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Closed-Loop Shape Control of a Roll-Bending Process

This paper describes a shape control system for a three roll bending process. The objective of this process is to impart a desired curvature to a workpiece at each point along the length of the part. Since final part shape is determined by the shape to which the workpiece is loaded and the amount of elastic springback, and since the latter is a strong function of the material properties of the workpiece, a truly closed-loop controller must account for these properties to insure consistent process performance. The scheme presented here accomplishes shape control by measuring both the loaded shape, the loading moment, and the effective beam rigidity of the material in real-time. As a result, a very accurate prediction of unloaded curvature can be made and an unloaded curvature control system can be designed. A series of experiments were performed to test this scheme by bending thin sheets of different materials to constant radii, specifying only the desired radius. The results indicate that the controller responds to the different material properties and produces accurate shapes regardless of variability in these properties.

Introduction

The forming of metal stock to continuously variable, simple curvature shapes is often accomplished by roll bending. Essentially an extension of three-point bending, the process involves imposing a bending moment on the material cross-section and at the same time rolling the material through the bending apparatus. A typical, but not unique, machine arrangement is shown in Fig. 1. In this configuration the material is driven by the center roll and this roll moves as shown between the outer rolls to provide variable bending moments. In this way each point along the length of the workpiece, as it traverses the rolls, can be given a specific maximum moment and a corresponding permanent deformation.

Roll bending is used in metalworking industries to form continuous curvature shapes from flat as well as contoured cross-section materials. It is especially useful in applications with critical dimensional tolerances, such as the rolling of circular sections for gas turbine components and pressure vessels sections. Because of difficulties in maintaining this dimensional accuracy, applications have been limited to constant curvature or circular shapes, with the major process control effort aimed at allowing for the elastic recovery or springback of the material as it is unloaded.

Dimensional control is difficult because different materials (or even identical materials from different lots) can exhibit significantly different springback characteristics. This is because springback is related to the yield stress and strain hardening nature of the material, both of which are extremely sensitive to the method of processing used. As a result, precision roll bending is a costly and time consuming operation that requires considerable effort in the upstream stages of material preparation to reduce these variations. The resulting manufacturing errors, when they do occur, give rise to expensive and slow manual reworking of the part to meet shape specifications.

Control of roll forming has been the subject of several investigations but none have yet addressed the central problem of on-line compensation for material springback. Foster [1] devised a control system that measured the shape of the part as it left the rolls, thereby allowing control of that shape. By locating the shape sensor outside of the bending region a significant transport delay in the controller resulted, which makes this system suitable for constant curvature shapes only. Others have concentrated on loaded shape control, that is the closed loop control of the part shape before it leaves the rolling mechanism [2], to study the dynamics of the process and through that determine the processing speed limitations of this device. This approach relies on a known and consistent springback of the material to achieve accurate shapes.

Open-loop or predictive methods have also been proposed [3], to be used, for example, with a bending machine that has numerical control of the roll positions. By knowing the workpiece material properties exactly (a difficult task to perform on a manufacturing shop floor) these models can be used to predict a time series of roll positions that will yield the desired shape. The sequential bending of ship frame members or beams has also been addressed from a control perspective [4]. In this work the springback of the material was determined by a series of loading and unloading cycles that eventually converged on the desired shape. Allison [5] employed a similar method in his research on control of brakeforming and achieved high accuracies without explicit knowledge of material properties.

In this paper, a control system for roll bending is presented that compensates for the springback in real-time by indirectly

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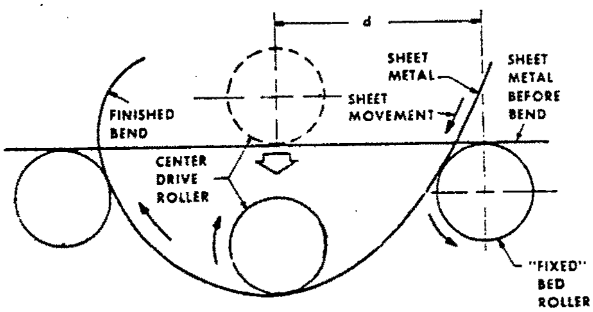

Fig. 1 A typical three-roll bending arrangement

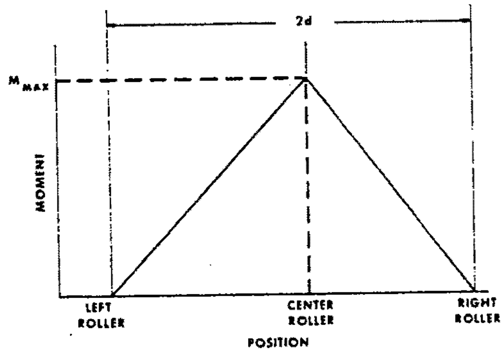

Fig. 2 Moment-arc length relationship

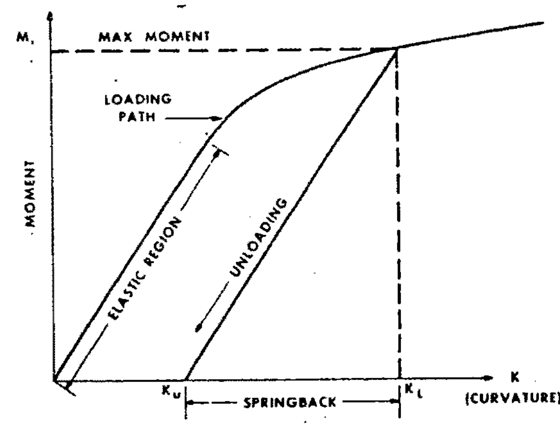

Fig. 3 Typical moment-curvature relationship

measuring the expected unloaded shape while still in the loaded state. The resulting process becomes a closed-loop unloaded part curvature or shape servo with the properties of command following (where the command is the desired sheet shape) and disturbance rejection that are characteristic of feedback systems. With this control scheme the process is rendered insensitive to material property variations and will always track the desired shape without operator intervention or experimentation. In essense, the first part is correct as are all subsequent parts, which means that set-up time is eliminated, single part production runs are feasible, and dimensional fidelity is greatly improved.

The Process

Although many manifestations of roll bending exist this work deals exclusively with the geometry shown in Fig. 1. As the part moves through the rolls, a point fixed in the workpiece sees a progressively increasing bending moment as it moves toward the center roll. Once past the center roll the moment decreases to zero when the part exists the machine. To insure that the location of the maximum bending moment is explicitly known, in this work the pinch roll is allowed to rotate about the center of the drive roll. In this way the pinch roll seeks the point of zero rate-of-change or maximum moment location. Although not strictly correct, it is assummed here that the moment in the part at any instant varies linearly with sheet arc-length as shown in Fig. 2. Notice in this figure that the center roll contact position on the workpiece is not necessarily mid-way between the outer rolls because of the floating roll arrangement.

Corresponding to each value of bending moment in the material is a local loaded curvature. This moment-curvature relationship is determined by the intrinsic properties of the material (specifically the modulus of elasticity, the yield stress and the strain hardening) and the material cross-section. It is analogous to the force-elongation characteristics of a uniaxial tensile specimen.

A typical moment curvature relationship is plotted in Fig. 3. Following the moment profile of a point fixed in the workpiece it can be seen that the curvature at that point will first increase linearly with increasing moment, corresponding to the elastic region. Then, if the yield moment is reached, it will follow the plastic portion of the curve incurring permanent deformation. At the maximum moment, which will occur at the center roll contact point, the maximum loaded curvature (K_l) for that point on the part will occur. As the moment begins to decrease, the material will unload elastically along a line nearly parallel to the initial elastic line, until the moment vanishes. The part now has, at that point, a permanent or unloaded curvature K_u.

The local springback ΔK is defined as the difference in the loaded and unloaded curvatures:

$$\Delta K = K_l - K_u \tag{1}$$

From the plot in Fig. 3 it can be seen that the springback is defined by:

$$\Delta K(s) = M(s)/(dM/dK)\,|_{\text{elastic}} \tag{2}$$

where $\Delta K(s)$ is the springback at a location s associated with a maximum moment $M(s)$ at that same point and the elastic slope is dM/dK, (the effective beam rigidity) which is a constant for a given material and geometry.

Several methods of determining springback are suggested by this relationship. In one, the moment-curvature ($M-K$) relationship is defined by *a priori* measurements such as stress-strain data or force displacement data taken during a static three point bending operation [6]. While this approach completely defines the material constitutive relations during the bending operation, it has the drawback mentioned above of relying on measurements made prior to the processing of the metal and on specimens other than those that will be used for part manufacture. Efforts at Batelle Labs [6] to control brakeforming by using such measurements and models have been quite successful provided a library of material properties is available and that significant lot variations do not occur.

The alternative to prediction of the constitutive relations is direct measurement of these properties on the part during the forming process. By using the roll forces and displacements, for example, it is possible to construct an approximate $M-K$ diagram that can then be used to calculate the springback for process control. This method has been successfully applied to control three-point bending [7] where the $M-K$ information is derived from in-process measurement of die forces and die displacements. A major source of error remains, however, in the estimation of sheet curvature since it is not directly measured.

A Closed-Loop Shape Control System

A third approach for real-time control in the face of springback exists and was pursued here.. It obviates knowledge of the complete $M-K$ relationship by substituting appropriate measurements. As a result the error producing

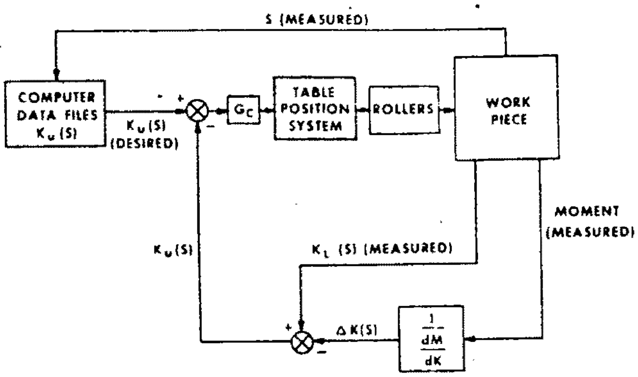

Fig. 4 Unloaded part curvature servo

and time consuming steps of constructing the entire $M-K$ curve and predicting curvature from assumed sheet shapes are eliminated, but additional instrumentation (in the form of force and curvature sensors) is required.

The objective of such a roll bending shape controller is to produce a part with a desired unloaded shape. This shape can be defined by the local curvature as a function of position along the part: $K(s)$, where s is the arc length along the part. What is needed, therefore, is an unloaded local curvature control system; but actual unloaded curvature cannot be measured until the sheet has left the machine thereby precluding any real-time compensation of local shape. However, if an accurate estimate of the local springback at the point of maximum moment (and maximum curvature) is known the unloaded curvature that will result can be found from rearranging equation (1):

$$K_u(s) = K_l(s) - \Delta K(s)$$

and substituting equation (2):

$$K_u(s) = K_l(s) - M(s)/(dM/dK) \quad (3)$$

It can now be seen that if $M(s)$ and $K_l(s)$ are measured and an accurate estimate of dM/dK is available, it is possible to have an on-line estimate of the unloaded curvature of the sheet at that point. With on-line estimation of $K_u(s)$ it is now possible to construct a block diagram of the proposed system (see Fig. 4). This system takes an arc-length varying curvature command and compares this with the estimated value of unloaded curvature to generate a curvature error. This error signal can then be used to change the center roll position to null out the shape error. The system will thus track the curvature commands and continuously respond to variations in material properties by measuring $M(s)$ and $K_l(s)$, which will reflect these changes.

System Dynamics

The previous development assumes a quasi-static situation for the bending process, and assumes that machine and workpiece dynamics effects are negligible. The extent to which rapid curvature changes and high rolling speeds are required will determine the significance of the dynamics of this process and the bandwidth requirements of the proposed shape control system. Although most roll bending is performed at low speeds relative to the machine and workpiece dynamics, there are instances where high speeds occur such as in steel can manufacture.

Considering the center roll position (Yp) as the system input it can be seen that the rate of change of this position is given by

$$dYp/dt = dYp/ds \cdot ds/dt$$

and that

$$dYp/ds \cong dK(s)/ds$$

Therefore the rate of change of the roll position is determined by the rate of change of curvature of the desired part dK/ds and the speed of rolling ds/dt, and this product will determine the frequency content of the input.

There are two elements in the roll bending process where mechanical dynamics must be considered: the roll positioning system and the workpiece itself. The roll system typically will be an electromechanical or electrohydraulic servo with second order dynamics typical of a high performance system. Such a system can be designed either by considering the sheet load as a disturbance or by approximating this load as purely elastic and incorporating it into the servo plant. In either case the roll positioning system block in Fig. 4 can be described by the closed-loop transfer function:

$$G_r(s) = \frac{K}{s^2 + 2\xi\omega_n s + \omega_n^2} = \frac{\text{roll position}}{\text{desired position}}$$

where ξ is chosen to be ≥ 1.0 and the bandwidth (ω_n) is a function of the size and power capabilities of the machine.

The second consideration is the dynamics of the workpiece itself as it traverses the rolls. Cook et al. [2] performed identification experiments on a three roll bending system and found that a transfer function relating roll position to sheet loaded curvature was well described by an underdamped second order system. Further they found that the position servo contribution to the transfer function was negligible. They then developed a state variable feedback control system to regulate sheet curvature and succeeded in obtaining good response.

The effect of workpiece dynamics on the present shape control scheme, however, is more involved and must be examined in greater detail. If each half of the workpiece (on either side of the center roll) is modelled as a cantelever beam with the clamp at the center roll contact point, a point load at the outer roll contact point, and a free section beyond the outer roll (see Fig. 5) the problem becomes one of lateral beam vibration. However, there are both elastic and plastic deflections involved and the beam length is continuously changing as the workpiece rolls through. A complete analysis of this situation is beyond the scope of this paper, but some order of magnitude estimates can be made.

Consider the beam in Fig. 5 to be in a loaded state. The variables of concern to the bending process are the moment and curvature of the beam at the clamped end. Vibration of the beam in this configuration will cause both of these to vary, however, for the elastic case moment and curvature are related by the constant beam rigidity. Therefore if the curvature is measured the effect of laterial vibration can be measured and compensated for. However, if plastic deformation occurs, measurement of both the loaded curvature and the associated moment at that point is required since their relationship is not longer known. The only reasonable moment measurement method is via the outside roll force, but this force will reflect both the static and dynamic roll forces and will no longer correctly reflect the moment at the center roll. This is because lateral acceleration of the beam outside the roll system will generate a dynamic moment at the outer roll contact point (see Fig. 5) whereas in the static case the workpiece moment vanishes at that point. Since the proposed control scheme assumes that outer roll forces adequately describe the center roll moment, the significance of the dynamic moment needs to be evaluated.

A detailed analysis of this system requires a model of a beam with a moving clamped end, a point load and a free section, but the actual significance of the vibration will be wholly dependent upon the amplitudes involved. On the other hand, if the modes of vibration of the workpiece can be identified and then avoided, either by band limiting the roll position system or the command to that system (as was done by Cook et al. [2]) the effect of workpiece dynamics can be minimized. Since the free section of the workpiece will clearly

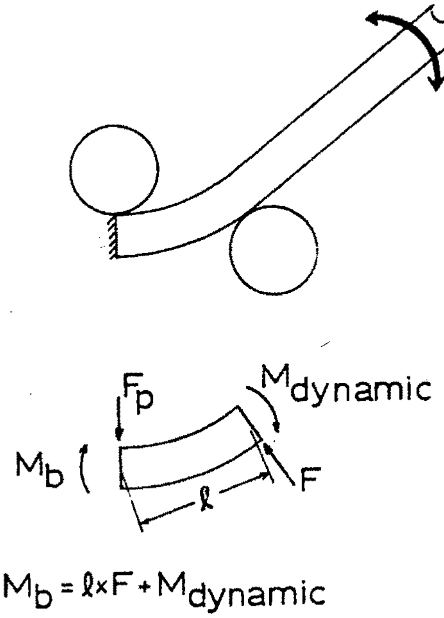

Fig. 5 Model for workpiece dynamics

dominate the workpiece dynamics, an estimate of the fundamental frequency of vibration will determine the required system band limit. If the outer roll contact point can be approximated as a clamp joint (which is justified by the short length of the loaded section relative to the free section), then the first mode frequency is given by

$$\omega_c = \frac{EI}{\rho A}\,\frac{1.875^2}{l}$$

ρ = material density
A = cross section area
l = beam length

For a workpiece of rectangular cross section with thickness t

$$\omega_c = \frac{t}{l^2}\,\frac{E}{\rho} \qquad (4)$$

(A typical result for 0.1 in. (2.5 mm) thick aluminum, 12 in. (0.28 m) long is 24 Hz.)

A viable control method would be to keep the frequencies of the curvature or roll position below ω_c, where the value is computed for the longest unsupported section of the workpiece. From equation (4) we can see that this limit will increase linearly with material thickness and inversely with the square of the free section length, thus long thin workpieces must be rolled at slower speeds than short or thick workpieces.

Implementation

The proposed shape control scheme requires a heavily instrumented forming machine. The maximum sheet moment, maximum loaded curvature, present sheet position and the location of the center roll-sheet contact point must all be measured. This has been accomplished with the following apparatus.

The roll machine and associated instrumentation used to test the proposed control scheme are shown in Figs. 6, 7, and 8. The center roll is mounted in the spindle of a vertical milling machine and remains stationary while the outer rolls are attached to the milling machine table. This table position is controlled by a computer commanded position servosystem. As mentioned above, one of the outer rolls is instrumented to measure the force vector acting on the roll as shown in the figure. The center roll has a pinch roll that is free to rotate about the drive roll axis. This angle of rotation (θ_t) is measured with a potentiometer to permit accurate calculation

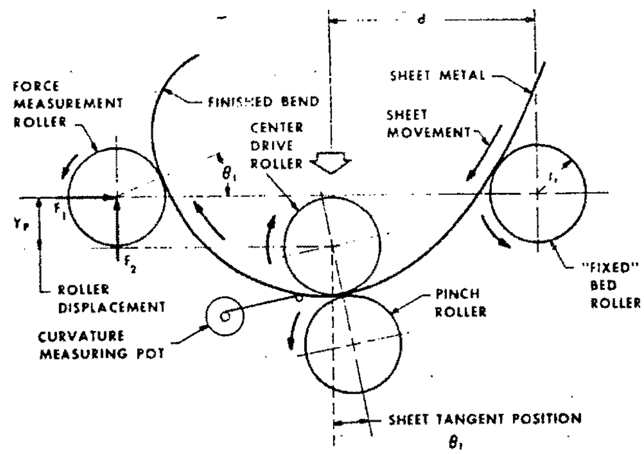

Fig. 6 Schematic of the experimental bending apparatus

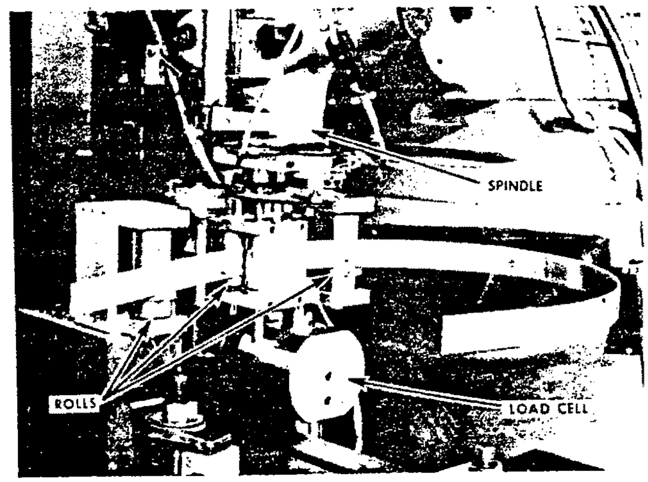

Fig. 7 The experimental bending apparatus

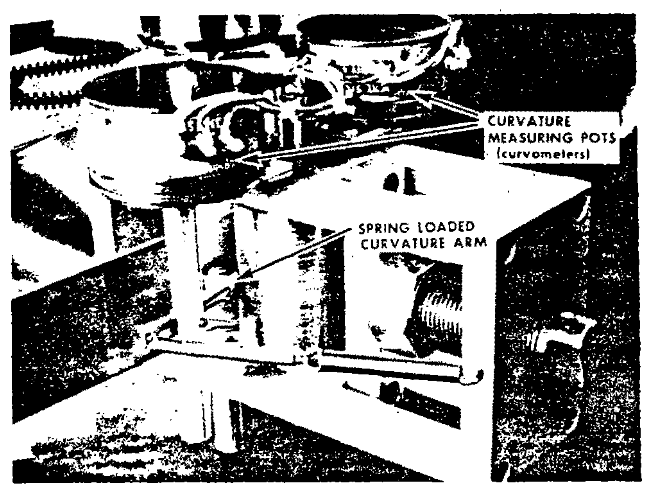

Fig. 8 Loaded curvature measurement device

of the sheet-roll contact position which is necessary for the sheet moment, $M(s)$, calculation.

The drive roll assembly (Fig. 8), which contains the pinch roll and rotates about the drive roll axis, is instrumented to measure the local sheet curvature. The measurement method involves using deviation of the sheet from the horizontal as it passes through the rolls as a local angle measure and an assumption of locally constant sheet curvature. To measure this elevation change a contact roller is attached to a potentiometer that provides an angular displacement θ_m, (see Fig. 8). To improve the signal to noise ratio of this

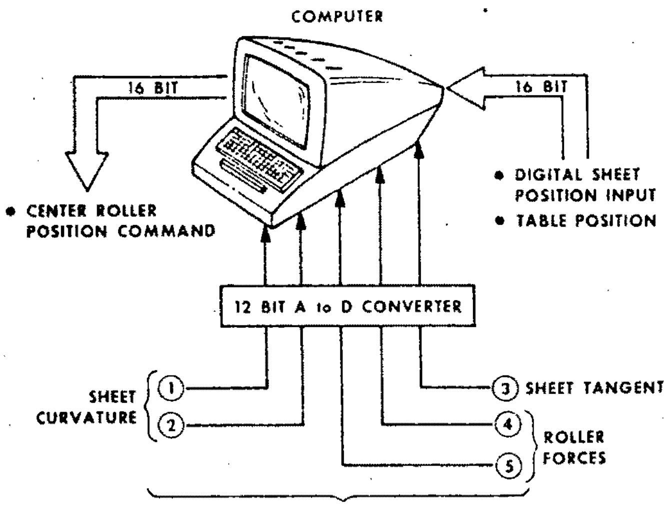

Fig. 9 Computer interfacing and data flow

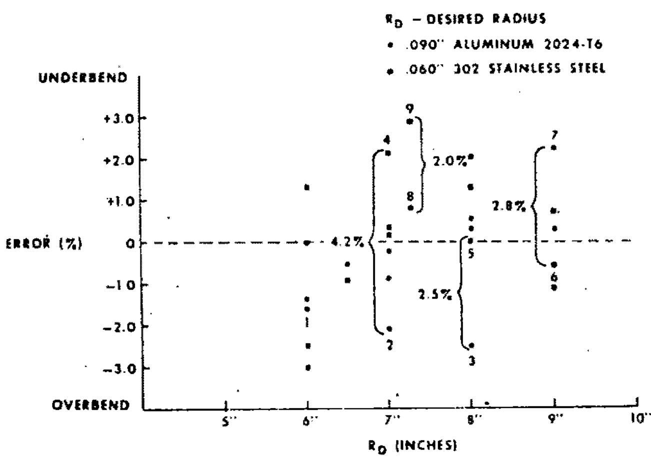

Fig. 10 Curvature error for constant radius bends

measurement and to compensate for nonequilibrium angular positions of the drive assembly during dynamic operation of the machine, this measurement is made on both sides of the center roll and the two values averaged.

(The distance of the contact point of the curvature measurement roll from the drive roll-sheet tangent point limits the spatial frequency resolution of the measurement and therefore the rate of change of curvature possible with this machine. There is clearly a tradeoff between keeping the measurement roller contact point close to the drive roll tangent point to more closely sense the curvature at that point, and moving this contact point out to maximize the rotation of the measuring potentiometer to improve the signal to noise ratio of the device.)

Although analytical expressions for curvature as a function of the potentiomenter outputs were derived, they are sufficiently involved computationally to preclude real-time implementation. Accordingly, to calibrate this "curvometer," a series of constant curvature rings were rolled and then placed (unloaded) into the center roll system. The angle, θ_m, from each of the curvature pots was then recorded for various radius sheets. For large ranges of curvatures this radius – θ_m relationship is nonlinear and a look-up table would be constructed, however, for the limited range of radii investigated here (5-10 in.) a linear relationship determined by a linear regression to the experimental data was found to be sufficient.

Finally, the sheet position within the three roll system is measured with a sequential encoder mounted on the drive roll shaft. This measurement is accumulated and used in conjunction with the drive roll-sheet tangent point measurement to determine what point on the sheet (denoted by the variable s) is presently at maximum curvature.

The maximum moment $M(s)$ is found from the force components F_x and F_y on the outer roll. These forces were measured with a two axis strain gauge dynomometer to which one of the outer rolls was mounted (see Fig. 7). The moment is given by:

$M = l_y F_x - l_x F_y$ where:
$l_x = d - r\sin\theta_t - r\cos\theta_f$
$l_y = Y_p + r\cos\theta_t + r\sin\theta_f$
$2d$ = outer roll separation
r = roll radius
Y_p = center roll position relative to outer rolls
θ_t = center roll tangent angle
θ_f = force vector angle as defined in Fig. 6.

The final element of the equation is the elastic slope of the moment curvature diagram dM/dK. This can be calculated by loading the sheet into the rolls and recording the moment and curvature at the center point as it is loaded. From a least squares fit to this data an accurate estimate of this slope can be found. It is also satisfactory to use predetermined values for the modulus and to calculate dM/dK from the elastic beam rigidity formula:

$$dM/dK = EI/(1-\nu^2)$$

where E is the modulus, I the area moment of inertia, and ν is Poisson's ratio. The divisor is necessary because wide sheet bending represents a plane strain condition to the elastic beam.

With these measurements it is now possible to calculate, for each increment of sheet position under the drive roll, the expected unloaded curvature $K_u(s)$ from equation (3). This permits implementation of the unloaded curvature scheme shown in Fig. 4.

The controller was implemented on a CROMEMCO Z2D microcomputer in FORTRAN. The computer received the measurement data as shown in Fig. 9 and commanded the roll position Y_p. In essence this program is a closed-loop unloaded curvature controller that minimizes the curvature error according to the current desired curvature command $K_u(s)_{des}$. To assure that zero steady state error in curvature would be achieved, the controller itself is a pure integrator; i.e., $G_c(s)$ in Fig. 4. $= K_c/s$.

The response of the resulting curvature servo was determined by assuming that the dynamics of the roll position servo were negligible (since it has a 60 Hz bandwidth) and that the load presented by the sheet was very small. Considering then only the controller ($G_c(s)$ in Fig. 4) and a first order low-pass filter that was required because of noise on the curvometer signals, the resulting closed-loop transfer function is:

$$T_c(s) = \frac{K_c/s}{1 + \dfrac{K_c}{s(\tau_f s + 1)}} = \frac{K_c(s + 1/\tau_f)}{s^2 + 1/\tau_f s + K_c/\tau_f}$$

K_c = controller gain and τ_f = time constant of the feedback filter. To achieve fixed damping ratio $\xi = 1.0$, K_c was chosen such that

$$\xi = \frac{1/\tau_f}{2\sqrt{K_c/\tau_f}} \geq 1 \text{ or } K_c \leq 4/\tau_f$$

However, the zero at $1/\tau_f$ also contributed to step response overshoot and K_c was further reduced to give monotonic step response. With $1/\tau_f \sim 1$ Hz, the resulting system was severely limited in bandwidth and could only be used for quasi-static control tests. Instrumentation that would not require such heavy filtering would, of course, directly improve the controller bandwidth as would alternative choices for $G_c(s)$.

Experiments

A series of constant curvature bending experiments were performed to evaluate both the steady-state accuracy of the curvature control system and to test the ability of the system to compensate for different material properties and geometries. Two materials were used (0.1 in. (2.5 mm) thick aluminum 2024-T6 and 0.060 in. (1.5 mm) thick 302 stainless steel) and radii of 6 to 9 in. (0.15 m to 0.23 m) were commanded. Since these materials have quite different elastic moduli and yield stresses, and because the roll separation distance (8 in.) (0.2 m) is wide relative to the material thickness, these tests represent a severe test of capabilities of the control system.

The procedure involved using predetermined values of the elastic $M-K$ slope (as described above) and specifying a desired part curvature (or radius). The test sheet was then loaded into the machine and the control program started. The sheet was rolled through at a rate of approximately 60 inches per minute (1.5 m/min) until a circle of at least one-half full circumference was rolled. This circle was measured and the radius compared to the desired radius. The test was then repeated for a different material.

The curvature error, plotted in Fig. 10 shows that no systematic error exists in the curvature either as a function of the desired curvature or as a function of the material being formed. This error, when expressed as a percentage of the desired curvature, is limited to a band of ± 3 percent. (It should be noted that the resolution involved in off-line measuring of the actural part radius was on the order of 1 – 2 percent of the radius as was the curvometer resolution.)

Conclusions

A real-time, closed-loop shape control system has been demonstrated for roll bending of continuous contours. The essential control properties of shape-command following and insensitivity to plant parameter (material property) changes have also been shown. The initial experiments have been limited to shapes of constant curvature and do not address the problems of machine and workpiece dynamics that will arise as arbitrary contours are formed at high speeds. However, the algorithm and instrumentation scheme described is fully capable of producing such contours with only a limitation on the processing speed of such shapes.

Enhancements of performance will be realized through improved instrumentation, implementation of a high tracking accuracy controller algorithm that continuously adapts to changing material dynamics and possibly through the addition of a post-processing shape measurement to act as a correction signal for the unloaded curvature estimate.

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