

**ANALYSIS OF WEB SPREADING INDUCED BY
THE CURVED AXIS ROLLER**

by

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ABSTRACT

This paper describes the development of a model for predicting the elastic deformations and stresses in a web crossing a curved-axis roller. The Finite Element Method was used to compute web displacements, forces and stresses. A preprocessor was developed to automatically convert the web material properties and roller geometry into a FEM mesh and a set of boundary conditions. The boundary conditions which produce web spreading were developed and incorporated into the model. The principal boundary conditions in this model are derived from the assumption that there is sufficient friction between the web and the roller to prevent slipping. Because of the nonlinear nature of the traction between the web and the roller, an iterative Finite Element solution technique was used. The model was used to perform a study of the effects of variations in geometry, material properties and operating conditions on the spreading behavior of the web/roller system. The results of this study are presented.

INTRODUCTION

A web is defined as any material in continuous flexible strip form [1]. The flexibility of the web is derived from the fact that the material thickness is small compared to the length and width of the material.

Web materials are usually delivered in the form of rolls, because of their compactness and ease of handling. Most web handling systems include equipment to unwind the roll of web material, transport it through the various manufacturing processes, and rewind it onto a roll. The material is usually supported, guided, and propelled by rollers.

The commercial pressures for increased productivity require higher and higher line speeds. As the speed of the line increases, the static and dynamic forces acting on the web become more extreme, increasing the likelihood of

defects in the material. Wrinkling is one of the most common web defects.

Several devices have been developed to remove wrinkles from webs or prevent them from forming. The most common spreading devices are the D-Bar spreader, the curved axis (Mt. Hope) roller, and the concave roller. The curved axis roller is the subject of this study. The concave roller is described in a companion paper. Portions of the concave roller paper that also apply to the curved axis roller model are referenced, and not repeated in this paper.

The curved axis roller is formed using a nonrotating bowed shaft. The bow in the shaft may be permanently fixed or variable by mechanical means. A set of bearings are placed over the shaft at even, closely spaced intervals. A tight flexible covering is placed over the set of bearings. The covering and the bearings rotate together as a unit. Figure 1 is a schematic drawing of a curved axis roller. As the covering rotates, points on the upper portion of the covering are moved out towards the end of the roller. This is caused by the bow in the shaft. Portions of the web in contact with the covering are also moved outward, provided there is sufficient frictional force between the covering and the web.

Although the curve axis roller is a relatively complex device when compared to most other rollers, its method of operation is intuitively simple. The roller is able to spread the web because points on the surface of the roller itself are moving outward. This is one of the advantages of the curved axis roller. Even when there is slippage between the web and the roller, as long as the material remains in contact with the roller, this simple spreading mechanism remains effective.

The curved axis roller does have several disadvantages. The first is the relative complexity and expense of this type of roller. The other disadvantages occur because of the need for a flexible covering on this device. The covering is not as durable as aluminum or steel rollers, particularly in harsh environments. The cover can be damaged by extreme temperatures and chemicals used in the manufacturing process. Also the behavior of the covering can change with time. The surface traction of the covering can change, reducing the effective spreading forces. The surface can deform into the spaces between the bearings, so that it no longer has the shape of a curved cylinder. Because of the alternating tensile and compressive stresses induced in the covering with each revolution of the roller, it can have a relatively short fatigue life. This results in cracking of the covering. In spite of these disadvantages, the curved axis roller is being used successfully in many web handling applications.

BACKGROUND

Shelton [2] used the idea of the idle arc as described by Swift to develop the principle of web transport and three corollaries. The principle of web transport is stated as follows:

If the friction between a moving web and a roller is sufficient to prevent slippage at the line of entering contact, the conditions at a given point in the entering span immediately upstream from the line of entering contact are transported toward, then around the roller in a plane which is perpendicular to the axis of rotation of the roller and which passes through the initial location of the point.

This principle is applied in much of the work in web guidance and control, and is used in developing the boundary conditions for the curved axis and concave roller models.

Shelton and Reid [3],[4] developed models for the lateral dynamics of webs and applied these models to web guide control systems. These web guide control systems generally rely on lateral shifting and pivoting of intermediate rollers to steer the web. The most important principle governing these devices is that the web will seek to align itself perpendicular to a roller in the entry span to that roller. Shelton used the equation for beam bending to model the lateral motion of the web due to the moments induced by the steering rollers. Shelton [5] also used the principles of web transport to investigate the dynamics of web tension control.

Pfeiffer [6] used the web transport principle and simple concepts from both narrow and wide web systems to offer rules of thumb for web guidance. He describes the spreading mechanism of the curved axis roller and the D-bar spreader. He also discusses factors governing the traction between the web and the roller.

The curved axis roller has been used in industry for many years. The insight gained from observing real applications of this device is useful in understanding the spreading mechanism.

Gallahue [7] describes the use of the curved axis roller to separate the web strips after passing through a slit. He recommends that two curved axis rollers be placed in series in a configuration so that all of the slits travel the same distance. This configuration minimizes the strains applied to the web material, and improves the quality to the wound roll by reducing the likelihood of interleaving or dishing.

Lucas [8] and [9] performed a study of the effectiveness of two of the common spreading devices; the curved axis roller and the D-bar spreader. He described the mechanism by which each of these devices works. He focused primarily on the use of these devices in separating web strips after slitting, and prior to winding on a roll. He lists the following problems associated with these spreading devices:

- (1) Good slit separation at the machine center, with poor spread at the edges
- (2) Good spread at low speeds and poor spread at high speeds
- (3) Good spread at high web tensions and poor spread at low web tensions
- (4) Poor wound roll edge quality
- (5) Web snapoffs at slit or spreader
- (6) Crowding of slits at wound roll
- (7) Thrusting of wound rolls against core boxes
- (8) Roll dishing

In his investigation, he discovers that excessive curvature of the curved axis roller can actually decrease the amount of spreading. He states that the elasticity of the material allows the web to spread only to a limited degree, and that the roller curvature should be compatible with this limited spreading. He also states that the effects of even the best spreading device are wasted if the web is not guided reliably.

Feiertag [10] developed a mathematical model for the spreading of an idealized web by a curved axis roller. He then used this model to develop design criteria for using the curved axis roller in wrinkle prevention, as guide rollers, and for slit separation. An idealized web is a web that has tensile stiffness in the machine direction, but no stiffness in the cross machine direction. This is a suitable model for a wrinkled web. From his analysis,

Feiertag concluded that less roller curvature should be used, and much longer entry spans should be used.

Reynolds [11] developed a two-dimensional finite element model of the curved axis roller. He used triangular linear simplex elements in his model. Because of the simple elements used, the number of elements had to be quite large. The program was also highly iterative, often requiring more than 100 iterations to converge on a solution. The combination of these two facts limit the program to running on a mainframe computer to give reasonable turn-around times.

THE CURVED AXIS ROLLER GEOMETRY

Figure 2 is a sketch showing the significant dimensions of the curved axis roller system. The dimensions are similar to those of the concave roller. An additional dimension is needed because the curved axis roller is not symmetric about its own rotational axis. Because of this, an angle indicating the orientation of the bow plane must be specified. The curved axis roller system also has a line of symmetry parallel to the machine direction located at the mid-width of the web. This symmetry is also used by the program to reduce memory needs.

The curved axis roller has a shape that cannot be matched by a web in an unstrained condition. As in the concave roller, the unstrained FEM mesh must be assembled using an average cylindrical roller. In the case of the curved axis roller, the roller diameter is constant, but the location of the center of the roller varies across the width of the web. Therefore, the unstrained model is assembled around a roller having the same diameter as the curved axis roller, and located at the average position of the curved axis roller. This is shown in figure 3.

Figure 3 shows that a point on the unstrained web is displaced in both the machine direction, and the direction normal to the surface. For the portions of the web actually in contact with the roller, the eventual boundary conditions will include known displacements for all degrees of freedom in both the machine direction and the normal direction. For those nodes not in contact with the roller, only the displacements normal to the surface are known in advance. In both cases, the geometry of figure 3 is used to calculate the normal direction displacements.

CURVED AXIS ROLLER MODEL BOUNDARY CONDITIONS

The governing effect in the behavior of the curved axis roller is the velocity of the points of the roller in contact with the web. As its name implies, the curved axis roller is a simple cylindrical roller whose axis of rotation is not a straight line, but an arc of a circle. Because all cross-sections of the roller have the same diameter, all points on the surface of the roller have the same velocity magnitude. It is the curvature of the roller axis that causes a variation in the direction of the velocity vector.

Although the curved axis roller is more complicated mechanically than the concave roller, its governing boundary conditions are more simple. Because all velocity magnitudes on the roller are equal, there is no tendency for the roller to shear any strip of web ahead of any other strip. This means that the FEM calculations on the entry span that were required for the concave roller are not necessary for the curved axis roller. In addition to the boundary

conditions shared with the concave roller, two other boundary conditions must be applied.

The process of making the unstrained web deform to the shape of the roller requires deformations both in the local Z direction and the local X direction. This is shown in figure 4.

The magnitude of the local X and Z displacements varies across the width of the roller. The figure shows that the nodes on the roller remain the same distance apart both before and after the required displacements. This means that there are no induced machine direction strains on the roller as were found in the concave roller.

These displacements applied to the nodes on the roller (which are applied as boundary conditions to force the initially unstrained web to become strained and conform to the curved axis roller geometry) do cause one problem. The first row of nodes in contact with the roller are also the last row of nodes in the entry span. The applied boundary displacements cause this first row of nodes to receive a machine direction displacement profile relative to the entry span that would induce local machine direction strains in the entry span. This is shown in figure 5. Because there is no physical reason for these displacements to exist (no velocity magnitude differential across the roller to induce this displacement / strain profile), they should not be left in the model. To remove these extraneous displacements the following procedure is used:

- (1) Start with the zero local displacements of the unstrained web.
- (2) Apply the fixed displacements in the X and Z direction required to make the web conform to the roller.
- (3) Subtract the local X displacement of the first node on the roller from that node and from all nodes that follow it on the roller.
- (4) Add in the local machine direction displacements at each node on the roller to apply the nominal line tension at those nodes.

As in the concave roller model, a final boundary condition is required to enforce the no slip condition. As before, all points on the roller move in a circle located in a plane perpendicular to the roller axis of rotation. Because the axis of rotation is not a straight line, these planes are not parallel. Instead, these planes extend radially from a line passing through the center of curvature of the axis. Because the planes are not parallel, the velocity vectors are not parallel. This is the principal reason for the spreading effect of the curved axis roller.

A multi-point constraint may still be used to relate all of the nodes on the roller to the initial point of contact. The constraint requires that all of the points on the roller having the same nominal Y location should continue to remain in a plane. They are forced to remain in the same plane as the velocity vectors. This plane is the local X-Z plane at each node on the roller. This type of constraint requires that a pair of nodes be separated by a constant offset in the local Y direction. This constraint is shown in figure 6. Node 2 is locked to node 1 with a constant offset. Node 3 is also locked to node 1 with a different constant offset. The offsets of the two constraints are selected so that nodes 1, 2, and 3 remain in the proper plane.

THE SPREADING PROCESS

The next stage in modeling the spreading rollers is the actual spreading process. This process requires an iterative search for a set of cross machine

direction displacements that are compatible with the condition of normal entry to the roller.

The search process can be posed as a nonlinear least squares curve fitting problem which in effect is a multidimensional nonlinear optimization problem. It can be stated as follows:

Find the set of applied forces which minimize the sum of the squares of the deviations from normal entry to the roller. The minimum value of this sum is known in advance to be zero.

The set of applied forces can be selected in two ways. A force can be chosen independently for each node at the end of the entry span. This gives as many independent variables for the optimization process as there are nodes across the width of the web. For the mesh chosen, this would give an eleven dimensional optimization problem.

A better approach is to use a function to define the force distribution across these nodes. The problem then becomes one for finding the proper values for the coefficients of this function to minimize the least square error. This can greatly reduce the order of the optimization problem. The simplest choice for the forcing functions are simple polynomials.

The lowest order polynomial is a simple constant but this does not allow any variation of force across the roller width. It seems unlikely that this would allow all of the nodes to approach normal entry to the roller.

The next order polynomial is a straight line. The line is defined by two coefficients, and does allow a force variation. If the linear force profile allows sufficient variation in force to approach zero error, then the problem is reduced from an eleven variable optimization problem to a two variable problem. This turns the problem from one that would probably never converge to a reasonable solution into one that should converge in a relatively short time.

The linear force function was implemented in the spreading roller analysis program. This simple function allows the search to converge in a matter of minutes to very acceptable accuracy. The Nelder-Mead Simplex method was used to perform the optimization process. The objective function for the search is given in equation (1).

$$\sum_{i=1}^{N_w} ((\text{Slope before roller}) - (\text{Slope after roller}))^2 \quad (1)$$

DEFORMATIONS AND STRESSES PREDICTED BY THE CURVED AXIS ROLLER MODEL

The distributions of deformations, stresses, and friction forces over the surface of the web are calculated for an example system by the spreading model program. The results are summarized using X-Y plots to display the spreading deformations, and 3-D contour lines to display the stress distributions.

Figure 7 shows the effective spreading for the base parameters for the curved axis roller. This plot shows several important features of the curved axis roller model.

The first thing to notice is that the slope of the curves is not zero in the area where the web contacts the roller. The amount of spreading does not remain constant over the surface of the roller. Instead, additional spreading occurs. This was expected from the geometry of the roller. The spreading effect of the curved axis roller occurs as a result of two separate mechanisms. The first and most obvious mechanism is the spreading action of the roller cover rotating on the curved shaft. The second is the steering of web streamlines so that they approach the roller normal to the line of contact. The slope of the streamlines at the end of the entry span is the same as the slope over the roller. The plot exhibits both of these effects, an increase in spreading over the roller with a smooth transition in the entry span. The slope of the curves become negative nearly instantly as the web leaves the roller. There are no friction forces in this region to sustain the spreading that was developed over the roller. The displacement streamlines also converge to zero as in the concave roller. The span length affected by the roller is again approximately one web width before and after the roller.

Figure 8 shows the distribution of machine direction stresses in the curved axis roller model. The first thing noticed is that the range of MD stresses in the curved axis roller is not nearly as large as that of the concave roller. In addition, the high and low stresses occur in very localized regions. Over most of the web, the MD stresses are essentially uniform and equal to the nominal stress in the line. The near uniformity of the MD stresses should not be surprising. All cross-sections of the curved axis roller have the same diameter. Therefore, the MD strains over the roller should be nearly uniform. Also, there is no shearing action as was seen in the concave roller model.

Figure 9 shows the cross machine direction stress distribution for the curved axis roller model. At first glance, it looks very similar to the CD stress distribution for the concave roller. The largest stresses are on the roller at the center of the web, with the stresses dropping to near zero at the edge of the web. There are two essential differences. The first is the absence of parallel contour lines over the surface of the roller. Because additional spreading occurs over the roller, the CD stresses continue to increase over the roller. The second difference is the absence of the large region of compressive stresses in the exit span. The shearing mechanism that caused these compressive stresses in the concave roller is not present in the curved axis roller.

There is a region of compressive stress indicated at the edge of the web in the entry span. In contrast to the concave roller, both the magnitude of the stress, and the size of the region are relatively small.

Figure 10 shows the shear stress distribution for the curved axis roller model. Both the range and the magnitudes of the shear stress distribution are smaller than those of the concave roller. This is consistent with the previous plots; the curved axis roller does not exhibit the same shearing mechanism as the concave roller.

The plot shows that the shear stress has a value of zero at all points on the web symmetric centerline. On the roller surface, the shear stress increases to a maximum value at the edge of the web. In the entry span, the shear stresses quickly dissipates to a nearly uniform value of zero. In the exit span, the shear stresses are slightly negative near the roller, and dissipating to a nearly uniform value of zero.

ANALYSIS OF PARAMETER VARIATIONS

The previous section examined the deformation and stress distribution over the entire web surface for a base set of parameter values. In this section, the parameters will be varied around those base values. This study required approximately 35 runs of the computer model. It is not feasible to discuss the resulting stress and deformation plots for each of those runs. Instead, representative values will be tabulated from each of those runs, and combined in a set of summary plots. These summary plots will be examined for trends in the response of the model to parameter variations.

The parameter values used in this section are identical to those used in the concave roller study, and are given in Table I of the concave roller paper.

Roller Profile Radius of Curvature

Figures 11 through 14 show the effects of curvature on the curved axis roller. The roller radius of curvature is intuitively the most significant parameter for both types of roller. The roller curvature is the reason that these rollers spread the web. The amount of curvature is the only thing that differentiates these rollers from simple cylindrical rollers. The curves show that the models produce results that match intuition.

The max spread and max friction curves show decreasing values with increasing radii of curvature. A radius of curvature of infinity produces a cylindrical roller. Thus, the behavior of these rollers should approach the behavior of a cylindrical roller as the radius of curvature approaches infinity. This behavior is shown by all of the curves for both the concave and the curved axis roller. For large radii, both the max spread and the max friction approach zero. In addition, both the max and the min MD stresses approach the nominal line tension, and the max and min CD stresses approach zero.

Bow Plane Angle

Figures 15 through 18 show the effects of bow plane angle on the curved axis roller. These curves suggest the reason for the configuration in which these rollers are normally used. The bow plane angle for the curved axis roller is normally selected so that the wrap angle is bisected by a plane perpendicular to the bow plane. For a 90 degree wrap angle, this would require a bow plane angle of 45 degrees down from the horizontal. The MD stress curves show that this gives the minimum variation in MD stresses. The reason that this minimum variation occurs in this configuration can be seen from the geometry of the roller. In this configuration, the total path length of all web streamlines are essentially equal. Any deviation from this optimal configuration results in different path lengths for different streamlines, and therefore a larger MD stress distribution.

The curve for max spreading displacement shows a maximum displacement for bow plane angles between 30 and 40 degrees, with decreasing amounts of spreading for the larger angles. This behavior is the result of two conflicting conditions. The curved axis roller spreads the web both in the entry span, and on the roller. The spread in the entry span is caused by the web being steered to normal entry. The steering has maximum effect when the bow plane is oriented parallel to the entering web span (zero degrees wrap angle), because the spreading components of the velocity vectors are in the plane of the web. At any other angle, the velocity components must be projected into the plane of the web using a cosine function, diminishing the spreading effect on the entry span.

The spreading that occurs on the roller is also a combination of two things: the length of the web in contact with the roller, and the angle between the web surface and the bow plane. Maximum spreading on the roller occurs when the web is parallel to the bow plane. This optimum orientation occurs at only a single line of contact. The best orientation for spreading on the roller is the orientation most commonly used with the curved axis roller as was described in the previous section (bow plane of 45 degrees for a 90 degree wrap angle).

The bow plane orientation for maximum spreading in the entry span does not coincide with the orientation for maximum spreading on the roller. It stands to reason that the bow plane orientation for maximum total spreading is a compromise between these two orientations. The max spread curve shows that this compromise occurs somewhere near 30 degrees. If obtaining the maximum spreading was the only objective, this curve would suggest that the industry change the manner in which curved axis rollers are installed. But, as was shown in the previous section, spreading is not the only consideration. The stresses induced in the web, and the forces required for spreading are also important.

The curve for max coefficient of friction shows that friction is not heavily dependent on the bow plane angle.

The curves for the max and min MD stresses show very interesting behavior. The max and min stresses converge as the bow plane angle increases from 30 degrees to 45 degrees. After 45 degrees, the curves diverge. Again, the MD stress variation is larger as the bow plane angle deviates from the optimum value of 45 degrees.

The curves for max and min CD stress show only slight variation for variations in the bow plane angle. They do show larger CD stress variations for bow plane angles less than 45 degrees. In addition, the best value (smallest compressive stress) for the min CD stress occurs with a bow plane angle of 45 degrees.

RESULTS

The general results obtained in the concave roller model also apply to the curved axis model, and are not repeated here.

The curved axis roller model predicted friction values that were higher than expected, although the friction values are lower than those predicted by the concave roller model. The MD forces in the curved axis roller are not the predominant forces. Both the MD and the CD forces are of similar magnitude.

Common usage of the curved axis roller orients the roller so that the wrap angle between the web and the roller is bisected by a line perpendicular to the bow plane. For an incoming horizontal web with a 90 degree wrap angle, this would require a bow plane angle of 45 degrees down from horizontal. The curved axis roller model shows that this convention is used for very good reasons. This orientation produces acceptable spreading deformations, but it is the stress distribution that is the primary reason for using this orientation. Deviation from this optimal orientation results in significantly greater MD stress variations.

RECOMMENDATIONS

The principal recommendations for extension of this work pertain to extending the capabilities in the model. Three new capabilities in the model

are of immediate interest to this author. First, the ability to allow slipping should be added. This will require a significant increase in computing power to perform the large number of iterations in a reasonable amount of time. Because of rapid improvements in computer speed, accompanied by reductions in price, machines capable of modeling slipping should be available to most engineers in the near future.

The model should also be modified to allow the web to move off of the centerline of the roller. Because these spreading rollers are de stabilizing devices, it would be useful to calculate the maximum displacement of the web centerline, and the resulting stress distribution. This would be a first step in modeling the lateral dynamics of webs on spreading rollers.

Finally, the spreading model should be combined with a wrinkle model to investigate the ability of these rollers to prevent wrinkling. A very simple wrinkle model might be a lateral compressive force or displacement distribution at some point in the entry span. The maximum compressive stress remaining at the entrance of the roller should be a good indication of the ability of the roller to prevent wrinkling.

ACKNOWLEDGMENTS

The authors of this paper would like to thank the sponsors of the Web Handling Research Center at Oklahoma State University for their financial and technical support of this research.

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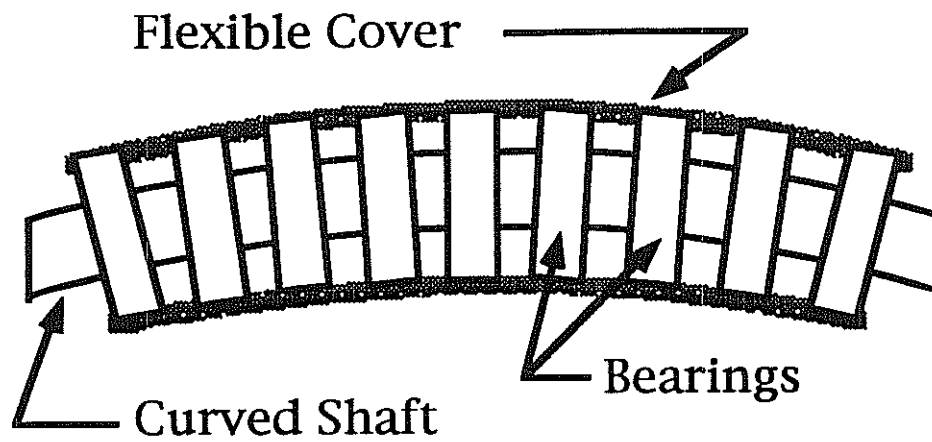


Figure 1. A Curved Axis Roller

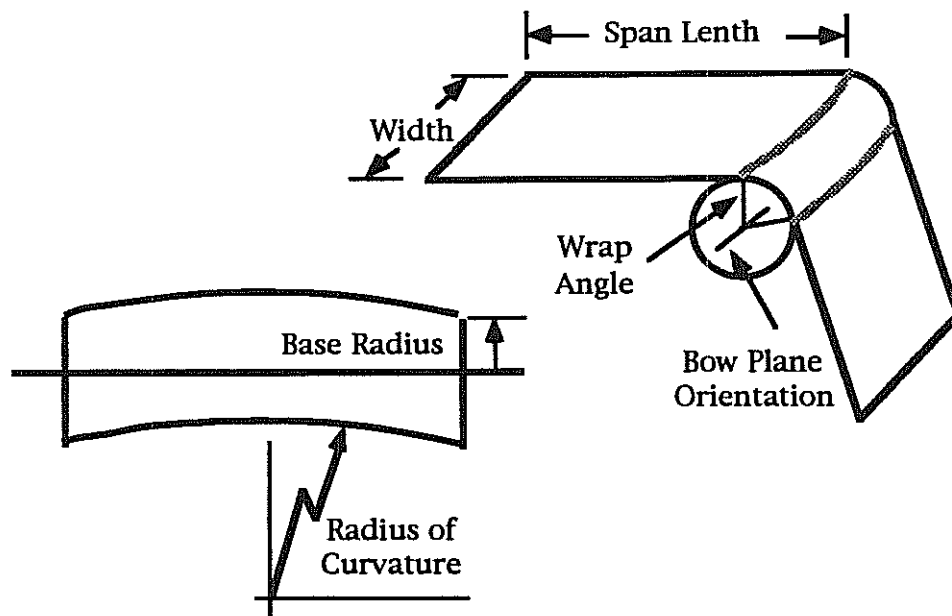


Figure 2. Curved Axis Roller Dimensions

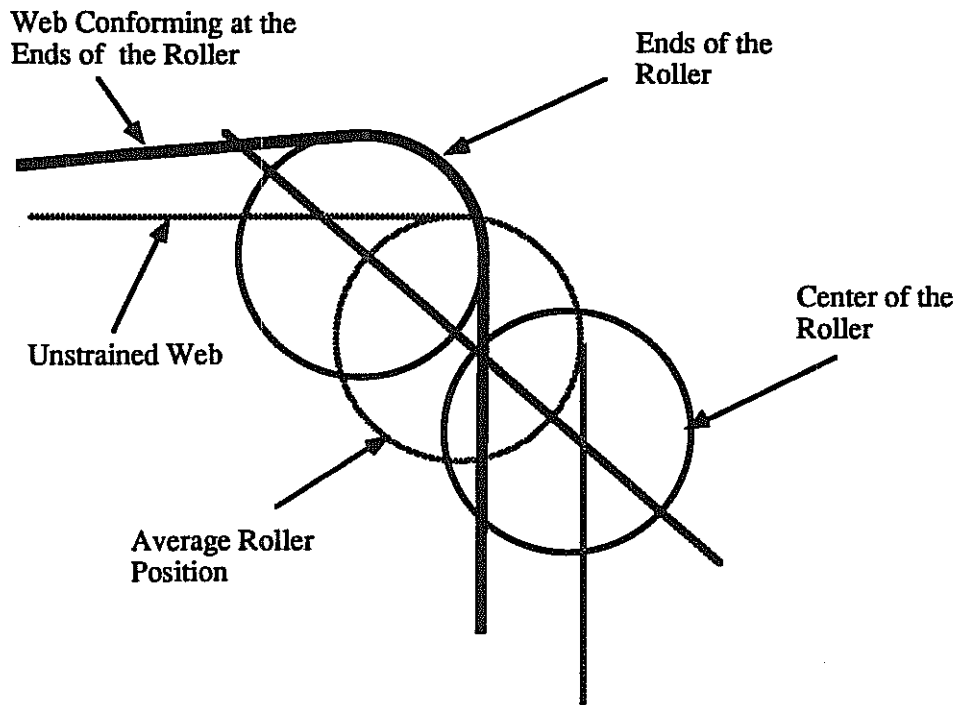


Figure 3. Web Deformed from Average Roller Position

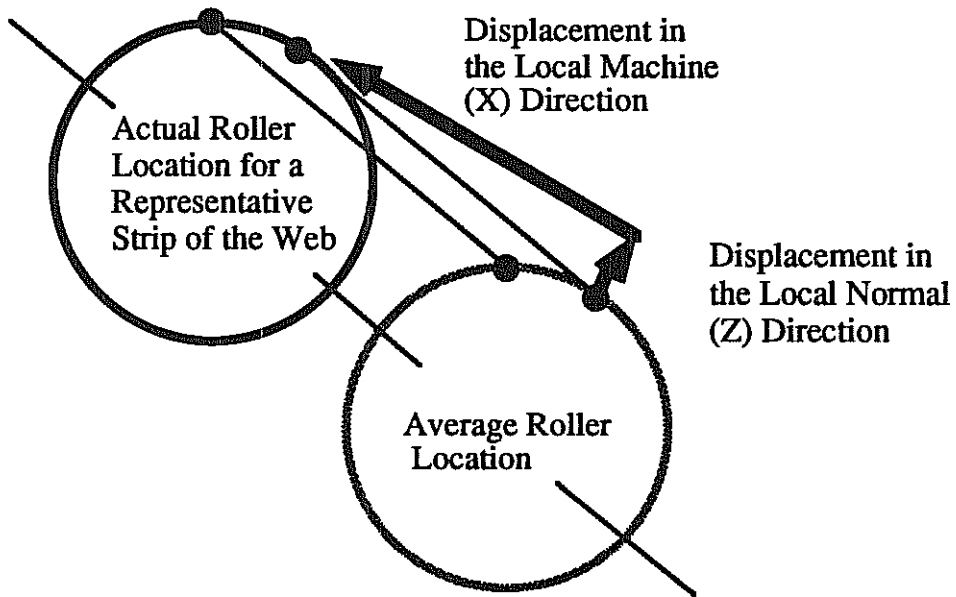


Figure 4. X and Z Displacements Required to Conform the Web to the Roller

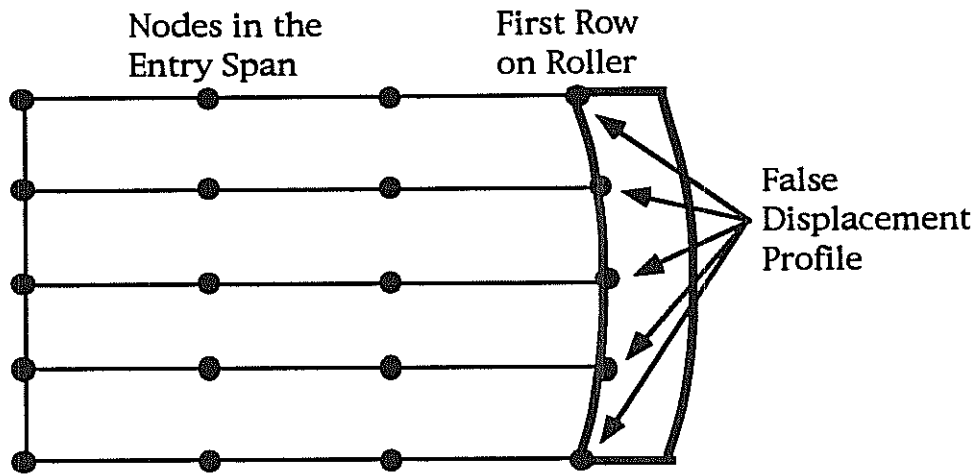


Figure 5. False MD Displacement Profile in the Entry Span Resulting from Applied Roller Displacements

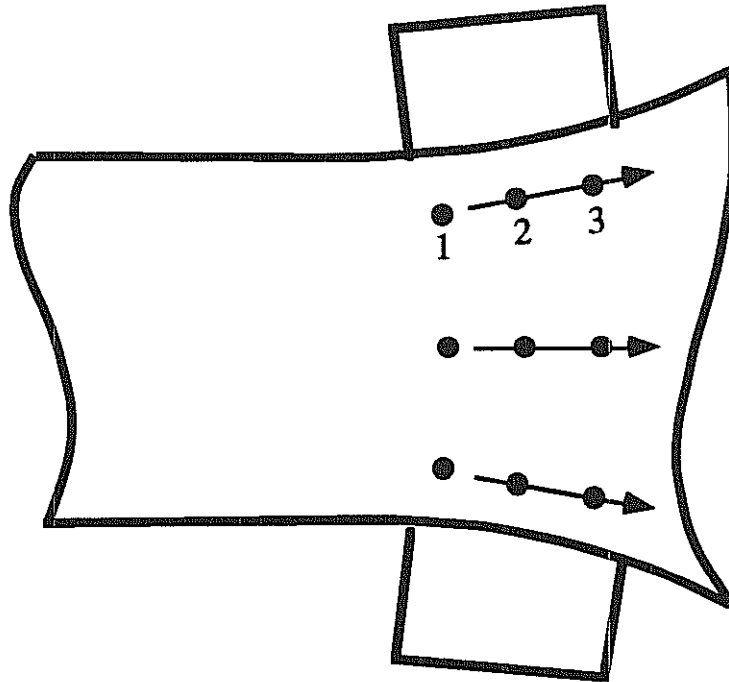


Figure 6. Multi-point Constraints for the Curved Axis Roller

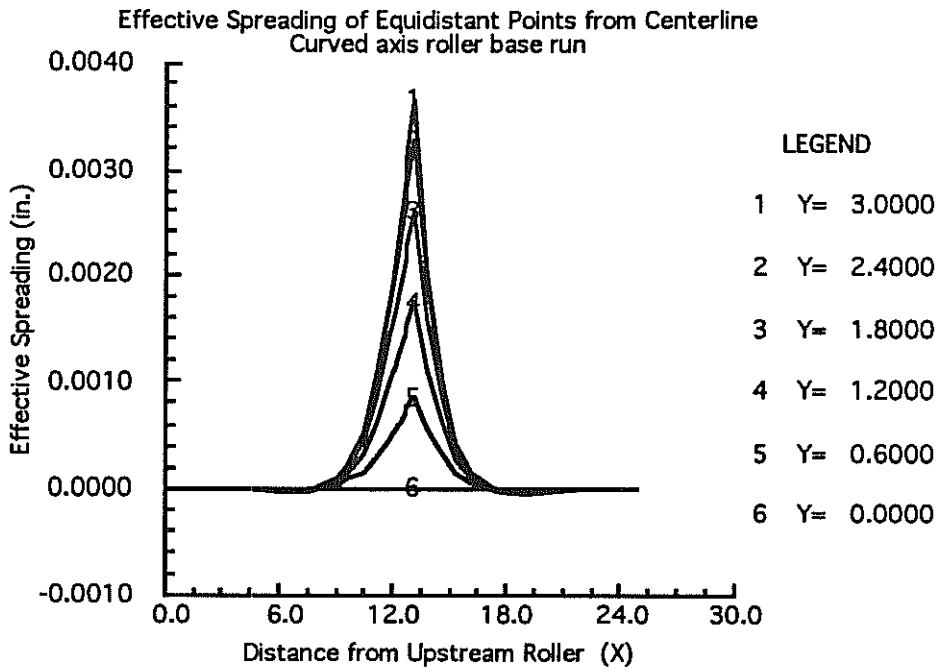


Figure 7. Curved Axis Roller Base Run Effective Spreading

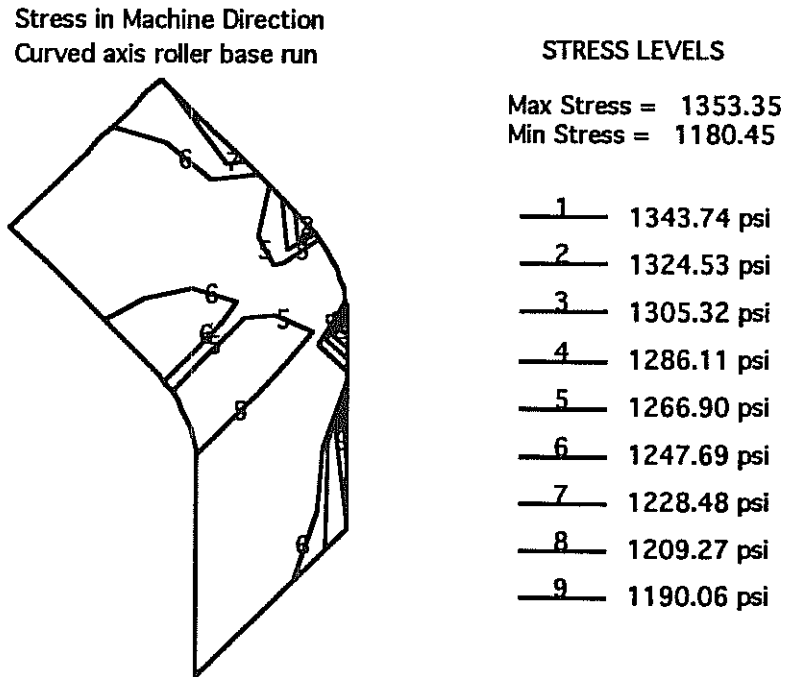
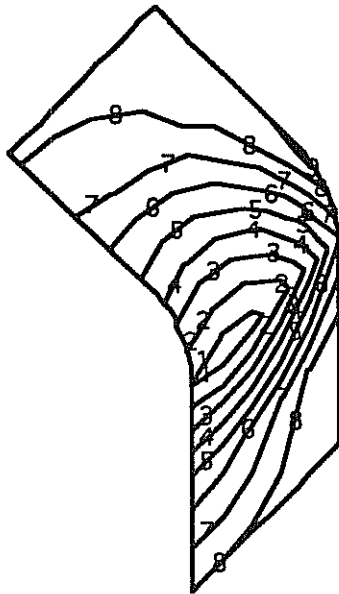


Figure 8. Curved Axis Roller Base Run MD Stresses

Stress in Cross Machine Direction
Curved axis roller base run



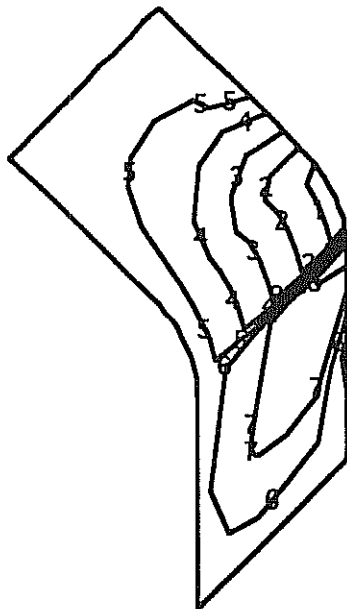
STRESS LEVELS

Max Stress = 176.83
Min Stress = -20.71

1	165.86 psi
2	143.91 psi
3	121.96 psi
4	100.01 psi
5	78.06 psi
6	56.11 psi
7	34.16 psi
8	12.21 psi
9	-9.74 psi

Figure 9. Curved Axis Roller Base Run CD Stresses

Shear Stress
Curved axis roller base run



STRESS LEVELS

Max Stress = 76.29
Min Stress = -57.56

1	68.85 psi
2	53.98 psi
3	39.11 psi
4	24.24 psi
5	9.37 psi
6	-5.51 psi
7	-20.38 psi
8	-35.25 psi
9	-50.12 psi

Figure 10. Curved Axis Roller Base Run Shear Stresses

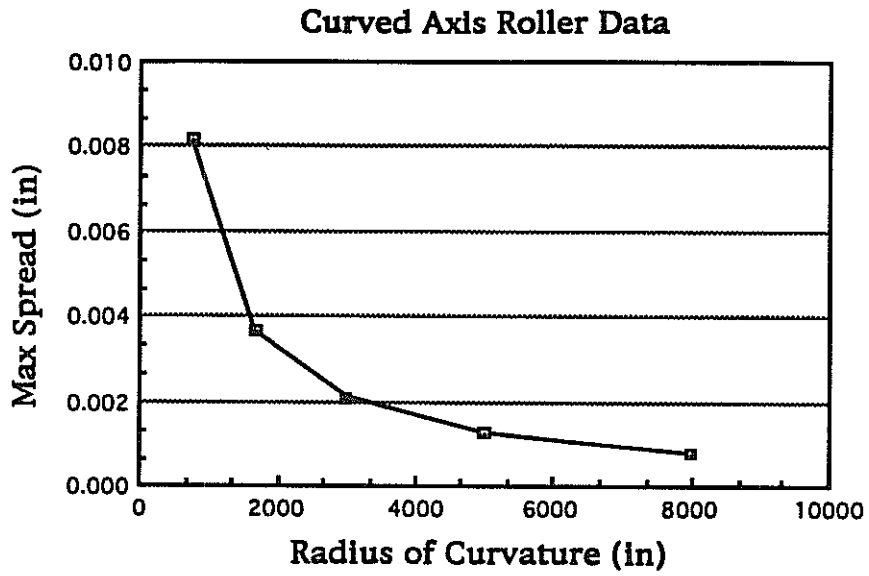


Figure 11. Curved Axis Roller - Spread vs. Radius of Curvature

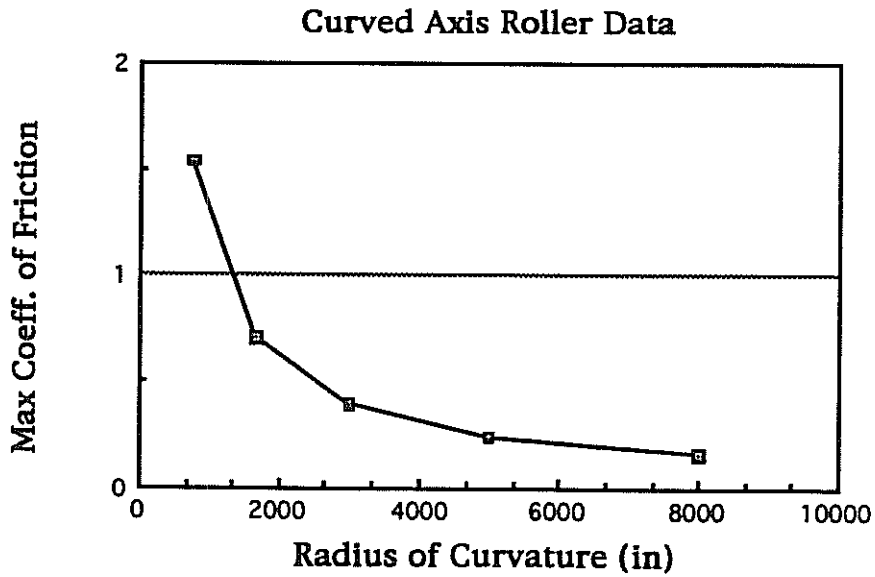


Figure 12. Curved Axis Roller - Friction vs. Radius of Curvature

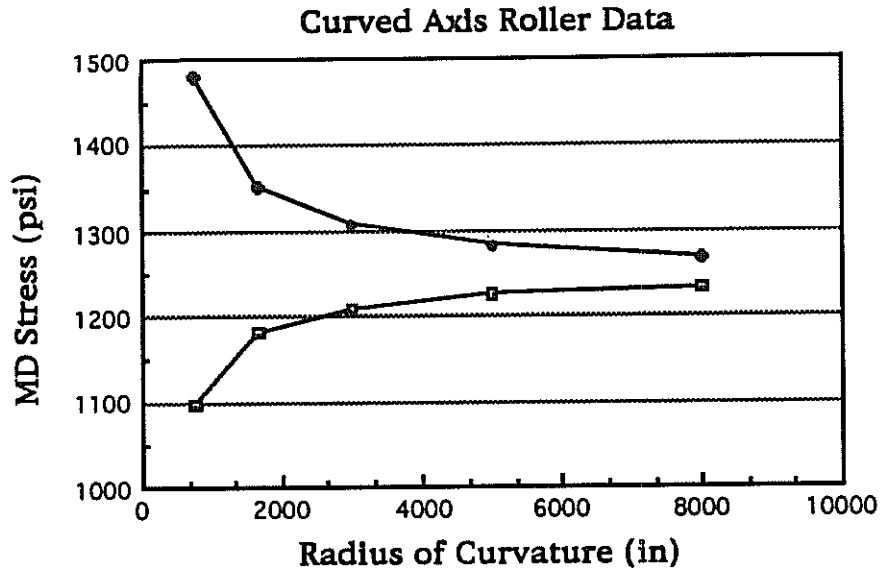


Figure 13. Curved Axis Roller - MD Stress vs. Radius of Curvature

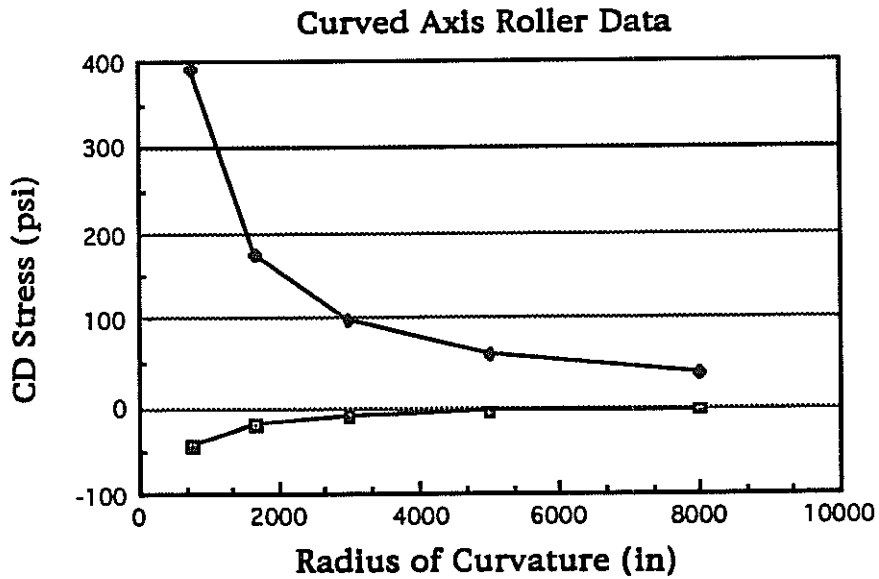


Figure 14. Curved Axis Roller - CD Stress vs. Radius of Curvature

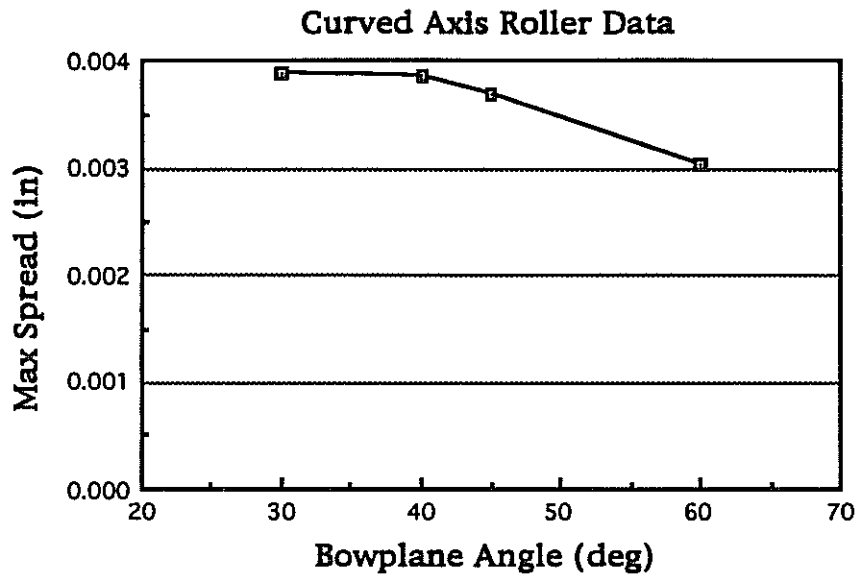


Figure 15. Curved Axis Roller - Spread vs. Bow Plane Angle

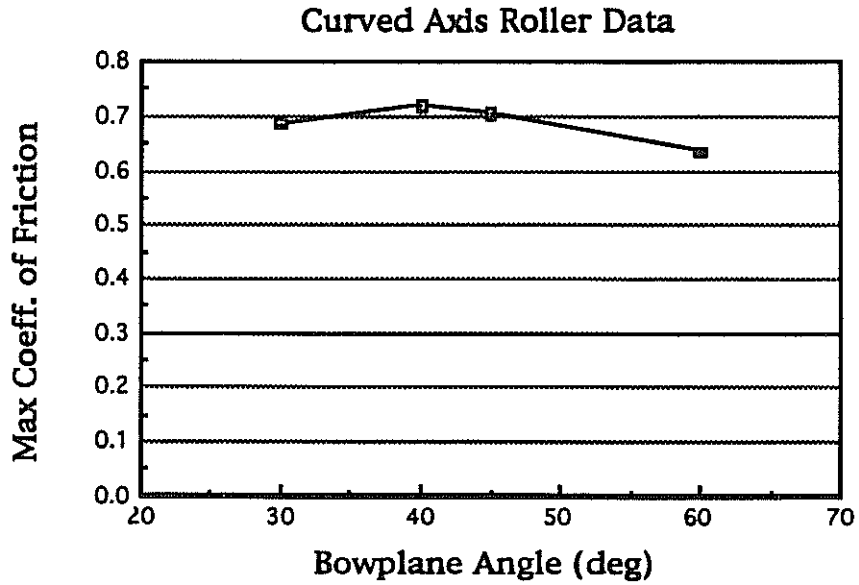


Figure 16. Curved Axis Roller - Friction vs. Bow Plane Angle

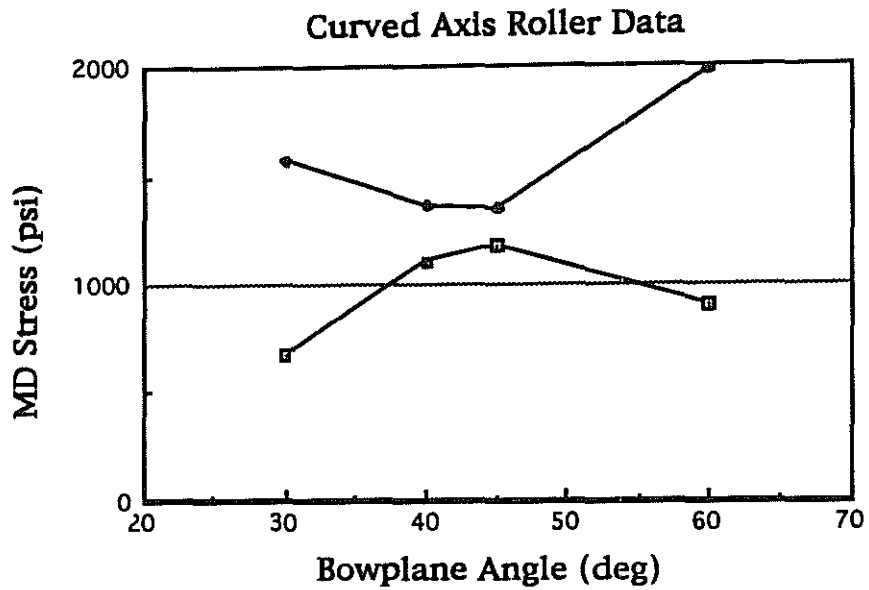


Figure 17. Curved Axis Roller - MD Stress vs. Bow Plane Angle

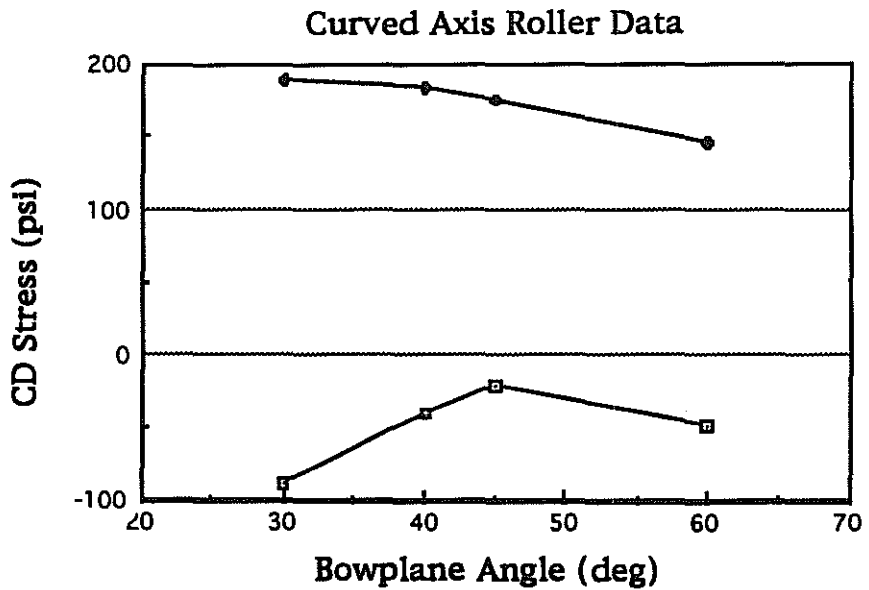


Figure 18. Curved Axis Roller - CD Stress vs. Bow Plane Angle

QUESTIONS AND ANSWERS

Q. Isn't it true that to remove wrinkles, you simply need to eliminate the lateral compression, and don't need to induce lateral tension?

A. You are correct, and as a matter of fact, I've done some initial modeling to induce some lateral compression and then to look at the effect of the roller, but I would again say, there's what you want to do, and then there's what you're going to do. You may only want to make it laterally taut with no lateral stress, but you're simply not going to be able to design a system to do that. So, I think that these models are valuable in simply telling you, if you do put one of these rollers in, you're going to get more than just lateral tautness, what kind of stresses are you going to get, and can your system stand those stresses?

Q. It seems to me that you really need to incorporate slipping into this model as well.

A. Well, I would say, and I hope that Bruce Feiertag would back me up, that you would prefer to design a system, first of all does not have the amount of curvature that would cause the slip, you'd rather design it so that there is no slip, which would be a very mild curvature.

Bruce: Let me respond to that as well, we just talked about the dynamic consideration. Traction, coefficient of traction, is one of the most difficult things to keep constant, on any roller, and if you've got a different traction from the left side to the right side of the roller, you're going to be in very bad shape, because you've got slippage going on. So, then we have many people who want no slip whatsoever, because slipping means problems in terms of their web finish. So, I think there are many design cases, where you want no slip.

Q. Can the curved axis roller actually cause wrinkling down stream from the roller?

A. I would say with the curve axis roller and no slip, I don't think so, but the no slip is really the big qualification. With the concave roller, my models show downstream from the roller, about half web with the _____, an interesting compressive zone, which would actually be putting the wrinkle back in.

Q. Do you ever see the web lifting off of the roller and not conforming to the shape of the roller?

A. Not with the model parameters that I've looked at, but I suspect if I went to really extreme curvatures, very small radii curvature, I probably would see negative forces to make it conform particularly for the concave roller at the center, I would expect to see that. I haven't I may go back and do that this afternoon, that would be an interesting thing to do and I would expect to see that.

Q. But, what that is really saying is you've got that much curvature in the roller, you want it to contact the roller.

- A. Right, and it'd be kind of ridiculous to design such a device.
- Q. Do your friction curves represent the average value of friction required on the roller?
- A. Well, for the friction plots, it was the point on the roller having the most severe friction requirement. And, of course, for the friction requirement distribution across the roll was not uniform, so that was the worst one.