Design and Analysis of Adjustable Inside Diameter Mandrel for Induction Pipe Bender

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ABSTRACT: Pipes are bended to various radius in pipe bending machine (COJAFEX PB 600R). When the pipe length is insufficient mandrel is used. For various diameter pipes various mandrels are used which increases the cost and time of operation in order to avoid this difficulty a mechanism is designed by us in SOLIDWORKS to adjust the diameter of the inside mandrel to suit various diameters of the pipe. The existing mandrel is compared with the designed mandrel and analyzed in ANSYS WORKBENCH.

Keywords: Mandrel, Induction pipe bending, Bend roll force

I. INTRODUCTION:

Pipe bending is a process wherein the pipe to be bend is heated to a specified temperature by high frequency induction heating coil and bending is done in hot condition. The pipe is clamped at the forward end by the radius arm clamp. The tail end of the pipe is locked in the pushing carriage clamp. The pushing carriage moves on the main track as it is fitted with a chain, which is driven by a DC Motor at a fixed constant speed through a gearbox.

When current is supplied through high frequency current transformer to the coil, heating of pipe starts. The heating temperature is strictly controlled depending upon the material of the pipe and its wall thickness. When the specified temperature is attained, the DC Motor is started at a preset speed which drives the pushing carriage in the forward direction. As a result the pipe begins to advance being bent simultaneously.

Immediately after the bending process, quenching will follow. The Radius Arm is designed so that the pipe will be bent by the same distance as the one between the center of a pair of rollers and the axis point of the arm. The arm rotates in natural mode around the pivotal point. The heated portion begins to transform its elasticity due to the difference of strength between the heated and unheated portion of the pipe. Induction bends are stronger than elbows with uniform wall thickness. Induction bending does not need bend dies or mandrels.

A simple clamping/inductor set cover a wide range of radii and wall thicknesses. Induction bending is a clean process so no lubricant is necessary. Cojafex offers a range of induction bending machines covering the requirements of all major industries. In addition to the traditional single bend machine Cojafex offers a range of spool bending machines that allow you to make spools, i.e. multiple bends in different planes in one pipe. The smaller models can be executed with an integrated automatic pipe loading system.

II. MANDREL:

Normally mandrels are used to hold the pipes and also when the pipe length is insufficient. Depending upon the internal diameter of the pipes the mandrels are taken and used for pipe bending.

In that case, amount of using mandrels are increased for each pipes which are meant for pipe bending to avoid that a mechanism is designed and kept inside the mandrel so that it can expand and contact according to the diameters of the pipe. A range of diameter is selected and within that range this mandrel is designed.

III. ADJUSTABLE MANDREL:

The construction of the mandrel design is made using the available materials in the company. Those materials come under American Society of Testing and Materials. The construction of the mandrel body requires the following components

- Mandrel body
- Front cover plate
As the screw rod is rotated in clockwise and anticlockwise direction, It moves the cut section of the inside mandrel inside and outside to certain range of diameters so that the pipe can be placed or seated over the mandrel. When the screw rod rotates clockwise the connecting rods push the slot upwards and when the screw rod rotates anticlockwise the connecting rod pulls the slot inwards by which the ranges of diameters are controlled. For supports various plates and bearings are kept at front, rear and places required.

### Table I

<table>
<thead>
<tr>
<th>S.NO</th>
<th>PARTS</th>
<th>MATERIALS</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Mandrel body</td>
<td>A335 P12</td>
</tr>
<tr>
<td>2</td>
<td>Front cover plate</td>
<td>A387 Gr2 Cl2</td>
</tr>
<tr>
<td>3</td>
<td>Plain bearing ID 45</td>
<td>IS 305 Gr AB1</td>
</tr>
<tr>
<td>4</td>
<td>Screw rod</td>
<td>A182 F2 Cl3</td>
</tr>
<tr>
<td>5</td>
<td>Rear cover plate</td>
<td>A387 Gr2 Cl2</td>
</tr>
<tr>
<td>6</td>
<td>Plain bearing ID 50</td>
<td>IS 305 Gr AB1</td>
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<td>7</td>
<td>Lower eye end</td>
<td>A182 F2 Cl3</td>
</tr>
<tr>
<td>8</td>
<td>Right connecting plate</td>
<td>A387 Gr2 Cl2</td>
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<tr>
<td>9</td>
<td>Left connecting plate</td>
<td>A387 Gr2 Cl2</td>
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<tr>
<td>10</td>
<td>Upper eye end</td>
<td>A182 F2 Cl3</td>
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<tr>
<td>11</td>
<td>Connecting fork</td>
<td>A182 F2 Cl3</td>
</tr>
<tr>
<td>12</td>
<td>Connecting pin</td>
<td>A193 B7</td>
</tr>
<tr>
<td>13</td>
<td>Mandrel expansion Segment</td>
<td>A106 Gr C</td>
</tr>
</tbody>
</table>

### A. Design of square threaded screw:

\[ F = 4.96 \times 10^5 \text{N} \]

Total tensile pull, \( W_1 = 2 \times F \)

\[ W_1 = 2 \times 4.96 \times 10^5 \]

\[ W_1 = 9.94 \times 10^5 \text{N} \]

\[ W_1 = \frac{\pi}{4} d_c^2 \sigma_t \]

\[ 9.94 \times 10^5 = \frac{\pi}{4} d_c^2 \times 760 \]

Core diameter, \( d_c = 40.81 \text{ mm} \)

For design consideration we took, \( d_o = 51 \text{ mm} \)

\[ d_o = d_c + p \]

Here Pitch, \( p = 9 \text{ mm} \)

\[ d_o = 51 + 9 \]

Outer diameter, \( d_o = 60 \text{ mm} \)

\[ d = d_o - \frac{p}{2} \]

\[ d = 60 - \frac{9}{2} \]

Mean diameter, \( d = 55.5 \text{ mm} \)

\[ \tan \alpha = \frac{p}{\pi \times d} \]

\[ \tan \alpha = \frac{9}{\pi \times 55.5} \]

\[ \tan \alpha = 0.0516 \]
Helix angle, $\alpha = 2.95^o$
Effort, $P = W_1 \times \tan (\alpha + \Phi)$

$P = W_1 \times \tan (n \times d \times \tan \alpha)$

$P = 9.94 \times 10^5 \times 0.0516 + 0.1$ \[\mu = \tan \Phi \]

Coefficient of friction, $\mu = 0.1$

$P = 1.51 \times 10^6 \times N$

$T = \frac{P \times d}{2}$

Torque, $T = \frac{1.51 \times 10^6 \times 55.5}{2}$

$T = 4.2 \times 10^6 \ \text{Nm}$

Shear stress, $\tau = \frac{16 \times T}{\pi \times d^3}$

$\tau = \frac{16 \times 4.2 \times 10^6}{\pi \times 51.3}$

$\tau = 161.25 \ \text{N/mm}^2$

B. Design of nut:

In order to have good stability we shall provide $n = 4$ threads in the nut

Thickness of the nut, $t = n \times p$

$t = 4 \times 9$

$t = 36 \text{mm}$

$t = 40 \text{mm}$

C. Design of pins in the nut:

Since the pins are in double shear

Pin diameter, $d = 2 \times \pi \times d_1 = \tau$

For A193 B7 shear strength

$\tau = 689.47 \ \text{MPa}$

$4.94 \times 10^6 = \pi \times d_1^2 \times 689.47$

Pin diameter, $d_1 = 21.35 \text{ mm} = 30 \text{ mm}$

D. Design of connecting fork:

Due to the load, the links may buckle in two planes at right angles to each other. We know that on the link

$\frac{F}{2} \times \frac{4.96 \times 10^5}{2} = 2.48 \times 10^5 \ \text{N}$

Assuming a factor of safety = 5, the links must be designed for a buckling load of

$= (2.48 \times 10^5) \times 5$

$= 1.24 \times 10^6 \ \text{N}$

Cross sectional area, $A = \frac{\pi}{4} \times d_2^2$

$= 0.785 \times d_2^2 \ \text{mm}^2$

Moment of inertia, $I = \frac{\pi}{64} \times d_2^4$

$= 0.049 \times d_2^4 \ \text{mm}^4$

Radius of gyration, $k = \sqrt{\frac{1}{A}}$

$= \sqrt{\frac{0.049 \times d_2^4}{0.785 \times d_2^2}}$

$= 0.25 \times d_2$

Direct tensile stress, $\sigma_t = \frac{9.94 \times 10^5}{2 \times \pi \times d_2}$

$= 486.58 \ \text{N/mm}^2$

$\sigma_t \max = 3055.35 + 3131.64 = 53.16 \ \text{N/mm}^2$

Since the maximum tensile stress is within the limits the design of square threaded screw is safe.

Length of the link, $L = 134.56 \ \text{mm}$

Rankine’s constant, $a = \frac{1}{7500}$

According to Rankine’s formula, buckling load ($W_c$),

$W_c = \frac{\pi \times A \times d_2^2}{1 + a \left( \frac{d_2}{L} \right)^2}$

$= 1.24 \times 10^6 \times \frac{517.0785 \times d_2^2}{1 + a \left( \frac{d_2}{L} \right)^2}$

$= 1.24 \times 10^6 \times \frac{517.0785 \times d_2^2}{1 + \frac{d_2^2}{7500}}$

$= 1.24 \times 10^6 \times \frac{517.0785 \times d_2^2}{1 + \frac{d_2^2}{7500}}$

$= 3055.35 \times d_2^2 - 117997.76 < 0$

$= 3055.35 \times d_2^2 - 117997.76 < 0$

Taking ve sign

$d_2^2 = 3055.35 \times (4+117997.76)$

$d_2 = 3093.4$

$d_2 = 55.6 \ \text{mm}$

Rounded Value diameter, $d_2 = 60 \ \text{mm}$

Now let us consider the buckling of the link in a plane perpendicular to the vertical plane.

Moment of inertia, $I = \frac{d_2^4}{64}$

$= 0.049 \times d_2^4 \ \text{mm}^4$

Cross sectional area, $A = \frac{\pi}{4} \times d_2^2$

$= 0.785 \times d_2^2 \ \text{mm}^2$

Radius of gyration, $k = \sqrt{\frac{1}{A}}$

$= \sqrt{\frac{0.049 \times d_2^4}{0.785 \times d_2^2}}$

$= 0.25 \times d_2$

Length of the link, $L = 134.56 \ \text{mm}$

$= 67.28 \ \text{mm}$

$W_{cr} = \frac{\pi \times A \times d_2^2}{1 + a \left( \frac{d_2}{L} \right)^2}$

Substituting the value of $d_2 = 60 \ \text{mm}$, we have
\[ W_c = \frac{1.46 \times 10^6}{1.002} \]
\[ W_c = 1.456 \times 10^6 \text{ N} \]

VI. ANALYZIS:

A. Basic steps in analyzing:

(i) Initially, choose static structural from analysis systems.

(ii) Import the assembly model to be analyzed.

(iii) Choose the engineering data and add required number of materials. Enter the material properties like tensile strength, compressive strength, ultimate tensile strength, density.

(iv) Open the model and choose the corresponding material for each part of the model.

(v) Mesh the model by using quadrilateral or all triangular elements.

(vi) Supports should be provided at the screw rod, outer diameter mandrel pipe, front cover plate, rear cover plate and plain bearings.

(vii) The total load of 700kN should apply on the expandable mandrel segment.

(viii) Under solution choose total deformation, Stress and Strain.

(ix) Solve the model.

(x) View the results.

B. Conventional Mandrel:

Stress:

Strain:

Displacement:

Since the buckling load is more than the calculated value \((1.24 \times 10^6)\), therefore the link is safe for buckling in a plane perpendicular to the vertical plane.

C. Adjustable Mandrel:

Stress:

Strain:

Displacement:
TABLE II
Comparison

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Conventional mandrel</th>
<th>Adjustable mandrel</th>
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<tbody>
<tr>
<td>Stress</td>
<td>55.486 Mpa</td>
<td>265.3 Mpa</td>
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<tr>
<td>Strain</td>
<td>2.8 * e-4</td>
<td>1.4 * e-3</td>
</tr>
<tr>
<td>Displacement</td>
<td>0.2817</td>
<td>0.38452</td>
</tr>
</tbody>
</table>

VII. CONCLUSION:

The main objective of the project is to analyze the pipe bending process using mandrels and to minimize the difficulties faced during the bending process. The work level is reduced as much as possible by creating a design followed by analysis of the mandrel used presently and the newly designed mandrel. Implementation of suggested method resulted in achieving the following benefits:

1. Quantity of using various mandrels is limited.
2. Time for bending various pipes is reduced.
3. A range of pipes are bended using a single mandrel.
4. Costs of preparing various mandrels are reduced.
5. Stress, strain and displacement analysis is made and compared to check the effects of mandrels while bending.
6. REFERENCES

1. Chitoshimiki and et al. (2000) “Deformation and fracture properties of steel pipe bend with internal pressure subjected to in-plane bending”.

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