challenges Encountered

Initially the entire process to arrive at an optimal design to attain fine partitioning of mesh elements was difficult. The simulation analysis did not produce good results and we had to run through different design options like reducing the number of bolts and opting for a collar rod with guide pins and the best part to introduce push rod to minimize the lateral strain to the machine.

B. Future Scope

This research paper proposes the scope of how some design innovation or improvisation to the existing design can make a big difference to the life, accuracy and thereby performance output of a machine. In future, additional design modification can help utilize the full force from the hydraulic power pack with minimal loss of pressure across the machine components.

VI. ACKNOWLEDGEMENT

We would like to express our gratitude and appreciation to the Scientific officer M. Krishnamoorthy as well as Central Workshop Division, IGCAR Kalpakkam who gave us the golden opportunity to do this wonderful research work to design and develop this machine and successfully put it into function.

REFERENCES

1. Hydraulic Pipe Bending Machine, Filed Feb. 17. 1950, Flanagan John E, Gustave Lidseen (1930) V Patented June 28, 1932 UNITED STATES, PATENT OFFICE HYDRAULIC MACHINE Application filed February 17, 1930. Serial No. 428,879.

2. N. N. Jadeja, M.C. Baraiya, A.B. Jasoliya, R.U. Jagad, V.R. Makwana, "Research Paper of Manually Operated Pipe Bending Machine", PARIPEX - INDIAN JOURNAL OF RESEARCH, 2014.

3. P.N. Awachat, Ghulam Dastgeer, Aman Gohite, "Design and Analysis of Re-bar Bending machine", IJIRAE, Issue 4, Volume 2 (April 2015).

4. Kulkarni Priyanka V., Mukhekar Vishnu A., Kute Abhishek V., Asst Prof Joshi Aniket V, "Design and Analysis of Pipe Bending Machine", IJCRT | Volume 6, Issue 2 April 2018.

5. "Design Data Book", Mr. P.s. Gill, 3rd Edition

6. Mr. V.B. Bhandari, "Design of machine element", Mr. Mc-Graw Hill Publication, 4rd Edition.