# OPTIMIZING NIPPED ROLLER SYSTEMS

**By**

**Timothy J. Walker**
**TJWalker + Associates Inc.**

## ABSTRACT

Nipped rollers are central to many web converting processes – printing, coating, embossing, calendaring, and winding. Uniform products require uniform load and pressure from the nipping process. Uneven nipping not only creates product variations, but can lead to high levels of waste from web shifting, wrinkling, wound roll defects.

Imperfections are inherent in nipping – from variations in roller diameters, product thickness, and roller deflection. For most nipped process, many variations are hidden by the compliance of a rubber roller covering. However, any nipped roller system has competing design factors. Hard rubber coverings last longer, but are less compliant. Smaller diameters create higher pressure, but will have more bending and deflection under uneven loading. Is there an answer or guideline to rubber roller mechanics besides "What did you use before?"

This paper reviews the process of balancing rubber covering and roller design based on one simple goal: ensure roller bending deflections are less than rubber roller compliance. The models for rubber covering and roller deflection will be combined to give a clear view of this goal and provide design guidelines to ensure it is achieved. In addition, the paper will review methods to assess nipped process uniformity and remedies to create more uniform nipped processes.

## NOMENCLATURE

| | |
|---|---|
| $a$ | distance from roller end to core shell to shaft support, m |
| $b$ | nipped roller machine direction contact length, m |
| $E$ | Young's modulus of elasticity, Pa |
| $E_0$ | Young's modulus for rubber without confinement effects, Pa |
| $F$ | nip load force, N |
| $I_x$ | moment of inertia along x-axis, $m^4$ |
| $N$ | distributed load over nip width, N/m |
| $r_1$ | rubber roller outer radius, m |

$r_2$ secondary base roller outer radius, m
$r_{EQ}$ equivalent radius, m
$r_{CO}$ rubber roller core cylinder outer radius, m
$r_{CI}$ rubber roller core cylinder inner radius, m
$t$ nip roller cover thickness, m
$w_R$ width of rollers, m
$w_B$ distance between live shaft journal bearing supports, m
$w_S$ distance between inset internal core supports, m
$y_c$ deflection at the center of the roller, m
$y_{MAX}$ maximum deflection of the roller, m
$y(x)$ deflection at distance from roller center, m
$\delta$ rubber cover indentation at center of MD contact length, m
$\nu$ Poisson's ratio, dimensionless
$\varphi$ Ratio of roller deflection to rubber indentation, dimensionless
$H_{SA}$ Rubber hardness (International Rubber Hardness Degree, Shore A Durometer)

## INTRODUCTION

Though most rollers in converting and similar processes are not nipped, many of the value-adding processes are dependent on uniformity of nipped roller friction, dimensions, or pressures. Some of the nipped roller functions critical to processing webs include:

- Increasing the frictional bond to a driven roller to control web speeds and tensions.
- Controlling air between web and roller: eliminating concerns about loss of traction and slipping from air lubrication, minimizing air in dry lamination, reducing the insulating air between the web and chilled or heated roller, preventing backside corona treatment, and limiting the volume of air allowed into a winding roll.
- Controlling coating thicknesses set by geometry of a gap or pressure between nipped rollers (in addition to other coating variables: roller surface texture, coating viscosity, process speeds, etc.).
- Applying thickness direction pressure to aid in compressing, embossing, or micro-replicating a web.
- Calendering a web to increase length, reduce thickness, or burnish the web surface.

In these valuable nipped roller functions, process and product quality is dependent on cross-machine direction (CD) uniformity. Each process may have a different threshold for how much variation will cause a defect, but minimal CD variation is usually rewarded.

## MECHANICS OF RUBBER COVERED NIPPING ROLLERS

The average pressure between nipped rollers is the total of normal forces over the area of contact {1}. The area of contact between nipping rollers is the width of the roller times the machine direction length of contact (a.k.a. MD footprint length). Nip forces divided by width (or web or roller) is the nip distributed load (or simply nip load) in units of force per width (N/m or lbs/in). The footprint length, b, (see Figure 1) is a function of roller radii and indentation or overlap of the rollers beyond tangential contact.

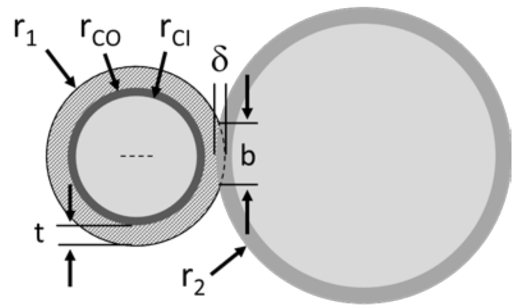


Figure 1 – Geometry of Nipped Roller Footprint

$$P_{avg} = \frac{F}{w_R b} = \frac{N}{b} \qquad \{1\}$$

Nip loads range from a gentle touch to more load than an asphalt steam roller (up to 30 tons per 2m width). The nip load a given process needs is mostly dependent on what material the nip is expected to control (gas, viscous liquid, solid) and process speed (i.e. time in the nip).

Nipped roller systems are controlled by either setting applied force or roller-to-roller position. Nip positioning systems may set a gap between rollers or indent one roller into the other by a set dimension beyond tangential contact. If nipped roller are set by a controlled gap, the indentation will be the difference between the compressed web thickness and the gap spacing. Indentation control is usually reserved for nips with at least one rubber covered roller.

Force-loaded systems can be as simple as a bottom fixed roller and the the weight of a top roller. For controllable loads, either pneumatic or hydraulic cylinders can relieve roller weight or create loads greater than gravity (or where nipping is perpendicular to gravity).

End-loaded nipping rollers have much in common with bridges. Both are simply supported beams with distributed loads (see Figure 2). The offset of the distributed load relative to the simple supports induces a moment in the beam, creating curvature and deflection (non-straightness).

When nipping rollers deflect under load, they bow away from each other, moving contact loads and pressures toward the ends of the rollers. Without compliance in the nipping system, forces and pressures in the centerline of the nip will quickly go to zero.

Equations for simply supported beams with a uniform distributed load provide a first estimate for nipping roller deflection. {2} This equation is used for dead shaft rollers and assumes the bearings do not transfer a moment from the shaft bending to the core shell cylinder bending. For live shaft rollers, the moment applied to the core shell cylinder from the shafts or journals increases the deflection of the core cylinders. {3} This increase in deflection is worse with longer shafts or journals.

$$y_c = \frac{5N{w_R}^4}{384EI_x} \qquad \{2\}$$

$$y_c = \frac{12N{w_R}^4}{384EI_x}\left(\frac{w_B}{w_R} - \frac{7}{12}\right) \qquad \{3\}$$

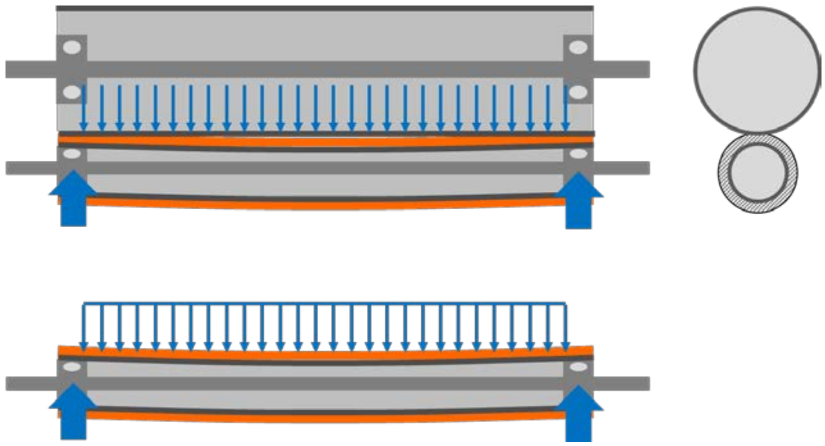

Figure 2 – Simply Supported Roller Shell with Uniformly Distributed Load

The bending moment of inertia of the core thin-walled shell cylinder is a function of the outer and inner radii or diameters. {4} Though this equation appears to show bending and deflection will decrease with diameter to the fourth power, the difference of two close fourth order terms is closer to cubic. Doubling roller diameter will reduce deflection closer to eight times.

$$I_x = \frac{\pi({r_{CO}}^4 - {r_{CI}}^4)}{4} \qquad \{4\}$$

Based on the bending equations, to minimize deflection of live or dead shaft rollers with simple end supports, we have the following options:

- Decrease end load (force per width)
- Decrease roller width
- Increase elastic modulus of roller core
- Increase outer roller diameter
- Decrease inner roller diameter
- Decrease bearing width (if live shaft)

Many of these improvements are determined by the process and not available options to reduce deflection. Roller width and nip loads are set by the product and process needs. Rollers are made from steel (high modulus). Nipping rollers will usually have above average diameter. *Even pursuing all these options will not lead to zero deflection.* Many webs are quite thin relative to deflection and provide minimal benefit to compensate for the inevitable center to edge nip process variations.

Since thin webs and metal rollers are inherently insufficient to alleviate deflection-induce nip variations, rubber coverings are added to one (sometime both) of the nipping rollers. The compliant rubber cover with low modulus and significant thickness (many times thicker than the web) is able to conform around the roller variations, reducing center to edge nip variations.

Past work in compliant roller covers has shown a relationship between rubber covering indentation and the force per width created by the rubber compression. {5} Rubber covers are elastic materials that have a stress-strain relationship, but in nipping rollers this relationship is complicated by the cylinder-cylinder geometry and plane strain limits of rubber across a wide nip.

$$N = \frac{2}{3}\frac{(1-\nu)^2}{(1-2\nu)}\frac{E_0}{(1-\nu^2)}\frac{\sqrt{2r_{eq}}}{t}\delta^{3/2} \qquad \{5\}$$

Equation {5} is based on a cylinder against a plate (base radius is infinity), but can be applied to cylinder-cylinder contact by substituting an equivalent radius. {6}

$$r_{eq} = \frac{r_1 r_2}{r_1 + r_2} \qquad \{6\}$$

Equation {5} shows the force per width created from indentation. In most nipped processes, we know the load, but the indentation is harder to estimate. If equation {5} is reversed, indentation can be found from nip load, roller geometry, and rubber properties. {6}

$$\delta = \left(\frac{3}{2}\frac{(1-2\nu)}{(1-\nu)^2}\frac{(1-\nu^2)}{E_0}\frac{Nt}{\sqrt{2r_{eq}}}\right)^{2/3} \qquad \{7\}$$

These equations require the elastic modulus and Poisson's ratio of the rubber cover, which are rarely known. Thankfully, past empirical work provided an empirical equation to estimate elastic modulus from rubber hardness. {8} This work also found a typical or average Poisson's ratio for various rubbers is 0.46.

$$E_0 = 145.7e^{0.0564H_{SA}} \qquad \{8\}$$

Exploring these relationships provides a clearer picture of the key variables that determine rubber covering indentation. Many times, roller covering hardness is viewed as the only option to increase compliance and indentation. Softer rubbers are more compliant, but an equally important variable is the covering thickness. For example, the compliance of a 55A, 6.25mm covering may be the same as 65A, 12.5mm or 75A, 18.75mm covering (see Figure 3). Thicker rubber coverings will cost more and increase the roller diameter, but the compliance and covering life are greatly improved.

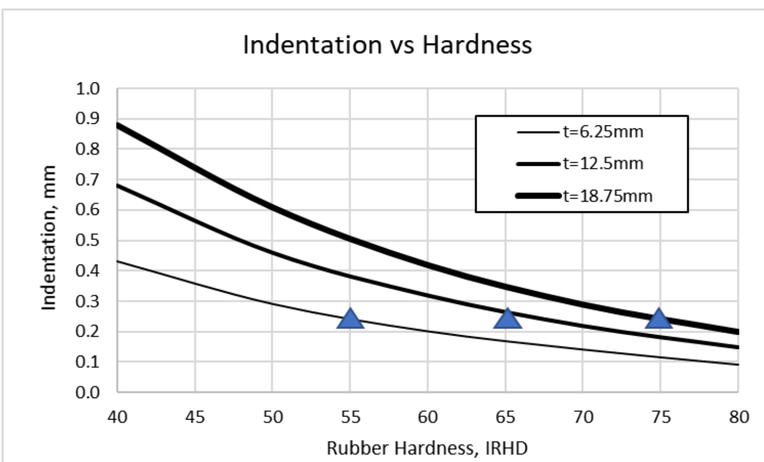


Figure 3 – Rubber Indentation vs. Rubber Hardness and Cover Thickness

Another concern with changing rubber hardness is how average pressure and time in the nip change, possibly changing a process effectiveness. Softer rubbers will have longer contact lengths, whether from hardness or thickness, which will increase time in the nip (see Figure 4) and decrease average pressure (see Figure 5). Whether these time and pressure significantly change a process, such as coating or calendering is unclear.

Many processes view pressure and time as interchangeable variables. For a given load and roller equivalent radius, changing hardness or thickness does not significantly shift the product of time and average pressure. Some processes have been shown more strongly a function of pressure, independent of time, which will make increasing compliance a problematic option.

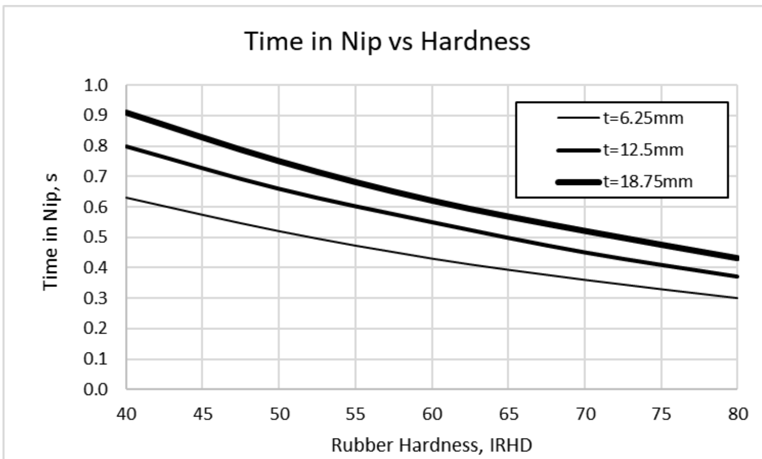


Figure 4 – Time in the Nip Footprint vs. Rubber Hardness and Cover Thickness

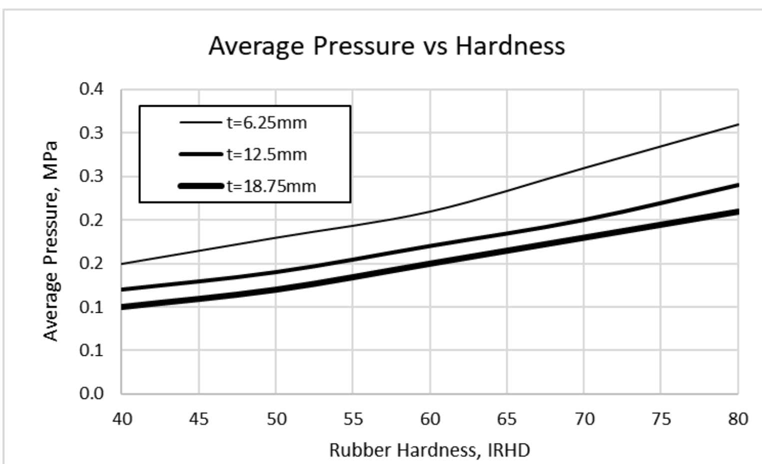


Figure 5 – Average Nip Pressure vs. Rubber Hardness and Cover Thickness

Besides using equation {7} to estimate rubber compliance, a commonly used measurement can also lead to an estimate rubber indentation. Indentation is a function of machine direction footprint length and radius. Equation{9} calculates footprint length for a cylinder against a plate, but can be used for cylinder-cylinder contact if equivalent radius {6} is used.

$$b = \sqrt{8 r_{eq} \delta} \qquad \{9\}$$

Reversing equation {9} allows indentation to be estimated from footprint measurements. {10}

$$\delta = \frac{b^2}{8 r_{eq}} \qquad \{10\}$$

Substituting equations {5} and {9} into equation {1} shows the linear relationship of indentation to average nip pressure. {11} Doubling indentation will double average nip pressure.

$$P_{avg} = \frac{2}{3}\frac{(1-\nu)^2}{(1-2\nu)}\frac{E_0}{(1-\nu^2)}\frac{\sqrt{2 r_{eq}}}{t}\delta \qquad \{11\}$$

**Deflection-Indentation Ratio**

Roller coverings should be designed with sufficient compliance to minimize pressure profile variations from roller deflection. For a given load, equations {2} or {3} estimate deflection and equation {7} estimates rubber indentation. Comparing these two values provides a potential criterion to determine if a rubber covering of a nipped roller is sufficient to perform its intended deflection compensation function. {12}

$$\varphi = \frac{y_{MAX}}{\delta} \qquad \{12\}$$

If the deflection-indentation ratio, φ, is greater than unity, there is potentially no contact or pressure in the center of the nipped process. If this ratio is well below unity, the roller will successfully minimize pressure profile variations. The critical ratio will differ for individual processes, but a target φ of 20-50% provides a high level of confidence for process success.

Recommending a deflection-indentation ratio, φ, is less than one or with a target of 20-50% is helpful, but reasonable goals for each component of the deflection-indentation ratio are also needed. Nipping rollers are often sized to keep deflection around 0.05-0.15mm. Based on this level of expected deflection, rubber indentation targets should be 0.25-0.50mm.

These calculations are based on a simplified view of the problem. This analysis makes the following assumptions:

- All deflection is from the rubber covered nipping roller. The nipping roller contacts a base roller that is considered infinitely stiff without any contribution to deflection or nip variations.
- The nip load profile is uniformly distributed load across the entire roller face width.The nip load does not change due to roller deflection.
- Roller deflection is based on thin walled cylinder, simply supported at the roller ends or in an overhang beam design with two internal supports a distance ‘a’ from the end of the roller.
- Rubber indentation is calculated from average load per width.

Clearly, nipped roller deflection would change the uniform load from a flat profile to a 'smile' profile with more force per width at the nip edges than the center. Ignoring the deflection difference of a uniform load vs a 'smile' profile over-estimates the deflection, making the calculations here more conservative and under-estimating safety factors.

**Example 1 – Reducing deflection-indentation ratio by hardness or radii.** Figure 6 shows a baseline case for the small nipping roller that was used to control a web squeeze coating application. The small diameter nip roller deflection, created significant center to edge coating variations (thicker in the center). To reduce the process variations, design options can be assessed based on improving the deflection-indentation ratio. Reducing the rubber hardness from 80A to 70A would improve the φ from 88% to 60%. Doubling the rubber thickness or core cylinder wall thickness, both increasing the overall radius, would improve φ to 57% or 34%, respectively. Making all three changes – hardness, rubber thickness, core cylinder wall thickness – could lower φ to 15% (but also increase roller diameter by 25%).

One important warning: In seeking a more compliant cover, be cautious for cases that can't increase overall diameter. Increasing cover thickness by reducing the core cylinder will often make φ worse, as the gains of thicker cover compliance are lost due to increases in deflection.

| | | Baseline | <Hardness | <Thickness | >CoreRadii | <All |
|---|---|---|---|---|---|---|
| roller width, mm | $w_R$ | 762 | 762 | 762 | 762 | 762 |
| rubber roller radius, mm | $r_1$ | 51 | 51 | **57** | **57** | **64** |
| rubber core outer radius, mm | $r_{CO}$ | 44 | 44 | 44 | **51** | **51** |
| rubber core inner radius, mm | $r_{CI}$ | 38 | 38 | 38 | **44** | **44** |
| based roller radius, mm | $r_2$ | 305 | 305 | 305 | 305 | 305 |
| core roller elastic modulus, MPa | $E_R$ | 2.07E+05 | 2.07E+05 | 2.07E+05 | 2.07E+05 | 2.07E+05 |
| core roller support inset, mm | a | 0 | 0 | 0 | 0 | 0 |
| rubber cover thickness, mm | $t_R$ | 6.4 | 6.4 | 12.7 | 6.4 | 12.7 |
| rubber Poisson's ratio, -- | $\nu$ | 0.46 | 0.46 | 0.46 | 0.46 | 0.46 |
| rubber hardness, Shore A | IRHD | 80 | **70** | 80 | 80 | **70** |
| nip load force, N | F | 2225 | 2225 | 2225 | 2225 | 2225 |
| nip distributed load, N/m | N | 2917 | 2917 | 2917 | 2917 | 2917 |
| rubber indentation, μm | $\delta$ | 84 | 122 | 130 | 81 | 183 |
| roller deflection, μm | $y_{MAX}$ | 74 | 74 | 74 | 28 | 28 |
| deflection/indentation ratio, -- | φ | **88%** | **60%** | **57%** | **34%** | **15%** |

Figure 6 – Examples of Improving Deflection-Indentation Ratio by Roller Design Changes

## ANTI-DEFLECTION ROLLER DESIGN AND ANALYSIS

The strongest variable to reduce deflection in a standard live or dead shaft roller is increased diameter. Changing from aluminum to steel will reduce deflection by three times. Increase wall thickness will have small benefits. Increasing diameter will be a nearly cubic effect. Figure 7 shows the significant reduction of deflection as a function of increasing roller diameter.

However, in many application, there may be process limitations that prohibit significant diameter increases, especially in upgrading existing equipment, such as a winder, where the web path and space are defined and difficult to modify.

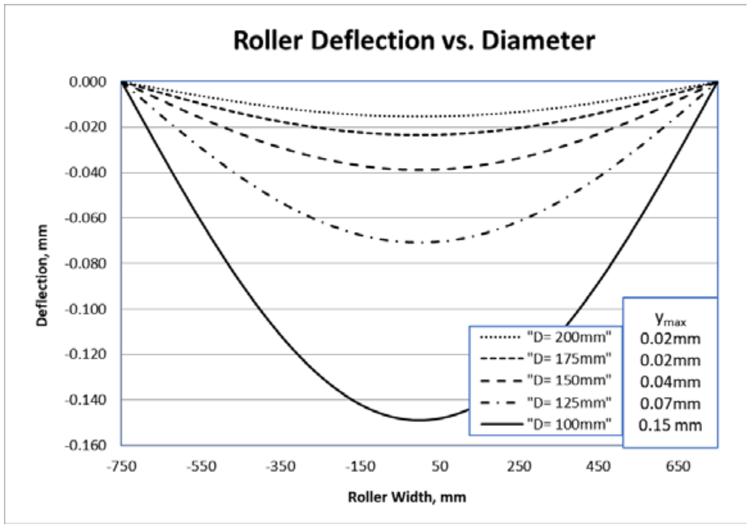


Figure 7 – End-Supported Roller Deflection Profile vs Diameter

Both dead and live shaft roller designs usually imply the shaft or journal connects to the core cylinder at its ends. Just as a table can have the legs anywhere along its length, the support for the core cylinder does not need to be at the roller ends. For assembly purposes, having the core supports at the end of the roller is easiest, but alternate roller designs with different internal structure can greatly reduce deflection under load.

The concept of minimizing beam or roller deflection not new. Supporting a beam from near the 'quarter points' is a first rule of thumb to minimize deflection. Bessel support points minimize beam sag and are centered at 0.56 times the length or 22% in from each end. Minimum sag occurs when the center of the beam sags the same amount as the end points. Many patents on roller design have noted how internal structure minimizes deflection under load. Figure 8 shows two figures from the 1953 patent "Antideflection Roll Assembly" clearly showing an overhang beam roller structure.

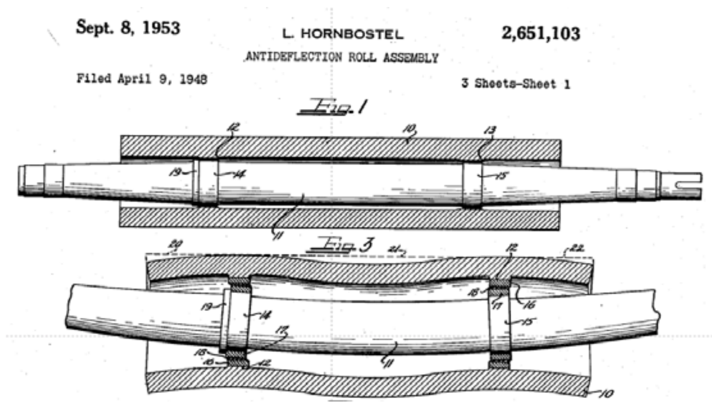


Figure 8 – Figure 1 and 3 from US Patent 2651103

The benefit of moving in the supports of the roller core cylinder is defined by the equation for a uniformly distributed load of a beam with overhanging ends. In addition to the roller and load data, predicting the deflection under load of the overhanging beam roller requires either the distance between the two support point (a distance less than the roller width) or the inset distance that each support point is from the roller end (see Figure 9).

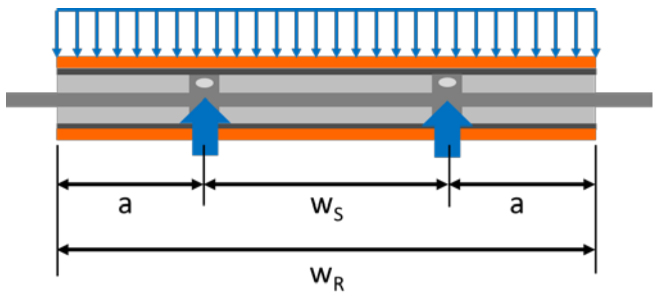


Figure 9 – Roller Shell with Internal Overhang Supports

The deflection vs cross-machine direction position is defined by equation {12} and is a function of load, roller width, support spacing, core cylinder material, and radii. The inset position is either defined by the spacing between the overhang support points or the distance in from the end. {13}

$$y(x) = \frac{\frac{1}{8}Nw_Rw_Sx^2 - \frac{1}{24}Nx^4 - \frac{1}{16}Nw_R{}^2x^2 + \frac{1}{384}Nw_S{}^2(24a^2 - 5w_S{}^2)}{EI_x} \quad \{12\}$$

$$a = \frac{w_R - w_S}{2} \quad \{13\}$$

**Example 2 – Reducing deflection using overhang beam structure.** For this case of a 1.5m width by 150mm diameter roller, moving the end support in only 0.1m reduced the deflection by nearly 50%. The deflection is reduced by over 90% by moving from the supports to the minimum sag points (see Figures 10 and 11).

| | | Baseline | a=0.1m | a=0.2m | a=0.3m | a=0.4m |
|---|---|---|---|---|---|---|
| roller width, mm | $w_R$ | 1500 | 1500 | 1500 | 1500 | 1500 |
| rubber roller radius, mm | $r_1$ | 75 | 75 | 75 | 75 | 75 |
| rubber core outer radius, mm | $r_{CO}$ | 50 | 50 | 50 | 50 | 50 |
| rubber core inner radius, mm | $r_{CI}$ | 38 | 38 | 38 | 38 | 38 |
| based roller radius, mm | $r_2$ | 125 | 125 | 125 | 125 | 125 |
| core roller elastic modulus, MPa | $E_R$ | 2.07E+05 | 2.07E+05 | 2.07E+05 | 2.07E+05 | 2.07E+05 |
| core roller support inset, mm | a | 0 | 0.1 | 0.2 | 0.3 | 0.4 |
| rubber cover thickness, mm | $t_R$ | 25.0 | 25.0 | 25.0 | 25.0 | 25.0 |
| rubber Poisson's ratio, -- | $\nu$ | 0.46 | 0.46 | 0.46 | 0.46 | 0.46 |
| rubber hardness, Shore A | IRHD | 70 | 70 | 70 | 70 | 70 |
| nip load force, N | F | 8800 | 8800 | 8800 | 8800 | 8800 |
| nip distributed load, N/m | N | 5900 | 5900 | 5900 | 5900 | 5900 |
| rubber indentation, µm | $\delta$ | 520 | 520 | 520 | 520 | 520 |
| roller deflection, µm | $y_{MAX}$ | 550 | 300 | 130 | 30 | 140 |
| deflection/indentation ratio, -- | $\varphi$ | **106%** | **58%** | **25%** | **6%** | **27%** |

Figure 10 – Reducing Deflection-Indentation Ratio by Overhang Internal Supports

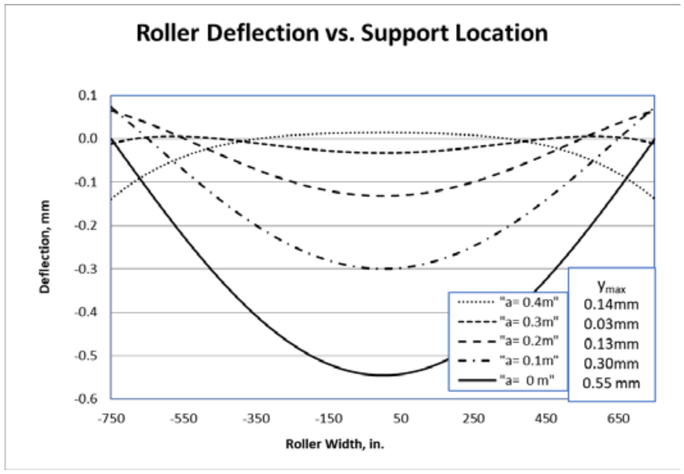


Figure 11 –Deflection Profiles of Overhang Internal Roller Supports

**<u>Example 3 – Reducing core deflection in winding.</u>** Beyond reducing roller deflection, the overhanging beam internal structure can help with winding where wide core deflection leads to wound roller defects. Core deflection become problematic for wide process (e.g. 2m), non-metal paper cores, and cores supported by narrow end chucks. The uneven nip loads of deflecting cores may form soft near-core layers from wrinkling or excess air, which contribute to many roll defects, including telescoping, cinching, cross-buckles, and tin-canning.

Nipped rollers are commonly used in winding to control the air entering the winding roll. If the nipping load deflects the core, the non-uniform nip load can create softness in the near-core layers from wrinkling or allowing too much air into the roll. Core deflection is greatly reduced by switching to large diameter, metal cores, or using a full width high modulus shaft inside the core. Bigger, stiff cores are the obvious solution to many winding problems, but customers and business economics often demand limited core diameter (e.g. 75 or 150mm inner diameter) and single-use or recyclable paper cores. Full width core shafts with a diameter sized to the core inner diameter may create ergonomic problems from their large size and weight.

The overhanging support beam provides an alternative design solution. Core deflection is greatly reduced if the core is supported by a shaft with two chuck at the minimum deflection points. If a small diameter through shaft has two chucks sized for the core inner diameter 22% in from each end, deflection of a paper core can be as low as an end-supported aluminum core of similar dimensions (see Figures 12 and 13).

| | | Baseline | a=0.4m |
|---|---|---|---|
| roller width, mm | $w_R$ | 2000 | 2000 |
| rubber core outer radius, mm | $r_{CO}$ | 88 | 88 |
| rubber core inner radius, mm | $r_{CI}$ | 75 | 75 |
| core roller elastic modulus, MPa | $E_R$ | 3500 | 3500 |
| core roller support inset, mm | a | 0 | 0.4 |
| nip load force, N | F | 700 | 700 |
| nip distributed load, N/m | N | 350 | 350 |
| roller deflection, $\mu$m | $y_{MAX}$ | 980 | 59 |

Figure 12 – Reducing Core Deflection with Overhang Internal Supports

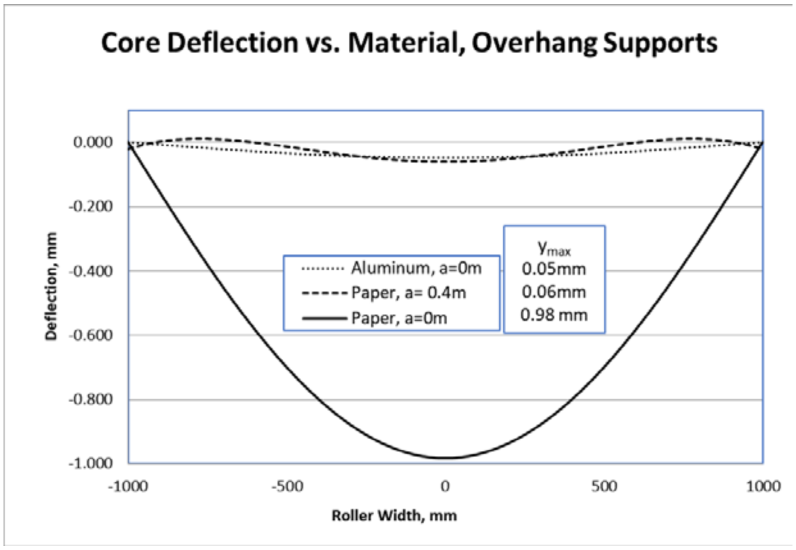


Figure 13 – Core Deflection Profile vs Core Material and Internal Overhang Design

## SUMMARY

Designing the perfect nipped rollers for a process will always be a challenge. As a process requires higher pressure, the obvious changes would be to increase load, decrease roller diameters, and increase rubber hardness. But these changes will all either increase roller deflection or decrease rubber indentation, creating more variations nip load and pressure profile. To avoid nipped roller systems with inherent problems, the deflection-indentation ratio, φ, provides a useful and revealing safety factor for a well-design nipped process.

Parallel, end-supported, high modulus, large diameter, cylindrical rollers will always be the default option for nipped processes, but as widths and loads increase other roller and assembly options must be considered. Proven options to reduce variations in nip load and pressure profile include crowning, crossing roller axis, and backup rollers. However, these options either require more space or must be adjusted for changing nip loads. The overhang beam roller support structure is more complex than an end-supported roller, but greatly reduces deflection, independent of load, and without increasing diameter. The overhanging beam is a proven concept that is widely used in bridges, tables, buildings, and maybe your next nipped roller.

## REFERENCES


1. Hornbostel, L, "Antideflection Roller Assembly," United States Patent 2651103, Sept. 8, 1953.
2. Johnson, K.L., Contact Mechanics, Cambridge University Press, 1989, pp.139-140.
3. Shelton, J.J., "Deflection and Critical Velocity of Rollers," Proceedings of the 3rd International Conference on Web Handling, Oklahoma State University, Stillwater, Oklahoma, June 1995, pp. 398-414.

4. Good, J.K., “Modeling Rubber Covered Nip Rollers in Web Lines,” Proceedings of the 6th International Conference on Web Handling, Oklahoma State University, Stillwater, Oklahoma, June 2001, pp. 159-177.
5. Verterra, Romel, (2017), Strength of Materials, Retrieved from http://www.mathalino.com/reviewer/mechanics-and-strength-of-materials/solution-to-problem-621-double-integration-method.