Novel roll stand for flexible profile bending

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Abstract. A new method and roll stand for the manufacturing of structural profiles with tailored dimensions and properties based on the combination of rolling and profile bending is developed. Through this merged process the outer contour of a semi-finished product is shaped by a set of driven flexible and rigid rollers. The profile inner side is supported by an adjustable mandrel, so that an arbitrary variation of the cross-section along the profile can be achieved. Additionally, a moveable guiding roll facilitates the bending of the product. Conventionally, the manufacturing of load-optimized bent profiles requires a sequence of operations, whereas the novel method enables the manufacturing of those components in one process step. This leads to a reduction in tooling cost and setup time. Hence, the production of lightweight structures in small batch sizes is facilitated. In this contribution, the new forming method and the design of the developed roll stand are presented alongside with an investigation of the effects of stress superposition on bending stresses during combined bending and rolling as well as the possible processes applications.

1. Introduction
Modern lightweight structures utilized in the automotive industry require complex load optimized parts with a high level of functional integration [1]. Usually they are made from high-strength steels or aluminium alloys. In order to meet these demands 3D bent profiles with varying cross sections and tailored material thickness are deployed [2]. The fabrication of bent profiles with tailored dimensions and properties is commonly realised within process chains, where an initial (flexible-) rolling step is followed by sheet bending or roll forming, then a welding step is carried out, succeeded by profile bending and a final hydro-forming step [3]. This kind of process chain lacks flexibility, is accompanied by high tooling cost and therefore is not feasible for smaller batch sizes limiting the use of structural optimized components to high-volume applications. In the past different process combinations where forwarded to approach the flexible production of bent profiles with tailored cross section dimensions for smaller batch sizes. Hermes et al. [4] proposed the incremental tube forming process (ITF), which is a combination of kinematic bending and spinning, allowing the manufacturing of bent tubes with varying diameters and material thickness in one process step at low tooling cost [5]. However, due to the incremental forming principle the productivity of ITF is comparatively low and the products final cross section is restricted to a circular shape. Hara et al. [6] established a extrusion process for square pipes with variable outer and inner dimensions using a movable mandrel and adjustable die segments which then can be combined with bending during extrusion [7]. Here major drawbacks are given in the area of material choice, as the extrusion process requires a low flow stress in the forming zone [8]. Staupendahl et al. [9] presented a method for the integration of roll forming into TSS-bending using the rollers of the
bending machine feeding system for a tube contour shaping prior to profile bending. Within this process the contour forming capability is restricted by the geometry and the fixed planar arrangement of the feeding rollers.

The different approaches for a flexible manufacturing of bent profiles with tailored dimensions and properties are all accompanied by unique drawbacks making them only suitable for specific applications. The aim of this paper is therefore to introduce a novel process which exploits the flexible manufacturing of load optimized structures in small batch sizes, for a wide variety of applications.

2. Profile bending with flexible rolling

2.1. Principle of profile bending with flexible rolling

During profile bending with flexible rolling a straight profile with constant wall thickness is drawn by driven rollers into the nip (figure 1) [10]. Here the profiles cross-section is formed by a set of flexible and rigid rollers enabling a gap free enclosing of the profiles outer contour. Due to the adjustable positioning of the rollers, the horizontal and vertical size of the rolling gap can be modified independently during the forming process. Simultaneously the profiles inner side is supported by a movable mandrel (figure 1 (b)). The geometry and engagement of the mandrel along the x-direction together with the rolling gap size is determining the rolled wall thickness. By moving a bending roller, which is placed behind the rolling gap exit, in the x-y-plane the bend curvature can be controlled.

![Figure 1](image)

**Figure 1** (a) Arrangement of the rollers for profile bending with flexible rolling; (b) Positioning of the mandrel for the adjustment of the rolled material thickness

2.2. Stress superposition in profile bending with flexible rolling

During bending with flexible rolling the rollers load the workpiece with the force $F_R$ (figure 2 (a)) and therefore superpose additional compressive stresses to the bending stresses, which are introduced by loading the workpiece with bending momentum $M_B$. The superposed stresses are acting orthogonal to the bending stresses in the sheet plane of each profile segment and additionally, when placing a mandrel inside the rolling gap, in material thickness direction. As a result, the yield locus shifts during combined bending and rolling along the Mises yield surface to lower bending stresses by the amount of $\Delta \sigma_{xx}$ (figure 2 (b)). According to the elementary bending theory [11], the bending moment can be calculated by integrating the bending stresses $\sigma_{xx}$ of the cross-section increments $dA$ and weighting them with their distance to the neutral fiber $d_N$:

$$M_B = \int_A \sigma_{xx} \cdot d_N \cdot dA$$  

(1)

Therefore, by decreasing the magnitude of the bending stresses a reduction of the bending moment can be achieved. This aspect has several advantages. Tozawa and Ishikawa [12] demonstrated that by lowering the bending moment reduced cross section distortion and springback can be obtained. In analogy Becker et al. [5] presented the suitability of stress superposition for the bending of high-strength
steel tubes with a dual-phase microstructure (UTS = 870 MPa). Tekkaya et al. [13] proved that with stress superposition during sheet metal bending the forming-induced damage can be delayed and the performance of components in fatigue tests improved.

**Figure 2** (a) Stress states in the outer an inner arc of the profile during combined bending and rolling without the usage of a mandrel; (b) 2D Mises yield surface for conventional and combined bending

For numerical modelling of the bending stresses $\sigma_{xx}$ in kinematic profile bending with combined rolling, the FEM was applied (solver: Abaqus/Explicit) with the parameters given in (figure 3). For a reduction of the computational effort a symmetry with respect to the $x$-$y$-plane is assumed. Therefore, the simulation of the initial bent-up-step, in which the bending roller is engaged in the direction of the workpiece, can be avoided. The tools (roller, bending roller and mandrel) are modelled as rigid bodies with R3D4-elements whereas the workpiece is modelled by an isotropic elastic-plastic material behavior with hardening according to the Gosh-extrapolation (figure 3 (b)). Effects of kinematic hardening and the anisotropy of the profile material are neglected. The workpiece is meshed with C3D8R-elements with an element size of 0.5 mm in order to avoid a strong distortion of the elements and for enabling the evaluation of the bending stresses over the material thickness. The bending roller is positioned in such a way that the required bending radius under load is facilitated and the rollers are driven in order to draw the workpiece into the rolling gap with a speed of 20 mm/s. Deviant from figure 1, a cuboid mandrel is placed in front of the rolling gap entry for absorbing the bending forces and supporting the profile before it enters the nip. Within the nip itself the profile is not supported from the inside. The friction between the rollers and the workpiece as well as between the mandrel and the workpiece is represented by the means of Coulomb friction law. The friction values were chosen according to Kalpakjian et al. [14] assuming a dry roller-workpiece-contact and a lubricated mandrel-workpiece-contact on the profile inside.

In figure 4 (b) the distribution of bending stresses $\sigma_{xx}$ at the rolling gap exit ($y$-$z$-plane) during bending of a square tube to a loaded neutral fiber bent radius $R_L$ of 800 mm with (bottom) and without rolling (top) is given. To ensure the comparability of the results, a profile with a cross-section equivalent to the one of the already tapered profile was simulated for the pure bending case. Through tapering the profiles edge lengths $a_0$ and $b_0$ by 10 % via rolling during bending a reduction of the maximum bending stresses magnitude in the inner and outer arch by 35 % was achieved. A shift of the neutral fiber could not be detected. Therefore, accordingly to equation (1), since the same cross-section for bending was examined, the required bending moment was decreased. By evaluating the forces acting on the bending
roller and assuming the lever arm length to be equal with the distance from the point of force application at the bending roller to the rolling gap exit (y-z-plane) the bending moment can be calculated. With those assumptions, a 78 % decrease of the bending moment can be assessed for a tapering of the profile by 10 % during bending of the radius $R_L = 800$ mm.

The normalized bending moment represents the ratio between the bending moment for bending with $(M_{b,R})$ and without $(M_{b,Con})$ simultaneous tapering by rolling. By the means of raising the tapering ratio for a constant normalized bending radius $R_L/a_0$ of 10 the normalized bending moment is reduced to a value of 0.3 for $\Delta a = \Delta b = 15 \%$. This reduction is attributable to the superposed compressive stresses that increase with the tapering ratio and therefore cause a further shift of the yield locus to lower bending stresses. Furthermore, for a constant tapering ratio of 10 % the normalized bending moment decreases with the tapering ratio. In order to determine the bending moment reduction for different tapering ratios with $\Delta a = \Delta b$ as well as for varying normalized bending radii $R_L/a_0$ further simulations were carried out. Hereby, in analogy to the prior described approach, the results given in figure 5 were determined.

In order to determine the bending moment reduction for different tapering ratios with $\Delta a = \Delta b$ as well as for varying normalized bending radii $R_L/a_0$ further simulations were carried out. Hereby, in analogy to the prior described approach, the results given in figure 5 were determined. The normalized bending moment represents the ratio between the bending moment for bending with $(M_{b,R})$ and without $(M_{b,Con})$ simultaneous tapering by rolling. By the means of raising the tapering ratio for a constant normalized bending radius $R_L/a_0$ of 10 the normalized bending moment is reduced to a value of 0.3 for $\Delta a = \Delta b = 15 \%$. This reduction is attributable to the superposed compressive stresses that increase with the tapering ratio and therefore cause a further shift of the yield locus to lower bending stresses. Furthermore, for a constant tapering ratio of 10 % the normalized bending moment decreases with the tapering ratio.
normalized bending radius due to the concurrent increase of the proportion between bending stresses and superposed stresses.

![Normalized bending moment](image)

**Figure 5** Normalized bending moment for a constant normalized bending radius with varying tapering by rolling (left) and for constant tapering by rolling with varying normalized bending radius (right)

### 2.3. Process Spectrum and applications of profile bending with flexible rolling

Within the process spectrum of profile bending with flexible rolling, it is possible to differentiate between the degrees of freedom regarding the initial cross section, the bending radius, the wall strength as well as the final cross section (figure 6).

![Process spectrum](image)

**Figure 6** Process spectrum of profile bending with flexible rolling

The initial cross section may either be open or closed and symmetrical or asymmetrical. Through a variation of the rollers and the mandrel geometry as well as the adjustable positioning of those tools, the rolling gap geometry can be adapted to these different initial cross section variants. Here, certain limitations for profiles with undercuts are given. The wall thickness of the profile can be rolled out by a constant or variable value, with the possibility of setting wall thicknesses differing from one another in the y- and z- direction (see figure 2a). Similarly, with respect to the adjustment of the final cross section,
there is the possibility of a symmetrical and asymmetrical tapering. Since the bending radius is kinematically adjusted, any 2D bending contours can be produced with one tool set. By temporarily opening the nip and then angularly turning the profile, it is also possible to implement spatially curved structures consisting of several 2D bends. For 3D bending, tooling similar to the Nissin MOS bending principle [16] with a variable tool geometry needs to be deployed. However, in terms of the minimum bending radius and the product contour, limiting boundary conditions are given by the roll stands collision geometry.

The process spectrum of profile bending with flexible rolling enables the manufacturing of optimized lightweight components within the range from supporting structures for buildings over lattice frameworks for racing cars up to the ribs of custom made boats. In these areas of application, especially by bending high-strength steels with reduced springback and cross-sectional deformation as well as with increased formability, new design approaches are possible. Likewise, the production of crash components with intentionally introduced predetermined breaking points is conceivable.

3. Technological implementation – design of the roll stand

For designing a roll stand the occurring rolling force and torque during a forming operation are the most critical influences. Therefore, a numerical investigation with an adaptation of the above introduced FE-model was carried out to determine the maximum rolling force $F_R$ and torque $M_R$ required during different possible forming operations (figure 7 (a)). Here, a distinction was made between a pure edge reduction of the profile, which was examined above, and a pure thinning of the material thickness. In the first case, only the edge lengths of the profile are reduced without the profile being supported by a mandrel inside the rolling gap (see figure 3). In the second case, only the wall thickness of the profile is rolled out and the inner cross-section is kept constant by a supporting mandrel placed inside the rolling gap. For the pure thinning case the mandrel is modeled as a cuboid with an edge length of the front surface equal to $a_0 - 2 \cdot s_0$ and a depth of 300 mm. During thinning the mandrel front surface is aligned with the center axis of the rollers. Based on the results, thinning can be identified as the design critical forming operation, because more than twice as much force and torque is required for thinning by $\Delta s = 10\%$ than for tapering by $\Delta a = \Delta b = 10\%$. A reduction of the required rolling force and torque can be achieved by smaller roll diameter. Hereby the contact surface between roller and profile is decreased. Another possibility for reducing the required rolling torque is the reduction of the mandrel friction $\mu_M$.

![Figure 7](image_url) (a) Rolling force and torque during tapering of a square tube without mandrel support and rolling of the tube material thickness with mandrel support; (b) prototype design of the novel roll stand for flexible profile bending and its main components
For the developed novel roll stand (figure 7 (b)) a roller diameter $d_R$ of 230 mm was chosen in order to increase the assembly space within the flexible rollers for housing a retraction mechanism as well as for the roller drive train and for the roller bearings. The maximum rolling force was set to 800 kN and the maximum torque to 5 kNm. Thus, allowing for the tapering of DP800 square tubes with an edge length of up to 60 mm having a wall strength of up to 10 mm by 10 %. Within the roll stand the engagement of the tools is adjusted via four pressure-independent block cylinders whose stroke length is measured by linear potentiometers for a precise control of the nip dimensions. The rolling force applied by the cylinders is measured indirectly by pressure sensors that are part of the hydraulic system. To drive the rollers independently, four servomotors are connected with cardan shafts to the associated drive train. This also allows the use of different roller diameters within one tooling set, as the circumferential speed can be adapted individually.

4. Conclusion
Kinematic profile bending with flexible rolling has significant advantages compared to existing processes for the manufacturing of profiles with tailored dimensions and properties in small batch sizes. A numerical investigation of the potential for stress superposition during combined bending and rolling was carried out showing the ability to reduce the required bending moment by over 80 % ($R_L/a_0 = 25$; $\Delta a = \Delta b = 10 \%$), enabling the minimization of cross section deformation and springback. Additionally, the process spectrum of kinematic profile bending with flexible rolling was analyzed showing the potential to exploit diverse applications with a vast range of materials at low tooling cost. Subsequent a novel roll stand for the implementation of the process principle was developed proving the methods feasibility and the ability for forming high-strength steels like DP800.

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