Stress and Springback Analyses of API X70 Pipeline Steel Under 3-Roller Bending via Finite Element Method

Wenbo Zhao¹ · T. Warren Liao¹ · Lampros Kompotiatis²

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Abstract 3-Roller bending is a widely applied manufacturing process, particularly in structural steel pipe industry. However, due to the difficulty and high cost of measuring stress distribution inside sheet material via traditional method, internal stress/strain response during forming is largely unexplored. The focuses of this study are two: (1) to map the radii of curvature as well as the stress inside the work piece during forming by utilizing the meshing mechanism of finite element method, and (2) to further provide some numeric guidelines for the configuration of the rolling system in order to improve production efficiency and product quality. The results of this study indicate that: (1) it is crucial to properly choose forming parameter in order to produce product with desired radii; (2) much like a gradual springback process, the radii of curvature gradually increase from the top roller to the exit-side bottom roller; (3) under the assumptions made in this study, to produce pipes with a specified diameter with varying configurations of the 3-roller system will not significantly change the final residual stress; and (4) finally, shifting of the neutral axis up to 2.0% of the thickness toward the compressing side during the forming process is observed.

KEY WORDS: Asymmetrical 3-roller bending; Residual stress; Springback; Pipe manufacturing

List of symbols

U	Y displacement of Roller 1 (top roller) between initial contact position and final rolling position (mm)	R_1	Radius of Roller 1 (mm)
A	The X component of distance between Roller 2 (bottom front) and Roller 3 (bottom rear) (mm)	R_b	Radius of Roller 2 and Roller 3 (mm)
S	The X component of distance between Roller 1 and Roller 2 (mm)	t	Thickness of work piece (mm)
Si	$\frac{S}{A}$ (no unit)	μ	Friction coefficient (no unit)
H	Y component of distance between Roller 2 and Roller 3 (mm)	V_{rel}	Relative speed between two contact surfaces when sliding (mm/s)
		FACT	Ratio of static to dynamic coefficient of friction (no unit)
		DC	Decay coefficient (no unit)
		r_0	Initial radii of curvature before springback (at point 1) (mm)
		r_f	Final radii of curvature after springback (after point 7) (mm)
		r_u	Upper limit of r_0 (mm)
		E	Young's modulus (MPa)
		K	Strength coefficient (MPa)
		K'	$K \left(\frac{2}{\sqrt{3}} \right)^{n+1}$ (MPa)
		n	Strain hardening exponent (no unit)

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✉ T. Warren Liao
ieliao@lsu.edu

¹ Department of Mechanical and Industrial Engineering,
Louisiana State University, Baton Rouge, LA 70803, USA

² Vice President of Technology, STUPP Corporation, 12555
Ronaldson Rd., Baton Rouge, LA 70807, USA

ν	Poisson's ratio (no unit)
σ_y	Yield stress (MPa)
ε_y	Strain at yield point (no unit)
D_0	$2r_0$ (mm)
D_f	$2r_f$ (mm)

1 Introduction

Medium-to-large-diameter cylindrical tubes are widely used in various applications such as gas and fluid transportation, tunnels construction and building construction. 3-Roller bending is a fast and cost-effective method to produce such mechanical parts [1]. A number of 3-roller bending studies have been performed via analytical, experimental as well FEM method. Hua *et al.* [1] developed an analytical model to predict the bending moment at the top roller contact point. Chudasama *et al.* [2] and Hua *et al.* [3] proposed two different analytical methods to estimate the bending force on rollers. Ktari *et al.* [4] studied the effect of roller configuration on the final diameter of cylindrical section using FEM simulation. Gandhi *et al.* [5] investigated the effect of the top roller relative position in the Y direction on the final diameter of cylindrical section in a symmetrical 3-roller system using both analytical and experimental methods. On the other hand, Feng and Champlaud [6] studied the effect of the lateral roller (Roller 3) position in an asymmetrical system using both FEM and experimental methods. However, most above-mentioned studies focus on the external response, such as bending moment, responding force and final diameter of formed section. The deformation and stress response while work piece is still inside the bending system are rarely studied. Most existing studies assume that springback occurs only after the work piece left the bending system. Later, our results will be presented to show that this assumption has been incorrect.

This paper presents the results of our study on the internal stress and external deformation of the work piece, not only when it is still inside the bending system but also after it leaves the bending system. Such a study is nonexistent in the open literature. In this paper, both symmetrical and asymmetrical 3-roller bending systems are studied using FEM implicit analysis. Figure 1 shows the basic configuration of a 3-roller bending system modeled. Roller 2 is fixed, and the center of Roller 2 is defined as the reference of the global coordinate system (0, 0, 0). The directions of x , y and z axis are also shown in Fig. 1.

The simplified forming process is described as follows. Roller 1 is first lowered down along y to bend the steel sheet by a displacement of U (negative). Next, the work piece is continuously fed into the system in the left-to-right direction shown in Fig. 1, and the worksheet is gradually bended by three rollers. Lastly the worksheet exits the roller system

in the cylindrical shape. The worksheet is fed in by rotating Roller 1; the friction between Roller 1 and worksheet will drive the worksheet into the bending system. More details about the movement of the three rollers can be found in Sects. 2.2.3 and 2.2.4. Based on the data provided by the STUPP Corporation, in order to manufacture cylindrical pipes of a pre-specified diameter within tolerable deviation, a proper configuration must be carefully chosen. The configuration parameters include the top roller (Roller 1) downward displacement, U , distance between Roller 1 and Roller 2, S , distance between Roller 2 and Roller 3, A , and y position of Roller 3, H . Generally lower residual stress is desirable because several studies have shown that high residual stress can cause or aggravate structural issues of API pipeline such as corrosion cracking [7], sulfide stress corrosion cracking and stress-oriented hydrogen-induced cracking [8], which is a significant contribution source of pipeline leaks and rupture [9]. This study sets out to investigate the effect of U , Si ($Si = S/A$) as well as H on final cylindrical diameter and stress distribution during and after forming. Since 3-roller bending of work piece can be approximately described as pure bending condition, only circumferential stress and strain are studied in this work. Hereafter, stress and strain refer to circumferential stress and strain components unless otherwise specified.

The major contributions of this study are twofold. First, the study provides insightful information about the changes in radii of curvature and in stresses as the worksheet moves through the 3-roller bending system, which have never been thoroughly studied before in the open literature and have been invalidly assumed constant in previous studies. Secondly, a more complete study is carried out to investigate the effects of a number of 3-roller bending configurations on final diameter as well as on residual stress. The effects of parameters such as H and Si on the two above-mentioned responses, which have never been discussed in the open literatures to the best of our knowledge, are discussed. In real-world production, carefully chosen H and Si are as important as the other parameters such as U and A , which have been the focus of a number of past studies. The remainder of the paper is organized as follows. Section 2 describes the basics of FEM Modeling for the study. Section 3 presents the simulation results obtained by the FEM models built for the studies together with discussion. In the last section, the paper is concluded.

2 FEM Modeling

Because the steel worksheet is fed in via friction force between the rollers and the worksheet, the subject modeling problem is essentially a contact problem. The details of FEM modeling are described in the following sections.

Fig. 1 Configuration of a typical 3-roller bending system

The geometries of the basic FEM model built in this study are as follows: The radius of Roller 1 $R_1 = 325$ mm; The radius of bottom two rollers: Roller 2 and Roller 3, $R_b = 286$ mm; The thickness of worksheet $t = 30$ mm; The above dimensions are taken from Ref. [4] with the intention to verify the model built in this paper with that reported in Ref. [4].

2.1 Assumptions

All studies presented in this paper were carried out under the following assumptions:

1. Plane strain condition in the XY plane is assumed. From the macroscopic point of view, the width change (deformation along the z axis) of steel sheet during 3-roller forming is small, which indicates small strain. Also since worksheet mainly endures bending moment within XY plane, strain in Z (transverse) direction is generally much smaller because of Poisson effect; more specifically, it is about 30% (0.3, Poisson's ratio) of strain in main loading direction, which is circumferential in this study. Strain in thickness direction is also much smaller than that in circumferential direction for the same reason.
2. No initial strain or residual stress in the worksheet before it is fed into the rolling system.
3. All three rollers are rigid bodies that are non-deformable, while the steel worksheet is a deformable body. Based on the data provided by the STUPP Corporation, rollers in their mill are made of 100Cr6 steel (German spec, SAE 52100 equivalent) which has much higher yield strength (typically higher than 2000 MPa) than the worksheet material (API X70 which has yield strength around 483 MPa). Therefore, the rollers will not reach the yielding point when bending the worksheet, which means that the rollers will always have strain far less than the worksheet.
4. The weight of worksheet is ignored.
5. Isotropic material properties are assumed. In reality, steel material properties are anisotropic. The study of Choi et al. [10] shows that X70 pipeline steel in the transverse (z) direction has higher yield strength and tensile strength than that in the longitude (x) direction by up to about 17%. Thus, anisotropic material model would be more appropriate. However, for simplicity this assumption was often made as in [1–5]. Since the plane strain situation is assumed and the current study focuses only on the material behavior in the circumferential direction, the assumption of isotropic material properties is justified because material properties in the transverse (z) direction and the thickness (y) direction are expected to have minimal effect on the results. We

hope to consider anisotropic material properties in the future work when 3-dimensional model is built.

6. Isotropic hardening is assumed. Since this study involves no cyclic loading, kinematic hardening or isotropic hardening will produce the same results. To save computing time, the simpler isotropic hardening model is hence chosen.
7. Static loading and unloading is assumed.
8. The change in worksheet thickness subject to the rolling process is neglected. As explained in assumption 1, strain in thickness direction is quite small.

2.2 Modeling Problem

2.2.1 Material Properties

The worksheet material of interest in this study is API 5L X70 pipeline steel. The material is defined by multilinear isotropic hardening model with Young's modulus of 210 GPa and Poisson's ratio of 0.3. The mechanical properties of the material are summarized in Table 1.

Strain hardening is described by a multilinear model beyond the yielding point. The uniaxial tensile test graph is shown in Fig. 2. Since this model involves large deformation, true stress (Cauchy stress) and logarithmic strain data are used [11]. Also, the Von Mises yield criterion is assumed.

2.2.2 Element Definition

The Plane183 8-node quadrilateral element in ANSYS is chosen to mesh the steel worksheet. PLANE183 is a high order element with structural degree of freedom as well as stress stiffening, large deflection and large strain capability. With the plane strain assumption made in Sect. 2.1, KEYOPT(3) = 2 is defined, which indicates each element is constrained to the plane strain condition with the Z component of strain = 0.

2.2.3 Contact Definition

Contact problems are highly nonlinear and often require carefully chosen parameters to avoid convergence difficulty and excessive computational time. In ANSYS,

contact is described by the interaction between two surfaces meshed by Target element and Contact element, respectively. In this specific modeling problem, contacting surfaces of rollers are meshed by Target element 169, while contacting surfaces of steel worksheet are meshed by Contact element 172. Roller 1 is designated as the driving roller that drives the worksheet into the bending system via friction. To this end, a high dynamic friction coefficient (MU) of 1.0 between Roller 1 and the inner surface of the worksheet is defined, while dynamic friction coefficient of 0.2 is defined between the worksheet and other rollers. To avoid convergence difficulty introduced by the sudden change of friction coefficients from static value to dynamic value when a contact surface reaches the sliding state, the decay coefficient (DC) is set to minimum, i.e., 0, and the ratio of static to dynamic coefficient of friction (FACT) is set to minimum value of 1.0. Equation (1) is the decay friction model used in ANSYS:

$$\mu = \text{MU} \times \left(1 + (\text{FACT} - 1)^{(-\text{DC} \times V_{\text{rel}})} \right), \quad (1)$$

where μ is the friction coefficient; and V_{rel} is the relative speed between the two contact surfaces in sliding.

Assuming $\text{DC} = 0$ and $\text{FACT} = 1$, then Eq. (1) is reduced to

$$\mu = \text{MU}, \quad (2)$$

which means that the friction coefficient is assumed to be equal to a constant value of dynamic friction coefficient without the effect of friction decay.

Fig. 2 Uniaxial tensile test data of API X70 pipeline steel

Table 1 Mechanical properties of API 5L X70 pipeline steel

Young's modulus (GPa)	Yield strength (MPa)	Ultimate tensile strength (MPa)	Poisson's ratio	Strength coefficient (MPa)	Strain hardening coefficient
210	520	770	0.3	883.2	0.1327

2.2.4 Boundary Condition and Load Definition

The motion of rigid surface of each roller is defined by the motion of a pilot node. By defining the boundary condition and loading on the pilot node, the boundary condition and loading on the corresponding roller are defined. To reduce computational time, Roller 2 and Roller 3 are set to have zero displacement in both the X and Y directions but can rotate freely about the Z axis. To accurately control the motion of Roller 1, tabular load is defined as illustrated in Fig. 3.

From 0 s to 20 s, Roller 1 moves U distance along the negative Y direction at constant speed $\frac{U}{20}$ mm/s and stays at that position from 20 s onwards till the end of the simulation (120 s). Roller 1 is set to have 0 rad of rotation about Z in the initial 20 s, and it rotates a total of positive 6 rad about Z at constant speed of $\frac{6}{100} = 0.06$ rad/s from 20 s onwards till the end of the simulation (120 s). Note that the steady state of the simulations is of interest, which means that the simulation results are time independent. The above-mentioned speeds are chosen in order to produce stable results with better convergence. The degree of freedom of the worksheet is constrained only by the contact between the three rollers and the worksheet.

2.3 Model Validation

Before studying the material of interest API 5L X70, the model is first verified by using the material data of S275JR reported in Ref. [4] with symmetrical setup and $U = -60$ mm. The final diameters obtained by our symmetrical model (with $H = 0$ mm, $Si = 0.5$) and the model reported by Ktari et al. [4] with identical roller configuration are given in Table 2.

From Table 2, it can be seen that both our symmetrical model and the model reported in Ref. [4] produce approximately the same final diameter with minimum

Fig. 3 Tabular load on Roller 1: Y displacement (*top*) and rotational angle about Z (*bottom*)

(<1%) difference. Since the symmetrical model in Ref. [4] has been experimentally verified, current model can be considered verified as well.

3 Results and Discussion

Once verified, the material data in the first built model is swapped from S275JR to API X70 as discussed in Sect. 2.2.1 for investigating the progressive change of radii and stresses inside the worksheet and the effects of setup configuration parameters, U , Si and H . Due to its high toughness, API X70 is one of the most widely used pipeline steels in the petroleum industry. According to the data provided by the STUPP Corporation, API X70 is the most frequently ordered steel grade in its mills. Section 3.1 describes the changes in radii of curvature and stresses between Roller 1 and Roller 3 during bending. Section 3.2 investigates the effect to U on pipe diameter as well as residual stress using the symmetrical ($Si = 0.5$ and $H = 0$) model. Sections 3.3 and 3.4 discuss the effect of Si and H , respectively, while keeping the remaining parameters fixed. Lastly, Sect. 3.5 presents the relationship between final pipe diameter and springback ratio, as well as the relationship between final pipe diameter and residual stress.

3.1 Radii of Curvature and Stresses in the Worksheet During Bending

To map the radii of curvature and stresses during the 3-roller bending process, several points on the worksheet along the circumferential direction are selected. As illustrated in Fig. 4, the point directly in contact with Roller 1 is designated as point 1, the point that is in direct contact with Roller 3 as point 7, points 2–6 in between points 1 and 7 and points 8 and 9 after point 7 with the same circumferential segment length.

The method used to calculate radii of curvature is described below:

One small segment of bended work piece with length dL is taken out for analysis as shown in Fig. 5. ε_i and ε_o are the circumferential mechanical strain on the inner and outer surfaces of the worksheet, respectively.

Table 2 Comparison of final diameters between our symmetrical model and the model reported in Ref. [4]

A (mm)	D_f (mm)	D_f [4] (mm)	Difference (%)
750	1367	1378	0.80
700	1172	1180	0.68
650	993	1000	0.70

Fig. 4 Illustration of selected positions

Fig. 5 Illustration of radii calculation

After bending, the inner surface of segment will have length $dL(1 + \varepsilon_i)$, whereas the outer surface will have length $dL(1 + \varepsilon_o)$. Since the change in thickness is neglected, the bended worksheet maintains its initial thickness t . Therefore, the following relationship between inner radii and outer radii holds: $r_o = r_i + t$.

It is obvious that:

$$dL(1 + \varepsilon_i)/dL(1 + \varepsilon_o) = r_i/r_o = r_i/(r_i + t),$$

After simple derivation, we have:

Fig. 6 Changes in radii of curvature during springback (numbers in parenthesis indicate the selected points) ($A = 750$ mm, $U = -50$ mm, $Si = 0.5$, $H = 0$ mm)

$$r_i = t(1 + \varepsilon_i)/(\varepsilon_o - \varepsilon_i),$$

Therefore, by recording the strain at every point on the inner and outer surfaces, the radii of curvature can be easily calculated.

3.1.1 Change in Radii of Curvature During Bending

Figure 6 shows a typical example of change in the radii of curvature from point 1 to point 9 when the setup configuration is as follows: $A = 750$ mm, $U = -50$ mm, $Si = 0.5$ and $H = 0$ mm. From point 1 to point 7, the radii of curvature gradually increase from 724 to 900 mm and then stay at the final value of 897 mm at points 8 and 9. This means that in continuous 3-roller bending, the springback process mainly occurs between point 1 and point 7, namely between Roller 1 and Roller 3. This is expected since the bending moment gradually decreases from point 1 to point 7 (where the bending moment becomes 0). This is similar to a graduate unloading process where load (or bending moment) is slowly removed. The springback ratio ($\frac{r_i - r_o}{r_i} \times 100\%$) in terms of radii of curvature is 19.3% in this specific case.

Based on the assumptions of linear elasticity, plane strain condition and no yielding region as shown in Fig. 2, Hosford and Caddell [12] developed a relationship between radius of curvature before springback (r_o) and final radius after springback (r_f):

$$r_o/r_f = 1 - \left[\frac{6K't^{n-1}}{r_o^{n-1}(n+2)2^n} \left(\frac{1-\nu^2}{E} \right) \right], \quad (3)$$

where E is Young's modulus, MPa; K is strength coefficient, MPa; n is strain hardening exponent, no unit; $K' = K \left(\frac{2}{\sqrt{3}} \right)^{n+1}$, MPa; ν is Poisson's ratio, no unit; t is worksheet thickness, mm.

Based on the assumptions of elastic—perfectly plastic material model (i.e., no strain hardening) and static loading and unloading, Bouhelier [13] proposed another model governing the radius relationship before and after springback:

$$r_f = r_0 \left/ \left[1 - \frac{3}{2} \left(2 \frac{r_0}{t} \cdot \frac{\sigma_y}{E} \right) + \frac{1}{2} \left(2 \frac{r_0}{t} \cdot \frac{\sigma_y}{E} \right)^3 \right] \right., \quad (4)$$

where σ_y the yield stress (MPa) and other variables are kept the same as in Eq. (3).

While bended, assuming the neutral axis is at the center of worksheet thickness, the circumferential strain on the outer surface is approximately t/D , where t is the worksheet thickness and D is the diameter of cylindrical section. In this study, typically t/D is between 1.2% and 4%. From Fig. 2, it can be seen that strain magnitude less or equal to 4% lies mostly in the yielding zone with little strain hardening effect. Therefore, theoretically speaking the model proposed in Ref. [13] is more appropriate to be used for model verification in this study. Figure 7 shows the radii of curvature results obtained in over 100 simulations with different combinations of U , S , A and H , to be discussed in details in Sects. 3.2, 3.3 and 3.4, in comparison with those derived from Eq. (4).

It is clear from Fig. 7 that the FE model developed in this study agrees very well with the springback model reported in Ref. [13] with the maximal difference of only 1.4%. This further confirms the accuracy of our models built for this study. Moreover, the results strongly suggest that the worksheet indeed mainly undergoes elastic—perfectly plastic phase, without much strain hardening effect.

3.1.2 Stress Responses During Springback

Springback essentially is deformation caused by the release of residual stresses. Figure 8 shows typical circumferential stress distributions through the worksheet thickness during

Fig. 7 Comparison of r_f between our FE model and the model reported in Ref. [13]

bending with springback in action when the setup configuration is symmetrical as: $A = 750$ mm, $U = -50$ mm, $Si = 0.5$ and $H = 0$ mm. At point 1 (blue Line in Fig. 8), the outer surface has the largest absolute tensile stress value (659 MPa), while the inner surface has the largest absolute compressive stress value (-661 MPa). The Euler–Bernoulli beam theory (a.k.a classical beam theory) indicates that under bending, the strain at a particular point kp in a 2D beam is proportional to the distance between that point and the neutral axis [14]. Since inner and outer surfaces are farther away from the neutral axis, they are expected to have the highest stress and strain magnitudes as well. If the neutral axis is at the center of the worksheet thickness, the inner and outer surfaces will have the same distance from the neutral axis, hence identical stress and strain magnitudes (albeit it is compressive on the inner surface and tensile on the outer surface).

However, as indicated by the red vertical dash line in Fig. 8, the neutral axis is not at the center of worksheet thickness; rather it is actually shifted toward the inner surface (compressing side) by 0.25 mm (about 0.83% of the thickness t). This observation is consistent with the study of Johnson and Meller [15], in which they found that during cold forming, the neutral axis shifts a bit toward the inside surface. Compared to the outer surface, the inner surface is slightly closer to the neutral axis. Considering the material model given in Fig. 2 and the Euler–Bernoulli beam theory, the inner surface is expected to have smaller absolute strain value than the outer surface, hence smaller absolute stress value as well.

At points 7, 8 and 9, the springback process has been fully completed and stresses left inside the worksheet are the residual stresses if the worksheet is not to be further processed. In actual pipe manufacturing, subsequent operations such as annealing are necessary to further reduce

Fig. 8 Circumferential stress distributions through thickness during springback ($A = 750$ mm, $U = -50$ mm, $Si = 0.5$, $H = 0$ mm)

residual stress. Other post-forming operations such as welding, hydraulic pressure test, sizing and straightening may also alter the residual stresses. Therefore, the residual stresses in the worksheet are expected to be further modified. Without considering subsequent processing operations, the distribution of residual stresses from the outer surface to the inner surface shows an alternative pattern of compressive and tensile. Our result is consistent with that of Rondal [16], in which a similar behavior of residual stress in cold roll forming was analytically studied and reported. More discussions about the effect of roller configurations on residual stresses are presented in Sects. 3.2, 3.3 and 3.4. To simplify the characterization of residual stresses, the stresses having the highest magnitude in both tensile and compressive (point *a* and point *b* shown in Fig. 8) are selected for discussion. In this specific case, the tensile residual stress at point *a* is 402.5 MPa, while the compressive residual stress is −444.1 MPa at point *b*.

During the springback process (from points 1–7), it is clear that the absolute value of stress at any point of the worksheet thickness decreases. The regions close to the neutral axis (12.5–17.7 mm) are dominated by elastic strain and far less affected by springback than the regions that are far away from the neutral axis (0–12.5 and 17.7–30 mm) which are dominated by plastic strain instead. One interesting behavior worth noticing is that on the outer and inner surfaces where stresses experience the most dramatic changes in springback, the stress polarity is reversed. Specifically, the outer surface has positive (tensile) stress of 659 MPa at point 1 (before springback) and negative (compressive) stress of −223.4 MPa at point 7 (after springback), which amounts to −882 MPa of change as a result of springback. On the other hand, the inner surface has negative (compressive) stress of −661 MPa at point 1 and positive (tensile) stress of 234 MPa at point 7, which amounts to 895 MPa of change as a result of springback. Goes et al. [17] observed similar behavior while studying the residual stress of elastoplastic wires subjected to bending. According to Marciniak et al. [18], in which elastic–perfectly plastic worksheet and the neutral axis at the center of the thickness are assumed, changing of stress on the outer surface as a result of springback is: $\Delta\sigma_o = -\frac{3}{2}\sigma_y = -795$ MPa, which is close to the observed outer surface stress change of −882 MPa in the subject study. Assuming perfectly elastic springback behavior, the strain change as a result of springback should be:

$\Delta\varepsilon = |\sigma_1 - \sigma_7|/E \approx 0.43\%$. The total mechanical strain on the worksheet surface after forming is:

$\varepsilon_t \approx t/2 \times \text{Radii} \approx 1.67\%$, which means that the total strain change as a result of springback is: $\Delta\varepsilon = \frac{\Delta\varepsilon}{\varepsilon_t} \times \varepsilon_t \approx 25.7\%\varepsilon_t$. This agrees well with the simulated strain output shown in Fig. 9, i.e., initial strain of

Fig. 9 Strain change during springback

2.05% (at point 1) before springback and final strain of 1.67% (22.7% decrease) after springback (at points 7, 8 and 9).

3.2 Effect of *U* (Top Roller Displacement in the *Y* Direction)

In this section, the basic symmetrical model presented in the previous section is simulated to study the effect of *U* on final diameter and residual stress, as presented and discussed in Sects. 3.2.1 and 3.2.2, respectively.

3.2.1 Effect of *U* on Final Cylindrical Cross-Section Diameter

As shown in Fig. 10, a number of simulations were performed by varying the top roller relative displacement distance (*U*) from −80 to −30 mm in an increment of 5 mm at 4 different distance-spans between the two bottom rollers: 750, 700, 650 and 600 mm. The final diameter D_f results range from 700 to 3200 mm. These results agree well with the analytical results reported in Ref. [5]. Generally speaking, lower absolute relative displacement in the *Y* direction ($|U|$) leads to larger diameter. While *U* is kept constant, increasing the distance between the two bottom rollers (*A*) results in larger diameter. The results also suggest that different combinations of *U* and *A* values can produce the same D_f of cylindrical cross-sections.

Theoretically speaking, it is possible to produce large diameter with smaller *A* and $|U|$. However, it is generally a better practice to use a larger *A* because it allows the use of large $|U|$ as well. The large gradient of D_f versus *U* ($dD_f/dU > 120$ mm/mm) at smaller $|U|$ implies that a small change in $|U|$ will greatly change the resultant diameter. In other words, it is more difficult to adjust $|U|$ to make a large pipe when *A* is small. Conversely, if the desired diameter is small, a smaller *A* value is generally preferred, even if the combination of large *A* and $|U|$ can produce the same diameter. The reasons are: (1) Large

Fig. 10 Effect of top roller displacement in the Y direction (U) on final diameter D_f

A and $|U|$ call for unnecessarily large rolling machine which takes up more space and can be more expensive and/or difficult to fabricate, transport and install; and (2) when A is large, to produce small diameter, a very large $|U|$ is required. As can be seen from Fig. 10, at large $|U|$ the gradient of D_f versus U can be very small. This implies that a small change in D_f required a large change in U . This can also make the practice of adjusting U to obtain certain D_f unnecessarily harder. Of course, every 3-roller machine in the real world is designed to operate within a limited range of diameters, occupying a tiny window of the entire range shown in Fig. 10.

3.2.2 Effect of U on Residual Stress

While performing simulations as described in Sect. 3.2.1, residual stresses as described in 3.1.2 were also recorded. The obtained results are shown in Fig. 11.

The top plot of Fig. 11 indicates that tensile residual stress decreases with increasing $|U|$, except when $|U|$ is smaller than 40 mm. For any $|U|$ fixed at value larger than 40 mm, configuration with larger bottom-roller distance (A) leads to higher tensile residual stress. Recall that a similar trend was observed between U and final diameter as discussed in Sect. 3.2.1. As a matter of fact, it is found that residual stress is correlated to final diameter and not dependent on roller configuration at all; more details are given in Sect. 3.5.2.

The absolute compressive residual stress shown in the bottom plot shows a barely noticeable opposite trend (increasing residual stress with increasing $|U|$) with relatively smaller amount of variation from 440 to 460 MPa. Compared to tensile residual stress, the magnitude of absolute compressive residual stress is almost always higher. This result seems to be inconsistent with the observation made in Sect. 3.1 that the outer half has larger stress magnitude than the inner half with the same distance to the center of the worksheet thickness due to the shifting of the neutral

Fig. 11 Relative displacement in the Y direction (U) versus residual stress (top tensile stress; bottom compressive stress)

axis toward the inner surface (compressive side) under bending. However, point a and point b are not chosen to be symmetrical to the center of thickness (as shown in Fig. 8), hence explaining why the tensile stress is not necessarily larger.

Also noticeable in Fig. 11 is that when $|U|$ is between 35 and 45 mm, residual stresses experience unexpected fluctuations, especially for configurations with larger bottom-roller distances ($A = 750$ mm and $A = 700$ mm). This is partly because when $|U|$ is small, as discussed in Sect. 3.2.1, the final diameter is very sensitive to U ; therefore, a small fluctuation of U can cause a great fluctuation in D_f . In other words, the system becomes less stable under these conditions. Also, high springback ratios ($>18.6\%$) are observed when $|U|$ is small (3–45 mm) and A is large (700 and 750 mm). As discussed in Sect. 3.2.1, both large bottom-roller distance (A) and smaller $|U|$ lead to an increase in final diameter ($2 \times r_f$). In fact, there exists a positive correlation between final diameter and springback ratio, which is discussed in Sect. 3.5.1. The exact cause of this unstable residual stresses under this particular combination is unclear at this moment. Further investigation of this unexpected behavior is needed.

3.3 Effect of S_i (The x Position of the Top Roller)

In this section, the effects of varying S_i will be discussed with U fixed at -60 mm and A set equal to 750 mm. Eleven simulations were performed with S_i varied from 0.375 to 0.625; namely, the distance between Roller 1 and Roller 2 along the x -axis is varied from $0.375A$ to $0.625A$, where A is the distance between Roller 2 and Roller 3. Another set of 11 simulations is repeated with another A value at 700 mm.

Section 3.3.1 focuses on the effect of S_i on final diameter (D_f) while the effect of S_i on residual stress is discussed in Sect. 3.3.2.

3.3.1 Effect of S_i on Final Diameter

Figure 12 shows the relationship between S_i and final diameter (D_f) for two different A values.

As can be seen from Fig. 12, varying S_i can also greatly change the final pipe diameter. This behavior can make the roller bending system more flexible without the need to make it much bigger. For example, in this specific system, if a larger diameter is desired, the operator has the option to position the top roller at around 0.42 to avoid choosing a larger A or smaller $|U|$. As discussed in Sect. 3.2, large A values can make the system larger and heavier, while small $|U|$ can cause instability.

Regardless of the A value, the final diameter is maximized when S_i is around 0.42, i.e., when the top roller (Roller 1) is positioned a little toward the negative x direction. The final diameter will be smaller if S_i is deviated farther away from 0.42. This is interesting because larger diameter is generally expected when the roller configuration is symmetrical ($S_i = 0.5$). This behavior can be partly explained by the fact that when $S_i = 0.5$, the worksheet in-between the two bottom rollers, Roller 2 and Roller 3, under bending is not actually symmetrical about Roller 1. Rather, when $S_i = 0.5$ the x coordinate of the point on the worksheet that has the lowest y value is around 0.557. On the other hand, when $S_i = 0.425$, the x coordinate of the point on the worksheet that has lowest y value is around 0.52. The above result shows that $S_i = 0.425$ leads to a more symmetrical worksheet profile in-between Roller 2 and Roller 3 than when $S_i = 0.5$.

It is also clear from Fig. 12 that when S_i is fixed, larger A values lead to larger final diameters, which is consistent with the results discussed in Sect. 3.2.1.

3.3.2 Effect of S_i on Residual Stress

As shown in the top plot of Fig. 13, tensile residual stress drops nonlinearly when S_i increases from 0.375 to 0.625 (S from 0.375A to 0.625A), while the absolute value of compressive residual stress (the bottom plot) increases

Fig. 12 S_i versus final diameter

Fig. 13 S_i versus residual stress (top tensile stress; bottom compressive stress)

nonlinearly with increasing S_i . This is partly because the neutral axis is shifted more away from the center of the worksheet thickness when S_i increases from 0.375 to 0.625. The higher the shift in the neutral axis means higher stress asymmetry about the center of the worksheet thickness, which can further lead to different stress behaviors between the inner half and the outer half of the worksheet. More specifically, when S_i is at 0.375, the neutral axis is shifted by 0.25 mm (0.83% of thickness) toward the inner surface (compressive side), whereas at 0.625, the neutral axis is shifted by 0.45 mm (1.5% of thickness) toward the inner surface.

The bottom plot of Fig. 13 indicates that there is an interaction effect between S_i and A . The effect of S_i on compressive residual stresses seems to be more sensitive when A is smaller and more subdued when A is larger.

There is a strong similarity between the trend of S_i versus residual stress discussed in this section and the trend of S_i versus final diameter discussed in Sect. 3.3.1. In other words, both residual stress and final diameter generally decrease with increasing S_i . This similarity suggests a positive correlation between final diameter and residual stress, which will be further discussed in more details in Sect. 3.5.2.

3.4 Effect of H (the Y Distance between Roller 3 and Roller 2)

This section focuses on the effect of H on final diameter and residual stress. In this set of simulations, A is fixed at 750 mm and 13 simulations were performed by varying H from -30 mm to 30 mm (negative value means Roller 3 is lower than Roller 2) for each of the three values of $S_i = 0.4, 0.5$ and 0.6 . In Sect. 3.4.1, the effect of H on final diameter is discussed, whereas in Sect. 3.4.2 the results related to the effect of H on residual stress are described.

3.4.1 Effect of H on Final Diameter

The effect of H on final diameter is highly dependent on the value of Si . As shown in Fig. 14, when $Si = 0.5$ or 0.6 , final diameter decreases almost linearly (linear regression models were fitted with R -square value of 0.9958 and 0.9978, respectively) with increased H . However, when $Si = 0.4$, final diameter increases with H . This phenomenon is probably related to the discussion in Sect. 3.3.1 that when Si is close to 0.42, the worksheet profile in-between Roller 2 and Roller 3 is approximately symmetrical. As Si deviates away from 0.42, the more asymmetrical the worksheet profile in-between Roller 2 and Roller 3 will be. This agrees with the observation made in Fig. 14 that when Si increases from 0.42 to 0.6, the absolute value of the tangent lines on the H versus D_f curve gradually increases, which indicates increasing influence of H on D_f . This suggests that the more symmetrical the worksheet profile in-between Roller 2 and Roller 3, the less effect of H on D_f is. Figure 13 also suggests that for some final diameter values, more than one combination of H and Si values can be used. Also worth noticing is that the gradient of the H versus final diameter curve is quite moderate; this means operators will be able to fine tune the final diameter by adjusting the value of H without much difficulty.

3.4.2 Effect of H on Residual Stress

The relationship between H and tensile residual stress shown in the top plot of Fig. 15 exhibits a similar trend of H versus D_f discussed in Sect. 3.4.1. With $Si = 0.5$ or 0.6 , tensile residual stress decreases with increasing H , whereas when $Si = 0.4$, tensile residual stresses slightly increase at first and then slightly decrease. The similarity once again indicates some correlation between final diameter and residual stress, as mentioned in Sects. 3.2.2 and 3.3.2.

The absolute value of compressive residual stress shown in the bottom plot of Fig. 15 indicates an overall increasing trend, whereas the tensile residual stress in the top plot suggests an overall decreasing trend, especially when

Fig. 14 H versus final diameter

Fig. 15 H versus residual stress (upper tensile stress; lower compressive stress)

$Si = 0.6$ (red line). This might be due to the large shift of the neutral axis when $Si = 0.6$. As described in Sect. 3.3.2, significant shift of the neutral axis has been observed. Specifically, when $Si = 0.6$, the neutral axis is shifted by 0.3 mm toward the inner surface at $H = -30$ mm and by 0.6 mm toward the inner surface at $H = 30$ mm. The larger shift of the neutral axis away from the center of the worksheet thickness means higher stress imbalance between the inner half of the thickness and the outer half of the thickness.

3.5 Relationship Between Final Diameter and Springback Ratio and Relationship Between Final Diameter and Residual Stress

The effect of springback ratio, computed as $(r_f - r_0)/r_f = (D_f - D_0)/D_f \times 100\%$, on final diameter is described in Sect. 3.5.1, whereas the relationship between final diameter and residual stress is described in Sect. 3.5.2.

3.5.1 Relationship Between Final Diameter and Springback Ratio

As mentioned in Sect. 3.2.2, some correlation between final diameter and springback ratio is expected. Figure 16 plots final diameter versus springback ratio.

Figure 16 clearly shows that a strong positive correlation indeed exists between final diameter and springback ratio. There is negligible variation in the springback ratios for any chosen final diameter. In fact, for an elastic-perfectly plastic material, Eq. (4) clearly indicates that for a given fixed r_f , r_0 can be determined. As a result, the springback ratio will be determined as well. With continuous increase in the final diameter, it is also possible for the corresponding springback ratio to reach 100%. As a matter of fact, when the initial bending moment is not large enough to introduce plastic strain inside the worksheet,

Fig. 16 Final diameter versus springback ratio

deformation is purely elastic and the worksheet will springback to its original flat profile after load is removed.

For the material considered in our FE models, the maximum elastic strain (strain at the yield point) is $\varepsilon_y = 0.33\%$, as shown in Sect. 3.1.1. Assuming that the neutral axis stays at the center of the worksheet thickness, when the surface strain of the worksheet is at the maximum elastic state, the approximate initial radius of curvature r_0 is $\frac{t}{2 \times \varepsilon_y} \approx 4500$ mm which is the upper limit of r_0 for the worksheet to have any plastic deformation, denoted as r_u . To have some amount of plastic deformation, the bending system should bend the worksheet to the extent with initial radius of curvature $r_0 < r_u$. Also note that r_u is the upper limit of r_0 ; when r_0 is close to r_u , the springback ratio may become too large to retain controllable bending results. For example, in the current study, when $r_0 = 1102$ mm, which is only $\frac{1}{4}$ of the theoretical r_u , the springback ratio is already 29.1%. Furthermore, as discussed in Sect. 3.2.2, a high springback ratio may cause instability in the bending process, which might start even when r_0 is much lower than r_u . Further studies are required to determine the threshold of r_0 to obtain and maintain stable bending results.

3.5.2 Relationship Between Final Diameter and Residual Stress

Based on the results presented and discussed in the previous sections, high correlation between final diameter and

Fig. 17 Final diameter versus residual stress

residual stress is expected. Figure 17 plots final diameter versus residual stress with more than 100 data points composed of different combinations of U , Si , A and H values as discussed in Sects. 3.2, 3.3 and 3.4.

The plots indeed show high correlation between final diameter and residual stress, tensile or compressive. As shown in the top plot of Fig. 17, tensile residual stress increases with final diameter, whereas the bottom plot of Fig. 17 indicates that the absolute value of compressive residual stress slightly decreases with increasing final diameter, except those few really high ones. Figure 17 shows some variation of residual stress for a given final diameter. However, when the final diameter is within [800, 1578 mm] domain, the relative standard deviation of the residual stress at a given diameter is less than 3.12% which means the variation is relatively small. There is not enough data for analysis for the situations where final diameter is larger than 1578 mm. Also, as discussed in 3.2, results with large diameter are often unreliable and should be processed with extra caution. When $D_f < 800$ mm, the distance between Roller 1 and Roller 2 and/or Roller 3 becomes almost too small for the worksheet to pass through, which in turn causes convergence difficulty and unreliable results. Based on the data provided by the STUPP Corporation, it is also not practical to produce pipes with such a small diameter using the roller configuration described in the subject study. Given the assumptions made in Sect. 2.1, it is clear from Fig. 16 that different configurations of three rollers that produce the same final diameter will generate approximately the same amount of residual stress in the formed worksheet. This leads to the conclusion that it is impractical to try to achieve lower residual stress by changing the configuration of rollers if the desired final diameter is fixed in a single pass 3-roller bending system described in this study. However, several other factors can potentially change the residual stress state if considered. For example, in the real-world production, the worksheet starts in the coil form and has to be uncoiled and flattened before it enters the bending system. In other words, the initial worksheet is not stress-strain-free as assumed in this study. After bending, there might be subsequent processing operations such as welding, annealing, sizing and testing. By studying and optimizing all the relevant factors/parameters in those processes, the better ways to minimize residual stress in the worksheet can possibly be found.

4 Conclusions

This paper has presented a thorough fundamental study of 3-roller bending process using finite element analysis. The results obtained by the 2D FEM models built in this study are verified with those obtained in previous studies with

finite element, experimental as well as analytical models whenever possible. The following important conclusions are derived from the results obtained from our models and simulations:

- (1) Carefully chosen values of A , U , Si and H are required in order to produce a certain diameter accurately and efficiently. Building a model to relate final diameter with parameters U , Si , A and H can provide some helpful guideline for small scale manufacturers to better configure their rolling machine to produce quality product with desired diameter. It can also be meaningful for engineers to design roller bending system with smaller size, higher flexibility as well as better production stability.
- (2) For the material model shown in Fig. 2, the worksheet with low thickness/diameter ratio (<0.05 in this study) subject to the 3-roller bending process for forming cylindrical shape as shown in Fig. 1 mainly undergoes elastic–perfectly plastic deformation without significant strain hardening effect.
- (3) Shifting of the neutral axis toward the compressive side (inner surface) up to 2.0% of thickness is observed, which causes asymmetrical distribution of stress and strain with respect to the center of the worksheet thickness.
- (4) Contrary to the assumptions made in most existing studies, the radius of curvature for the portion of the worksheet in-between Roller 2 and Roller 3 is not constant; rather, it increases almost linearly from Roller 2 to Roller 3.
- (5) During springback, stresses within the elastic strain dominated zones (zones close to the neutral axis) do not undergo much change, whereas in the plastic-strain-dominated zones (zones close to the surfaces of the worksheet), their absolute values dramatically decrease. The stresses on the inner and outer surfaces experience the greatest changes to the extent that the polarity is reversed.
- (6) A strong positive correlation exists between final diameter and springback ratio. Assuming that the neutral axis lies at the center of the worksheet thickness, the initial radius of curvature r_0 has an upper limit of $r_u \approx \frac{t}{2 \times \epsilon_y}$. A threshold for r_0 lower than r_u exists in order for the bending system considered in the subject study to produce stable bending results.
- (7) High correlation between the final diameter and the residual stress is observed. This implies that any attempt to reduce residual stress by changing roller configuration is impractical, if the desired final diameter is fixed. Additional parameters and/or processing techniques must be introduced in order to reduce residual stress in 3-roller bending modeled in this study.

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