INSTALLATION AND PERFORMANCE OF CLASSICAL WEB GUIDES

by

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ABSTRACT

This paper depicts proper installations of conventional web guides and examines the theory of web mechanics and the theory of automatic control as foundations for the practice of web guiding. Conditions which would damage the web are quantified. The adverse effects of interaction of spans and avoidance of such interaction are discussed.

Limitations to accuracy of conventional web guides are discussed. Unusual but proven variations for satisfying special needs are presented.

NOMENCLATURE

A, B	first and second sensors
E	modulus of elasticity (Young's modulus)
e	error
G	modulus of elasticity in shear
K	a parameter equal to $\sqrt{12\sigma_{avg} / E / W}$
K_0	open-loop gain of the control system
K_p	gain of a positioner
L	length of a free span of web
L	as a subscript, the condition at the downstream roller
N	nonlinear, non-frequency-dependent dynamics for describing function analysis
O	as a subscript, the condition at the upstream roller
s	Laplace transform operator (equal to $j\omega$)
V	velocity of the web
W	width of web
x	distance downweb from roller
y	lateral web deflection from its original position
$\in$	strain = $\Delta L/L$
θ	slope of the elastic curve (same as y′)
θ_r	roller angle
σ	stress
τ	time constant of a web span – seconds (= L/V)
τ_e	electrical time constant – seconds
τ_m	mechanical time constant – seconds
τ_T	total time constant over several spans – seconds
ω	frequency – radians per second

INTRODUCTION

Neither the mechanics of web behavior nor the control systems for positioning the web was new technology when web guiding systems evolved. However, lateral behavior of a web is determined by boundary conditions which are not necessarily obvious, instead of by known loads and often obvious boundary conditions for structural beams. Further, most of the analysis of beams with axial loads has considered compressive stresses, as in a column, instead of tensile stresses, as in a web.

Web guide control systems are handicapped by the difficulty of precise sensing of a moving web and by the impracticality of sensing the angle of the web, as well as its lateral velocity and acceleration, parameters which are readily available for positioning of machine tools and other rigid bodies. Precise sensing is particularly difficult when the need arises for a wide-gap, wide-band, linear sensor.

The incentive for this paper is the publication of several technical papers and books which demonstrate ignorance of simple rules of installation and of the performance of web guides. Some publications have repudiated, without verification, established principles of the lateral behavior of a web.

This paper presents an overview of web guiding and usable results, descriptively and graphically. Space allows little technical depth or detail.

LATERAL BEHAVIOR OF A WEB

Disturbances

Lateral conditions at the downstream roller of one span are transported to the next span, becoming a disturbance to the second span. Conditions which may be transported to the next span include lateral position, angle (slope), curvature (bending stress), as well as shear stress. The complicated combination of transported conditions precludes perfect downstream control of a web even if it was initially straight.

The behavior of two or more spans which interact is much more complicated than the transport of conditions across a roller which has enough friction to prevent any effect on the upstream span by conditions in the downstream span. Interaction of spans can render a web guide detrimental instead of beneficial, and in combination with a misaligned roller can cause large lateral errors and erratic lateral behavior whether or not a guide is present (primarily because of changes in the coefficient of friction in the absence of feedback control by a guide). A paper on interaction of two spans because of a misaligned roller or a steering guide, and the resulting self-induced disturbances, is planned by John Shelton for IWEB VIII.

If a web is not straight when flattened under zero tension, disturbances arise from this out-of-straightness, although tension reduces the out-of-straightness. If a splice is present within a wound roll, the splice is usually the primary source of out-of-straightness of the web during processing. The spliced ends may be offset laterally, an error seen as a step disturbance by a web guide, or the spliced ends may meet at an angle, a condition little better than the lateral offset. A web which is formed discontinuously, such as strip metal which is formed by progressive rolling of an ingot, is likely to have imperfections such as changing curvature or a variation of width near its ends, flaws which a web guide cannot correct. If a mill roll at the output of a process is intended to contain the same length as the unwinding mill roll, the splice and some web before and after are usually discarded or reprocessed.

The telescoping of an unwinding roll or deliberate oscillation of the web or the winding roll at the previous process ("scatter winding") for distribution of gage bands is a disturbance to an unwind guide or the first intermediate guide in the absence of an unwind guide. The imperfection of correction by the first guide then becomes a disturbance to the next guide. The disturbances to downstream guides are greater if the scatter winding was accomplished by an oscillating slitter trimming each edge by a varying amount than by an oscillating guide point positioning an initially straight web at the previous winder; however, partial permanent set in the oscillated shape may occur with the latter method of scatter winding.

Telescoping of a winding roll is neither a flaw of a rewind guide nor a disturbance to the guide, but is often mistaken for poor accuracy of a guide.

Camber, or initial curvature of an untensioned but flattened web, has defied efforts at analysis for more than thirty years. Absolutely constant camber would not require a web guide for lateral correction in a stable process line, but camber generally changes over time periods of various durations, creating one of the primary reasons for needing web guides. If the effect of constant camber on the lateral position of a web were quantified, the needed locations of web guides would be much more predictable. Shelton [1] presented a theory which seemed reasonable, but which was not well verified by experiments at the time, and which was further discounted in experiments by Swanson [2]. Benson [3] reached the conclusion from analysis which seems to be obvious for a web with constant camber but with enough tension to pull it taut across its width – that there is then no lateral error if the rollers are cylindrical and parallel; however, cambered webs have usually been observed to track toward the long side. If roller mountings are flexible, a cambered web may track toward its short side because of tilting of rollers from the nonuniform distribution of tension.

Boundary Conditions for Lateral Web Behavior

Many analysts of the behavior of structures have assumed boundary conditions, analyzed the problem, then tried to duplicate desirable assumed conditions in practice. A moving web, however, creates its own boundary conditions with no regard for the desires of the analyst, and these boundary conditions may be exceedingly difficult to formulate and verify. An example of such difficulty is the effort to understand the behavior of a web with constant and uniform camber, as discussed above. Success in establishment of the steady state downstream boundary condition of zero moment for an initially straight web with a misaligned downstream roller, and subsequently the dynamic condition of lateral acceleration proportional to the moment, was achieved by the author in 1968 [4]. It is noteworthy that this steady-state condition was discovered, neither as a bolt from the blue nor by profound mathematical derivations, but by careful tests of the lateral force at the misaligned downstream roller and subsequent calculations of the residual moment. The calculations clearly showed that the steady-state downstream moment was zero, a condition then recognized to be logical. The equations for dynamic behavior (the basic differential equations and the boundary conditions) were then established by a combination of mathematical logic, intuitive reasoning, and testing, with the latter guiding final formulation.

Damage to a Web by a Guide

The primary purpose of many web guides is the prevention of damage to the web caused by running off rollers and hitting stationary components. However, a web may be damaged and production interrupted by a web guide which is unsuitable because of concept, design, or installation. The primary types of damage by a guide are breakage, wrinkling, and stretching of an edge. Scratching caused by lateral slippage could occur, but is little more likely to occur at a guide than at other rollers.

Laterally shifting unwinders and winders do not alter the parallelism between rollers and the roll of material; therefore, damage to a web by a terminal guide is unlikely to occur, and will not be considered here.

A properly installed displacement guide maintains parallelism between all rollers as viewed into each web span, with the only nonparallelism resulting in twisting of the entering and exiting spans. The increased tensile stress at the edges and decreased stress at the center are usually negligible, and will not be considered further in this paper.

Slack Edge. A steering guide bends the web across its width, so that the bending stress is zero at the downstream roller and a maximum at the upstream roller. If the web was initially straight and flat, the condition causing a borderline slack edge causes the stress at the other edge to be twice that present in a tensioned but unbent web. Although many steering guides have appeared to be successful while imposing slackness across a large percentage of the width at the upstream roller, wrinkling sometimes accompanies the slackness, and the large increase in stress at the other edge may cause yielding of the taut edge which is not obvious. Nominal slackness of one edge is therefore a reasonable design criterion for a steering guide.

The conditions which cause borderline slackness of an edge are shown in Figure 1 for an isotropic web. Please note the curve labeled "0.1 (pure bending)", which is the result of neglecting shear deflection, and which is erroneous by a factor of about 30 compared to the correct dashed curve labeled "0.1". Also note that the permissible misalignment is a minimum at a value of L/W equal to approximately 0.7. Although this ratio is too low for success of a common steering guide, it is interesting that alignment of fixed rollers is most critical if L/W is about 0.7 if the web is isotropic. If the shear modulus is very low in

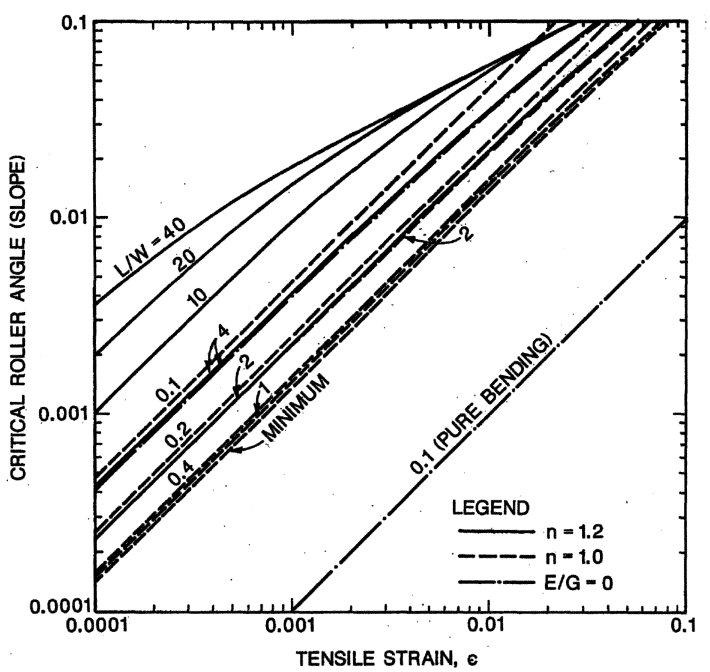

Figure 1 – Roller Misalignment Causing Slack Edge (Isotropic Web)

comparison to the tensile modulus, the worst value of L/W for avoidance of a slack edge is higher.

Wrinkling. Pinpointing the conditions which would cause wrinkling of a web is far more difficult than predicting a slack edge. One reason is that the lateral stress being transported into a span is unknown. The lateral stress may be tensile, particularly if the roller at the upstream end of the span is an effective spreader roller. For the rare case of the upstream end being the exit of a tenter, lateral tensile stress would be present, but then viscoelasticity, temperature changes, and other complications probably would preclude satisfactory modeling.

Another barrier to accurate prediction of wrinkling is the possibility that the laterally strained web may slide outward as it contacts a roller, as explained by Shelton [5] and Good, et al. [6]. Many webs suffer severe wrinkling at low speeds, but flatten out and run well at operating speeds, where air entrainment reduces the web-to-roller friction.

The basis for prediction of wrinkling as caused by a misaligned roller is that shear stresses (from bending of the web as a cantilever beam) lead to lateral compressive stresses, as explained by Mohr's circle. Such analysis led to Figure 2, which was well supported by tests of a specific web at different values of tension, as reported by Good, et al. [6]. The data points in Figure 2 were obtained at a low speed of the web, and

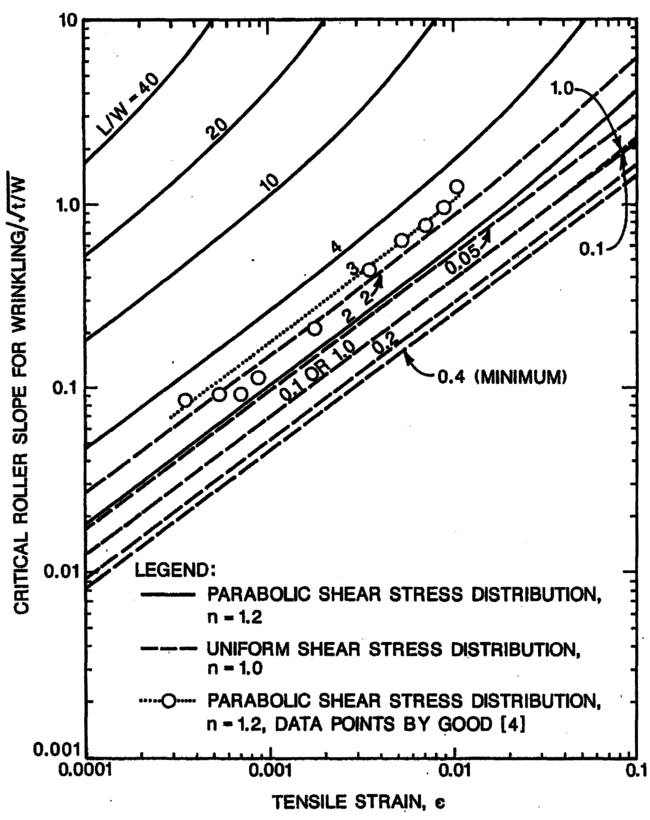

Figure 2 – Permissible Misalignment of Rollers for Avoidance of Wrinkling of Isotropic Web (High Web-Roller Friction)

with a roller covering which generated an extremely high coefficient of friction between itself and the test web.

The slackness of an edge at the upstream end of a span with a misaligned downstream roller is precisely predicted, as graphed in Figure 1, as long as friction on the upstream roller is sufficient to prevent the transfer of some of the moment to the next upstream span. The behavior of the isolated span is then usually insensitive to changes in friction at the misaligned roller and to minor changes in Poisson's ratio and other parameters. Wrinkling as shown in Figure 2, however, is extremely sensitive to friction between the web and the downstream roller. Low friction may allow potential wrinkles in the form of corrugations to slide outward as the web makes contact with the roller, as theorized by Shelton [5] and Good, et al. [6]. This avoidance of wrinkling because of low friction has been called "Regime 2" by Good.

Figure 2 is based on the assumption that the web is transported into the span with no lateral stress, and that no outward slippage occurs on the downstream roller. Furthermore, Figure 2 does not actually predict wrinkling, but only lateral buckling (troughing) of a web which was transported into the span under zero lateral stress. Basing the permissible misalignment for avoidance of wrinkling on this graph is therefore usually conservative, sometimes very conservative; otherwise, many steering guides which have proven to be satisfactory would have wrinkled the web. It is conceivable, however, that residual lateral compressive stresses transported from upstream might cause troughing or even wrinkling at lower valves of misalignment than the prediction of Figure 2. The relationship between troughing and development of a wrinkle (crease) has not been quantified.

Figure 2 shows that a span is most susceptible to wrinkling if L/W is 0.4 (for an isotropic web), contrasted to 0.7 for the worst value of L/W for misalignment as a cause of a slack edge. Susceptibility to wrinkling is nearly the same for L/W of 0.1 as for 1.0.

Proper Installation of Intermediate Guides for Prevention of Unwanted Steering

The proper installation of web guides has been publicized for several decades, yet a recent technical paper assumed that a steering guide is installed with the plane of guide roller motion parallel to the entering span instead of (properly) perpendicular to the exiting span, as shown in Figure 3.

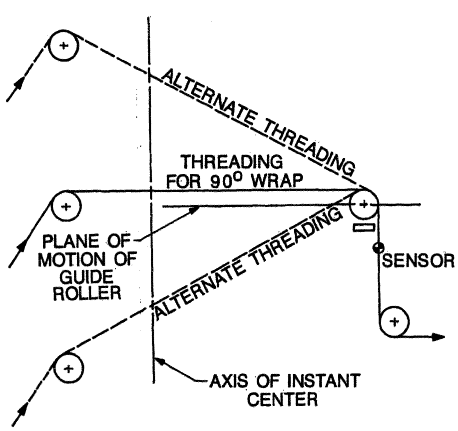

Figure 3 – Configuration of Steering Guide with Remote Instant Center

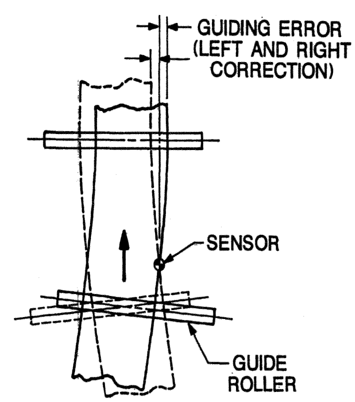

Figure 4 – Residual Error From Misalignment Between Guide Roller and Downstream Roller

Figure 4 shows the exiting span of a guide which does not have its plane of motion perpendicular to the exiting span, resulting in the reintroduction of error by the curving web downstream from the sensor; also, the usually short exiting span is unnecessarily stressed in bending, perhaps leading to web breakage or permanent stretching of the edge. On the other hand, a proper installation merely twists the exiting span, resulting in only a small increase in stress at the edges.

A steering guide corrects the error by laterally bending the web in the entering span. The bending stresses are easily calculated up to the point of slackness of an edge (with twice the average stress then on the other edge), and a long entering span is usually required. If the guide roller of a properly installed steering guide is wrapped by more or less than 90 degrees, harmless twisting of the entering span occurs, in addition to the bending. A steering guide with an overall wrap angle approaching zero (nip rollers or two or three wrapped rollers) or approaching 180 degrees is improper.

Sensors for web guides are often located improperly, sometimes because of the misconception that the sensor can simply be located where the greatest accuracy is needed. The overriding principle for location of the sensor is that feedback must be instantaneous and not greatly diminished from the magnitude of the dynamic positioning of the web by the guide. For a classical intermediate or unwind guide, this principle dictates location of the sensor in the exiting span, less than halfway to the next roller. A sensor farther downstream, unless the control system is greatly modified for utilization of two sensors as discussed on pages 12-14, results in a slow oscillation of the web guide, with the amplitude of oscillation decreasing as the speed of the web increases.

A displacement guide usually has short entering and exiting spans, thereby requiring that these spans be perpendicular to the plane of motion of the guide. This rule of perpendicularity should be observed for all installations, as illustrated for several variations in Figure 5. Illustrated in Figure 6 is a guide with a long entering span which suffers negative steering to the extent of canceling the correcting ability of the displacement action, resulting in a worthless guide or even one which amplifies an error as it bottoms out in trying to correct the error. This complete loss of guiding ability occurs when the projection of the entering span into the plane of motion of the guide is between L for a web with zero shear modulus and (3/2)L for the other extreme of an isotropic web with L several times greater than W, and with KL not greater than approximately unity.

There is no upper limit to the span between the two parallel rollers of a displacement

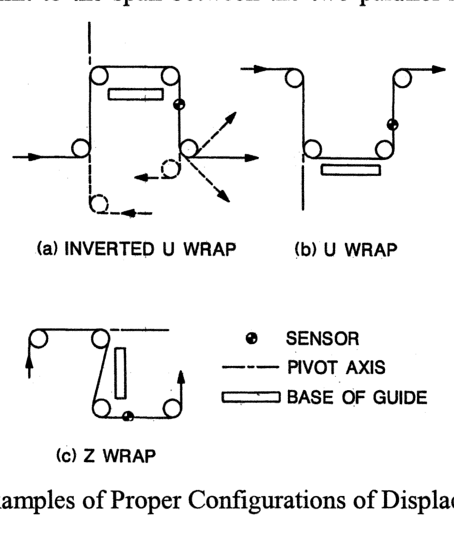

Figure 5 – Examples of Proper Configurations of Displacement Guides

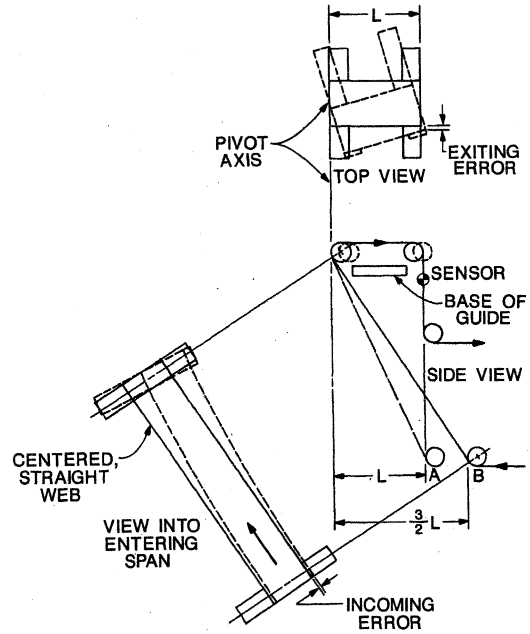

Figure 6 – Extreme Example of Improper Installation of Displacement Guide

guide which shift in unison about the center of the entering span. The guiding span could be, for example, a complete air flotation oven if the entering and exiting spans can be perpendicular to the plane of the oven. The rollers at the entrance and exit of the oven do not need to be on a single frame, but they could be synchronized with position sensors and actuators. Any number of parallel rollers can be mounted on the shifting frame between the entering and exiting spans of a displacement guide, and these rollers can be spreaders, drive rollers, or can even be a web process. Note, however, that the guiding action occurs in the exiting span, not on the shifting frame.

Modeling of displacement guides has shown that the "weave regeneration" studied by Sievers [7] can be greatly reduced by designing the dimension across the guide to be very long (several times the width of the web) and the exiting span (and the entering span if desired) to be very short (a fraction of the width of the web). The exact dynamic performance of such a guide is difficult to generalize, however, because of dependence on the dynamics of the controller/actuator, the lateral dynamics of the web, and the frequency of the disturbance.

Shifting Unwinders and Winders

Lateral control at unwinders and winders, "sidelay," was relatively successful before intermediate guiding was understood well enough to be routinely successful. Primary issues are location and mounting of the sensor(s), lateral stiffness of the shifting stand, and the relatively great power required to shift large stands. Steady-state behavior of the web is not an issue, because the web is transported over rollers which should be parallel and cylindrical. The fault of a conical winding roll is unrelated to the shifting of the stand. Dynamic faults, such as momentary wrinkling or slackness of an edge, are seldom encountered because of the usual smoothness of lateral disturbances.

It should be noted that lateral control of a winder is not guiding, but is chasing the wandering web; hence, the mounting and location of the sensor(s) are entirely different from those for a web guide. Terminal guides are discussed by Hopcus [8], and therefore will not be further discussed here except for the suggestion that an intermediate guide, particularly a displacement guide, could often be designed into a new process line near the unwinder or winder at lower cost yet with higher performance than that of a terminal guide.

WEB GUIDE CONTROL SYSTEMS

Dynamics of a Web Guide Controller. The dynamic capability of common web guide controllers has been known for more than forty years. The usual unity crossover of the open-loop frequency is approximately 5 cycles per second, and the maximum speed of correction rarely needs to be higher than 50 mm per second. Small high-performance electromechanical systems (with inner loops of current and velocity) may have frequency response three to five times higher, while cheap systems may be very poor, especially if the rotary-to-linear screw is subject to wear and consequent backlash.

Many process control engineers are perplexed by the absence of a PID compensator in the controller of a web guide, as a PID (usually using only the PI features) compensator is often needed for accurate control of liquid level, temperature, and other process parameters. However, the proportional-integral feature is inherent in the servo valve of a hydraulic web guide and in the motor of an electromechanical web guide. The output velocity is proportional to the error; hence, the output position for the open loop is the time integral of the error.

Figure 7 shows 1963 frequency-response tests of two pneumo-hydraulic (air-jet sensing) systems. Such a system had zero dead band unless it was overloaded, and could correct a ramp disturbance with approximately 0.03 units of residual error per unit per second of velocity of the disturbance, by virtue of its type 1 characteristic. Higher performance has rarely been needed.

In contrast to the "typical" model of Figure 7 along with the graphs of the actual tests, a recent paper modeled the dynamics of two low-performance web guiding systems as (1) an on-off system with dead band (either stationary in its dead band or traveling in either direction at 5 mm per second), and (2) a type 1 (integrating) system, but with the very low gain of 3.0 units per second per unit of error, approximately one-tenth the gain of the "typical" guide. The second model is a good representation of the "typical" guide with its gain reduced by a factor of 10, making the effect of the time constant(s) of the controller negligible, but such low performance would not be generally acceptable. In fact, the proportional band of many sensors is too narrow for a guide with this low gain to achieve maximum speed. The first model, with a velocity of 5 mm per second, is too slow for many

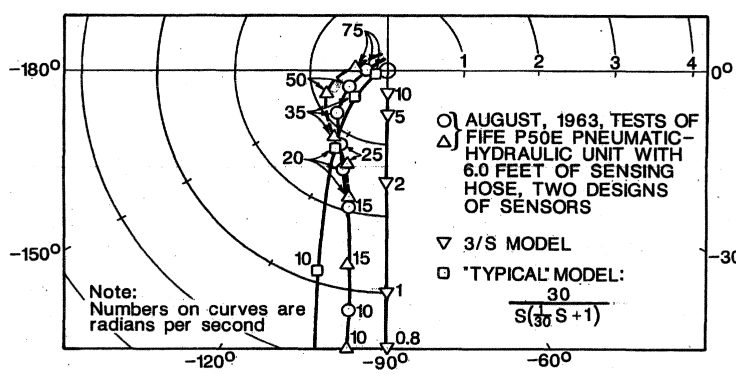

Figure 7 – Open-Loop Tests and Models

web guiding needs, and the necessary dead band would leave small errors unreduced.

The first of the above models represents an inexpensive web guide, such as a hydraulic system with a solenoid valve for left–right control, and a sensor consisting of two light beams and photocells with the dead band determined by the lateral spacing of the light beams and photocells. However, a large portion of the cost of a guide installation is commonly the mechanical components, such as structure, rollers, linear bearings, and the actuator, so that no major savings would result from the inexpensive controller.

Dead Band and Backlash. Some engineers have failed to recognize the difference between dead band and backlash, but the difference is monumental in a control system – dead band is a stabilizing influence, while backlash, especially if it is inertia-controlled (coasting output) is a serious destabilizing influence.

Physical concepts of dead band and backlash will now be discussed. The input/output characteristic for dead band as shown in Figure 8(a) has no hysteresis; that is, a given decreasing input causes the same output as a numerically equal increasing input. In contrast, the characteristic for backlash as shown in Figure 9(b) shows a different path for a decreasing input from that for an increasing input, resulting in a phase angle between the sinusoidal input and the output. If friction is insufficient to prevent coasting of the output member, an oscillation is almost certain, and this oscillation causes impact between the mechanical parts, increasing the backlash.

Describing functions for dead band and backlash were derived in the classical period of the discipline of automatic controls. Application of the describing function method entails finding the intersection of the locus of frequency-dependent terms (such as the loci of Figure 7) and the negative reciprocal of the nonlinear term, or –1/N. A plot of 1/N for dead band is shown in Figure 8(b), and a polar plot of –1/N is shown in Figure 9(c). A comparison of Figures 7 and 9(c) reveals that the locus of the dynamics of a usual web guide inevitably crosses the plot of –1/N of inertia-controlled backlash, resulting in a

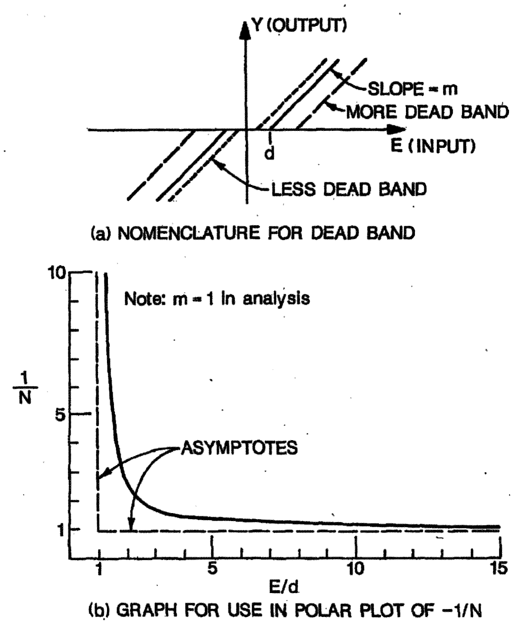

Figure 8 – Describing Function For Dead Band

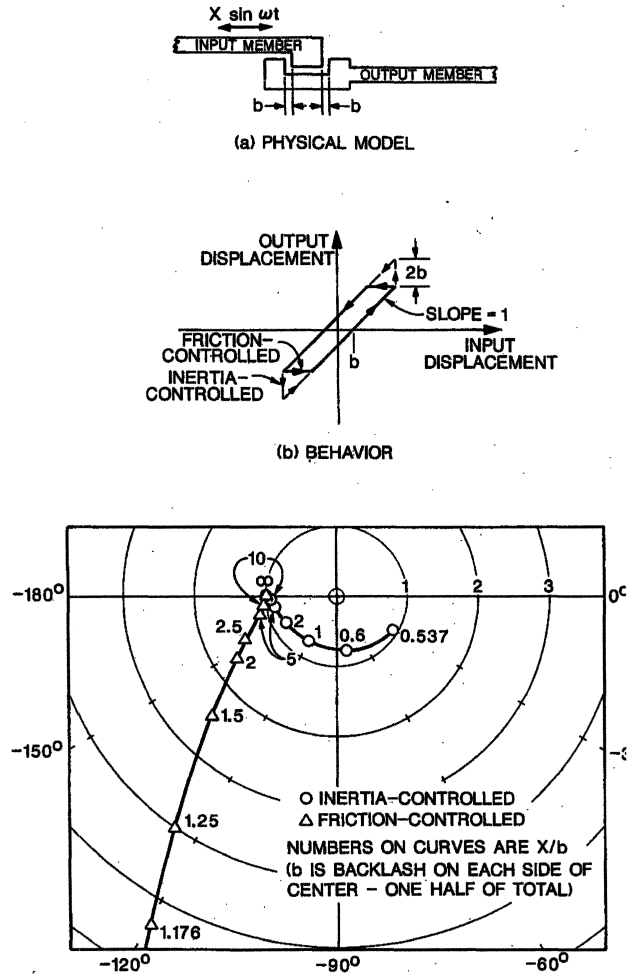

Figure 9 – Describing Function For Backlash

continuous limit-cycle oscillation with a fundamental frequency and amplitude as indicated by the values of frequency and amplitude at the point of the intersection.

Primary sources of backlash in a web guide are loose connections in clevis mounts of a hydraulic cylinder or an electromechanical actuator, and clearance between the screw and the nut of an electromechanical actuator. The actuator should utilize a preloaded ball nut/screw, as a plain screw and nut for conversion from rotary to linear motion is almost certain to cause trouble as wear occurs.

The goal should always be complete elimination of backlash. For hydraulic systems with clevis-mounted cylinders, backlash can be eliminated by using two cylinders, each pushing in opposite directions on the load, instead of one push-pull cylinder. The rod-end ports are then plumbed together and connected to the oil reservoir.

Generally, clevis or trunnion mounting of single actuators should be avoided. Intermediate guides can be located by double-slider or multiple-slider mechanisms, so that a linear actuator can be directly connected to the guide mechanism at a point which moves in a straight line.

In Figure 8, the output which is affected by the dead band is usually velocity. An example is a servo valve in which the spool is fitted for insignificant leakage when centered and the corners of the spool and ports are square, but the lands on the spool are wider than the control ports, so that a finite travel of the spool is required to result in a significant flow rate to and from the hydraulic cylinder. A second example of a hydraulic servo valve is one in which there is leakage across the control ports (from pressure to exhaust) and the load has significant breakaway friction, so that the spool must travel a significant distance to generate the pressure difference across the control ports as required to overcome the friction. The latter case is similar to the dead band in a electromechanical system with significant breakaway friction, where the current to the actuator motor must build to a certain level to overcome the friction. Such dead band in an electromechanical system can be reduced or virtually eliminated with inner feedback loops of current and velocity, as discussed on pages 14-16.

Guide Point Adjustment by Remote Downstream Sensor

In a small percentage of web guiding applications, space limitations preclude the installation of a web guide where the best accuracy is needed. In some such cases, a complete properly configured web guide (including the sensor) can be installed several short spans upstream, and a sensor at the location of the desired precision can then make small changes to the set-point of the guide to compensate for errors between the primary guide sensor and the secondary remote sensor. As discussed previously, mere location of the usual single sensor beyond the next downstream roller usually results in an oscillation which increases in magnitude as the web velocity decreases; further, the web is often broken as it stops in such an installation unless the guide is locked at a certain minimum web velocity.

If the cause(s) of the accumulation of error between the two sensors were known, an algorithm might be formulated from measurement(s) of the cause(s), then adjustment of the set point of the guide sensor would compensate for the error, and the remote sensor would not be needed. This would be a feedforward scheme which would not compromise stability of the system; however, neither the measurement of the secondary variables (such as camber in the web) nor the application of this measurement to a calculation of the change in set-point of the sensor is likely to be precise enough to be beneficial.

The recommended system utilizes an outer loop of slow integration for adjustment of the set point of the guide sensor. Instability (manifest as a slow, large-amplitude oscillation) limits the rate of integration. The integrator must be shut off when the web is stopped, and the rate of integration must be changed with major changes in web velocity.

The guide-point adjustment system is shown pictorially in Figure 10(a) and in block-diagram form in (b) and (c). For an edge guide (contrasted to a center guide), both sensors must be moved simultaneously or both sensor set points must be moved the same amount if the web width or the desired lateral web position is changed. The guide is shown as an unwind positioner, but could be an intermediate guide. Justification of the proposed consideration of the dynamics of the web between the sensors as a pure transport lag (equal to the time of travel of the web between the sensors) is shown in Figure 11. Although this graph is based on equal-length short spans [9], a 1986 derivation pointed to the same conclusion (approximation of the dynamics as a pure transport lag); furthermore, this approximation of the dynamics also applies to spans of unequal lengths.

One approach for control of the web at the downstream sensor while avoiding oscillation would be to periodically correct the error as measured by the downstream sensor, but avoiding a subsequent correction until the transport time between Sensors A and B has elapsed – the "Smith Predictor" approach. The effective sampling interval would

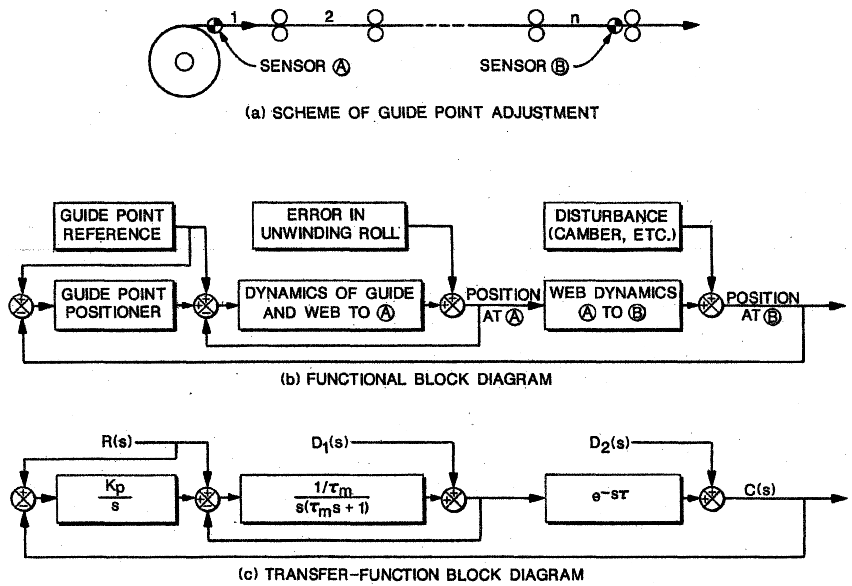

Figure 10 – Guide Point Adjustment by Downstream Sensor

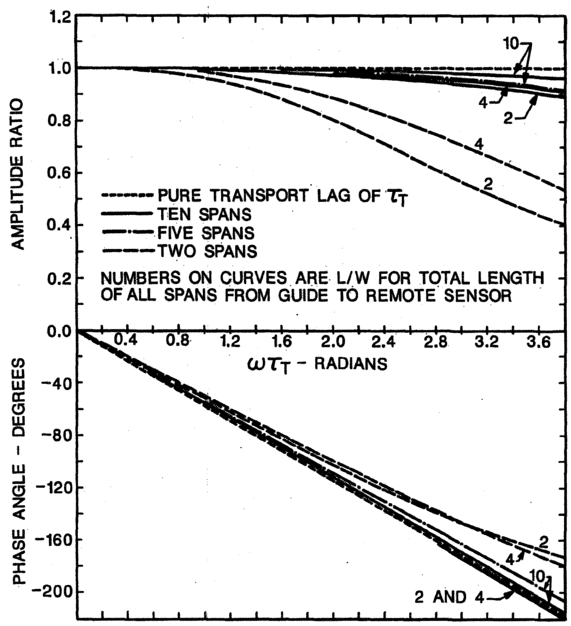

Figure 11 – Lateral Response of Multiple Equal-Length Short Spans
Between Guide and Remote Sensor

then be equal to the transport lag plus an allowance for the anticipated time required for correction by the guide. This control scheme would be accurate for some conditions, but could sometimes cause unwanted excursions, such as in the vicinity of a poorly aligned splice.

The following recommendation for control of the guide point (or position of the sensor, if the sensor has a narrow proportional band) of Sensor A in response to an error at Sensor B will result in smoother corrections than the periodic correction of the previous paragraph. Figure 10(b) shows an integrating guide-point positioner in the outer loop of the control system. The guide point of Sensor A (or the actual sensor) would be moved at a velocity proportional to the error at Sensor B, and inversely proportional to the transport time τ_T between the sensors.

Because of the 90 degrees of phase lag inherent in the integration 1/s, additional phase lag at the open-loop unity crossover frequency should be limited to approximately 45 degrees to assure a well-damped response.

At 45 degrees ($\pi/4$ radians) of phase lag caused by a pure transport lag τ_T, $\omega\tau_T = \pi/4$. For the approximation for multiple spans as discussed above, the relationship is quite accurate, as shown by Figure 11. For K_p/ω to be unity when the phase lag is 45 degrees:

$$K_p = \frac{\pi}{4\tau_T} \frac{\text{units/sec}}{\text{unit error}} \qquad \{1\}$$

Equation {1} demonstrates that the gain of the integrator must be inversely proportional to the transport time between Sensors A and B, or proportional to the speed of the web, and must be stopped (and probably reset) when the web is stopped.

If the gain is doubled from the proper gain of equation {1}, the open-loop unity crossover frequency is approximately doubled, resulting in a crossover near the "minus one" point of an open-loop polar plot, so that the oscillation frequency is approximately

$$\omega_n = \pi / 2\tau_T \text{ rad/sec, or}$$
$$f_n = 15/\tau_T \text{ cycles per minute,} \qquad \{2\}$$

where τ_T remains in seconds. Equation {2} is an aid in troubleshooting the integrating guide-point positioning system.

If the transport time between the two sensors is low (somewhat less than one second), the accuracy improves because of the faster response, but the guideline for adjustment of the gain of the integrator, equation {1}, may not be valid because of interaction between the positioner control system and the web guide control system.

Improvement of Web Guide Performance

Internal Feedback for Improvement of Electro-Mechanical Web Guide Performance. Most web guides of the last 50 years have been hydraulic. The better hydraulic servo valves have been difficult to match in performance, particularly in stiffness under load (achieving low dead band if the load has breakaway friction) by electromechanical systems. The modern trend, however, is toward electromechanical guides because of cleanliness (absence of leaks and spills of oil) and the scarcity of competent hydraulic technicians, and because of easier adaptation to computer control of processes. This topic is included to demonstrate that high performance (high frequency response and insensitivity to breakaway friction) can be achieved even for control of massive unwind and rewind stands in the metals industry, where hydraulic valves and cylinders are still used. It will also be shown that the capability of high frequency response is desirable even if the correction of high-frequency disturbances is not needed.

The need for adequate frequency response at the maximum load inertia for a guide actuator is obvious. Less obvious is the need for stiffness, or insensitivity to force disturbances. Velocity versus torque of a permanent magnet motor is shown in Figure 12, while a shunt DC motor with constant field current and controlled armature voltage has similar characteristics. The characteristics which are desired, however, are control of velocity which is unaffected by torque disturbances, as shown in Figure 13. The characteristics of a closed-center servo valve are closer to the desired behavior than the servo motor of Figure 12. The following discussion of inner feedback loops shows methods for approaching the desired behavior of Figure 13 with ordinary armature-controlled DC motors.

This paper is written with awareness that a web processing company is extremely unlikely to modify a web guide controller, or to build a special one. The purpose of the paper is to increase awareness of the high level of engineering behind a good electromechanical web guide, and to encourage critical evaluation of several aspects of the performance of a web guide.

The torque output of a given electric motor is determined by the current (actually, the magnetic flux density, which is determined by the current). Therefore, if the inability to generate sufficient torque is a problem, as in a guide with high breakaway friction, control of current needs to be improved. This improvement in control of current is not generally detrimental in applications where the improvement is not really needed. The following recommended control of current does not change the model of the "typical" web guide controller of Figure 7, but makes the model more accurate if breakaway friction is present.

Current feedback is accomplished by measuring the armature current as the voltage drop across a resistor which is in series with the armature. This voltage needs to be very low, so that the resistor for larger motors may be a copper bar. The control of current is further improved by inclusion of a PI controller after the summing of the current signal and the error signal from the sensor. A motor of adequate power along with proper current feedback compensation should have an electrical time constant which is negligibly small in comparison to the mechanical time constant. Specifically, τ_e should be less than 1/3 τ_m, where τ_m is the effective mechanical time constant after improvement by tachometer feedback, if a tachometer is used.

If the time constant of the PI controller for current is set equal to the original electrical

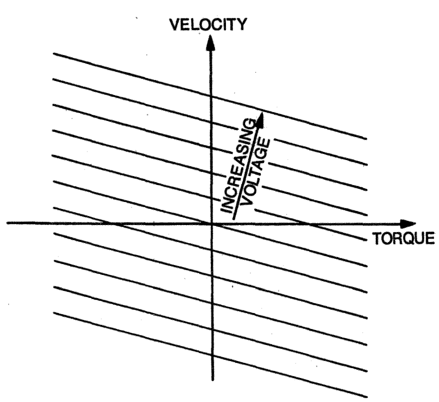

Figure 12 – Torque vs. Velocity of DC Motor with Control of Armature Voltage

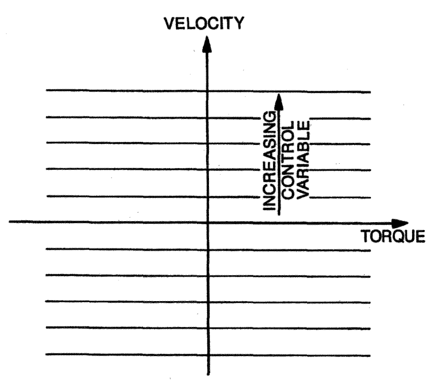

Figure 13 – Desired Torque vs. Velocity of Motor and Controller

time constant τ_e as suggested by Boulter and Gao [10], the resulting response has the same form as the original response, but with the potential of having the modified electrical time constant much lower than τ_e. The primary achievement from introduction of the current loop, however, is the elimination of the steady-state error caused by a load torque disturbance, hence elimination of the dead band inherent in a basic motor drive of a load on bearings with breakaway friction. In other words, the integration of the PI controller transforms Figure 12 to Figure 13. The feedback is necessary in addition to the PI control for prevention of conversion of the complete system to type 2, with consequent problems in achieving stability.

A tachometer mounted on the drive motor of a web guide actuator can reduce the mechanical time constant τ_m. Unlike a tension control system, where the goal is control of velocity, and unlike the current loop of the previous paragraph, a PI controller is not necessary in the velocity loop, except for proportional amplification. A deterrent to designing a velocity loop with an integral controller is that τ_m varies between installations because of different inertias of loads; furthermore, τ_m varies with the mass of the roll if a guide is shifting an unwind or rewind stand. With a turret stand, the inertia changes suddenly and at non-repeating times as a roll is loaded or removed. Therefore, a tachometer without integration is indicated.

Improving the dynamics with the current loop eliminated the need for considering the current and the torque disturbance, greatly simplifying the overall block diagram of Figure 14. The result of velocity feedback is also simple, not changing the form of the transfer function of the basic motor, but reducing the overall time constant.

Reducing Amplification of Disturbances with High-Performance Web Guides. A high open-loop gain at low frequencies results in an inversely-related dynamic residual error in correction of disturbance errors. The frequencies at which good accuracies are obtained are beyond the range of the polar graphs used in this paper. These graphs show relative stability, and are useful for improving stability. An improvement in stability, in turn, allows an increase in gain, resulting in improved low-frequency accuracy. A less obvious advantage of a high-performance web guide is the lesser magnification of errors at frequencies inside the unit disturbance circle.

Figure 15 shows the "typical" web guide with its frequencies of disturbances shown as the first number in each set of parentheses. If the dynamic response is improved by a factor of 5.0 by velocity feedback, as discussed in the previous article, and the gain is then increased by a factor of 5.0, the frequencies of disturbances are identified by the second numbers in the parentheses. At low frequencies, the open-loop amplitude increase of 5.0 results in a reduction of residual dynamic error to 1/5. The loci of the open-loop response

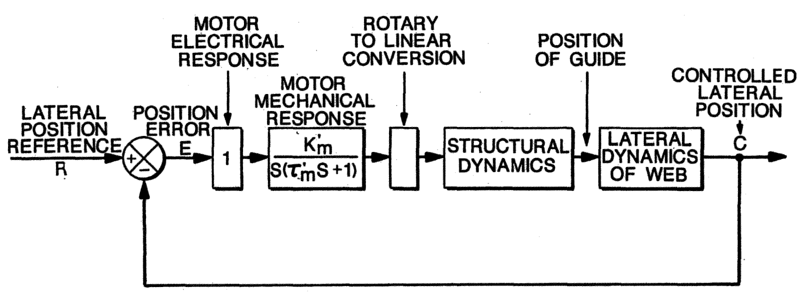

Figure 14 – Overall Block Diagram – Electromechanical Web Guide with Current and Velocity Feedback

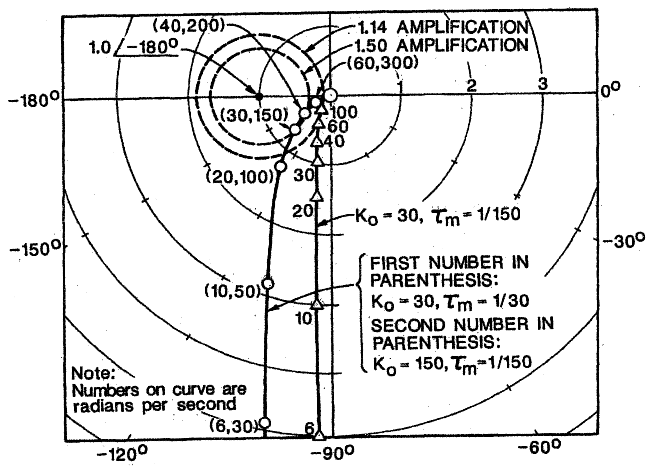

Figure 15 – High-Performance Web Guide Compared to "Typical" Model

are tangent to the 1.50 disturbance circle (radius of 0.67) between 30-40 radians per second for the "typical" system, and between 150 and 200 radians per second for the improved system. The maximum amplification for either system is thus 1.50, but this maximum is centered around 35 radians per second for the "typical" system and 175 radians per second for the "improved" system. This amplification around 175 radians per second is not desirable if a disturbance occurs at this frequency; however, this frequency is high for lateral disturbances.

If, in a given web guiding application, the "improved" performance is not needed, the gain can be decreased, with a decrease by a factor of 5.0 shown in Figure 15. The open-loop locus is then tangent to the 1.14 disturbance circle at around 100 radians per second. This means that the amplification of a disturbance is never greater than 14 percent, and this worse-case amplification is at a relatively high frequency.

Improvement of Low-Frequency Resonance with Lag Compensation. An extreme reduction of performance of a web guide occurs if compliance of the structure, with its low damping, results in a resonance near (or, worse yet, lower than) the desired open-loop unity-crossover frequency of the controller. The usual examples of compensation in textbooks are not effective because of the great increase in both amplitude and phase lag near the resonance. The need is to sharply cut off the response of the system to frequencies near the resonance, but the suddenness of a cutoff is limited by physical reality.

A compensator is usually identified by its effect on phase, "lag" or "lead". Although phase lead may be a goal for improvement of a control system, "lag" is a misnomer, as attenuation (reduction of amplitude) is the usual goal of a lag compensator, while the accompanying phase lag is inevitable. Similarly, a generally undesirable increase in amplitude accompanies a phase lead. Attenuation of the large amplitude of a structural resonance with no additional phase lag at lower frequencies would be desirable, but is impossible.

Analysis of a web guide with a low-frequency, poorly damped resonance has shown that the performance (loop gain and response at low frequencies) may be greatly improved with a cascaded second-order phase lag compensator with a high (1.0 to 2.0) damping factor and a cutoff frequency near the original resonant frequency.

It should be noted that the performance of an electrohydraulic web guide may also be limited by a low natural frequency caused by compression of the oil in the hydraulic actuating cylinders. The natural frequency may be especially low in applications in the metals industry wherein the cylinder for guiding has a long stroke for other functions, such as chucking. Such applications have often used sliding bearings (not anti-friction), resulting in a high damping factor and consequently better dynamics than a system with a low damping factor. The compensation discussed above would not then be generally helpful. Improvement would be accomplished by using a larger cylinder (requiring a larger pump, valves, fluid conduit, etc.), which would allow operation at a lower pressure. The precision required of hydraulically powered positioners in the machine tool industry forced the utilization of hydraulic motors (hence a small volume of oil under compression in the control circuit) driving ball screws, an exceedingly expensive solution. This problem of oil compression further suggests future employment of electromechanical web guides for all applications, including the metals industry.

The serious problems inherent in control of large masses, especially if accuracy is an important goal, further support consideration of a displacement guide just after an unwinder or just before a winder, as suggested earlier in this paper. Accuracy of the final winding might be improved by guide-point adjustment of the guide before winding by a sensor near the final windup, as discussed previously.

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