CLOSED SOLID-STATE-FLUID MECHANICAL MODEL FOR CALCULATING THE TRANSFERABLE TORQUE ON WRAPPED ROLLS

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ABSTRACT

Higher velocities of webs at the same or reduced web tension require basic improvement of the existing machine technique. Concerning the transferable torque between a web and a roll common work is restricted either to the foil bearing theory at constant web tension without solid-state contact, or to the solid-state friction, including in some cases the influence of a constant fluid pressure. In this paper a model for calculating the transferable torque between a roll and a web is introduced which is more conform to reality. In the web mounting and in the web launching area the gap width and the pressure are calculated taking into account the web tension. In the intermediate area, where contact occurs between the rough surfaces of web and roll, it is assumed that the gap width can be substituted by an equivalent gap width between smooth surfaces. The transferable torque is calculated taking into account the local fluid pressure. The calculations show that, in the contact area, the compressibility of the fluid has a significant influence on the pressure profile and, finally, on the transferable torque.

A device reducing the air entrainment by means of the so called „gap throttle effect" is introduced. It needs no seperate energy, works nearly without contact, and has our patent pending on it. The maximum velocity up to which torque is transferable, as well as the transferable torque at a constant velocity, can be increased by this considerably. The effectiveness could be verified at a test rig in our laboratory.

NOMENCLATURE

b	length in axial direction
F	tensile force
h	gap width
M	torque
p	pressure
R	radius
s	grid point distance
T	web tension
u	web deflection
U	velocity
x	circumferential coordinate
α	circumferential angle
η	dynamic viscosity
μ	coefficient of friction
ω	angular velocity

indices

1	mounting area
2	launching area
a	ambient
C	contact
e	east
eq	equivalent
G	sliding
i	grid point index
R	roll, friction
W	wrap, web
w	west

INTRODUCTION

Regarding a roll wrapped by a web we can distinguish two cases (see Figure 1):
1. The roll drives the web, power is transferred from the roll to the web ($M > 0$, $F_1 > F_2$)
2. The web drives the roll, power is transferred from the web to the roll ($M < 0$, $F_2 > F_1$)
The intermediate case, in which no power is transferred ($M = 0$, $F_1 = F_2$), is of little practical interest.
In practice sliding should not occur in the whole wrap angle α_W, but only in part of it, the so called sliding angle α_G. In the remaining part $\alpha_W - \alpha_G$ of the wrap, the web and the roll move with the same velocity. This area where no power is transferred is, in either direction of torque, at the mounting edge of the wrap.
Assuming that the web is perfectly flexible, that the contact pressure between the web and the roll with the radius R is only caused by the tensile force F within the web, and that the sliding friction strain between the two surfaces is proportional to the contact pressure (coefficient of friction μ=const), we can calculate the force ratio and the transferable torque from the Eytelwein equation [1]:

case 1, M > 0 ($F_1 > F_2$):

$$\frac{F_1}{F_2} = e^{\mu \cdot \alpha_G} \;;\; M = +R \cdot (F_1 - F_2) = +R \cdot F_1 \cdot \frac{e^{\mu \cdot \alpha_G} - 1}{e^{\mu \cdot \alpha_G}} \tag{1}$$

case 2, M < 0 ($F_2 > F_1$):

$$\frac{F_2}{F_1} = e^{\mu \cdot \alpha_G} \;;\; M = -R \cdot (F_2 - F_1) = -R \cdot F_2 \cdot \frac{e^{\mu \cdot \alpha_G} - 1}{e^{\mu \cdot \alpha_G}} \tag{2}$$

For both cases we can write:

$$\frac{F_{high}}{F_{low}} = e^{\mu \cdot \alpha_G} \;;\; |M| = R \cdot (F_{high} - F_{low}) = R \cdot F_{high} \cdot \frac{e^{\mu \cdot \alpha_G} - 1}{e^{\mu \cdot \alpha_G}} \tag{3}$$

The maximum absolute value of the torque is reached at $\alpha_W = \alpha_G$, when sliding occurs in the whole wrap angle:

$$|M|_{max} = R \cdot F_{high} \cdot \frac{e^{\mu \cdot \alpha_W} - 1}{e^{\mu \cdot \alpha_W}} \tag{4}$$

That means: in both cases, at given values of μ and α_W, the maximum torque is proportional to the higher web tensile force F_{high}. Thus, if a high torque is demanded, the higher web tensile force should be chosen as high as possible. Therefore, evaluating the result of any arrangement for increasing the transferable torque, it makes sense to regard the higher web tension force as given.
The transferable torque can be smaller than described by eqn. 4 or even vanish for two well-known reasons: one is the influence of centrifugal force on the web. Although it would be rather simple to take this effect into account we will neglect it in the following. The second reason is the entrainment of the ambient fluid, adhering to the moving surfaces, into the gap [2]. In close vicinity to the contact line of the two moving surfaces, the pressure increases, and a bearing force which tends to seperate the two surfaces from each other is generated. In a foil bearing this effect is used to avoid contact between the surfaces. In our case this effect reduces the transferable torque.

EXTENDED EYTELWEIN EQUATION

A rough approximation for taking this effect into account is given in the following.
It is widely accepted [3,4] that in a foil bearing the gap width and the fluid pressure are constant in a great part of the wrap, and that the constant gap width h_0 can be calculated using the formula

$$\frac{h_0}{R} = 0.643 \cdot \left(6 \cdot \frac{\eta \cdot (U_R + U_W)}{T}\right)^{2/3} . \tag{5}$$

The balance between the web tension T, the fluid pressure p and the ambient pressure p_a yields

$$T = (p - p_a) \cdot R. \tag{6}$$

The substitution of T in eqn. 5 by means of eqn. 6 and the assumption $U_R = U_W = U$ lead to the following adequate formula for the pressure:

$$p - p_a = \frac{0.643^{3/2} \cdot 12 \cdot \eta \cdot U \cdot \sqrt{R}}{h_0^{3/2}} = 6.187 \cdot \frac{\eta \cdot U \cdot \sqrt{R}}{h_0^{3/2}} \tag{7}$$

We now assume that eqn. 7 is not only valid for a foil bearing but also for a wrapped roll with contact between the surfaces, that in this case h_0 is the gap width in the contact zone, caused by the surface roughnesses, and that h_0 ist constant. For given values of η, U, R, and h_0 we can then calculate the generated pressure difference $p - p_a$ which is assumed to be constant in the wrap, too. Together with the contact pressure p_C this pressure difference is in equilibrium with the web tension. This leads to:

$$T = (p_C + p - p_a) \cdot R \tag{8}$$

The friction force between the web and the roll is proportional to the contact pressure p_C. Substituting the web tension T = F/b for the tensile force F leads to the following extended form of the Eytelwein equation 3:

$$\frac{T_{high} - (p - p_a) \cdot R}{T_{low} - (p - p_a) \cdot R} = e^{\mu \cdot \alpha_G}$$

With $p - p_a$ from eqn. 7 we finally get the maximum transferable torque per length unit:

$$\left|\frac{M}{b}\right|_{max} = R \cdot \left(T_{high} - 6.187 \cdot \frac{\eta \cdot U}{(h_0/R)^{3/2}}\right) \cdot \frac{e^{\mu \cdot \alpha_W} - 1}{e^{\mu \cdot \alpha_W}} \tag{9}$$

That means a linear decrease of the transferable torque with increasing velocity U. In principle something like this was expected. But some of the assumptions mentioned above seem to be contradictory, for example that the fluid pressure is supposed to be constant over the wrap: we know that there is an area at the end of the wrap where the fluid pressure falls below the ambient pressure. If we assume a constant gap width there should be a pressure drop in circumferential direction. Additionally, in the case of incompressible flow which is mostly assumed in foil bearings, this pressure drop should be linear.

In order to clarify these questions we developed a computer programme taking into account all effects which seem to be important.

BASIS OF THE COMPUTER PROGRAMME

On the one hand the generated pressure profile depends on the shape of the gap. On the other hand the shape of the gap results from the balance of forces exerted on the web surface and thus from the pressure within the gap. That means that the governing equations for the pressure and the deflection are coupled and have to be solved simultaneously. This is valid as well for a foil bearing as for a part of the wrapped roll.

Assumptions

- The ambient fluid is air. The flow in the gap is laminar and inertialess.
- The surfaces of roll and/or web are rough. The gap width can be substituted by an equivalent gap width h_{eq} between smooth surfaces.
- In part of the wrap angle there is contact between the two surfaces. Only there power is transferred. The influence of the shearing strain within the air on the friction force between the two surfaces is negligible.
- In the contact zone the equivalent gap width is constant: $h_{eq} = h_{eq,min} = \text{const.}$
- The web velocities U_{W1} and U_{W2} are nearly equal ($U_{W1} = U_{W2} = U_W$, see figure 2).
- The surface velocities U_W and $U_R = \omega \cdot R$ are equal or nearly equal ($U_W = U_R = U$).
- The web is inertialess.
- Transverse to the direction of velocity the web is infinitely long, i.e. there is no side flow.

Pressure p(x)

The pressure distribution follows the Reynolds equation. Taking compressibility into account it can be written as follows [5]:

$$\frac{d}{dx}\left[p \cdot \left(U \cdot h - \frac{h^3}{12 \cdot \eta} \cdot \frac{dp}{dx}\right)\right] = 0 \qquad \{10\}$$

In a finite formulation we can write for the western and the eastern boundary of one element (see figure 3):

$$\left[p \cdot \left(U \cdot h - \frac{h^3}{12 \cdot \eta} \cdot \frac{dp}{dx}\right)\right]_w = \left[p \cdot \left(U \cdot h - \frac{h^3}{12 \cdot \eta} \cdot \frac{dp}{dx}\right)\right]_e \qquad \{11\}$$

With the approximations

$$\left(\frac{dp}{dx}\right)_w = \frac{p_i - p_{i-1}}{s_x} \text{ and } \left(\frac{dp}{dx}\right)_e = \frac{p_{i+1} - p_i}{s_x} \qquad \{12\}$$

we finally get:

$$p_i = \frac{p_{i-1} \cdot p_w \cdot h_w^3 + p_{i+1} \cdot p_e \cdot h_e^3 + 12 \cdot \eta \cdot U \cdot s_x \cdot (p_w \cdot h_w - p_e \cdot h_e)}{p_w \cdot h_w^3 + p_e \cdot h_e^3} \qquad \{13\}$$

At both boundaries of region 3 (see figure 2), where s_x is larger than in the regions 1, 2, 4, and 5, a modified form of eqn. 13 is used, taking into account different values $s_{x,w}$ and $s_{x,e}$. If the influence of compressibility is neglected the pressures p_w and p_e have to be removed. Then we get:

$$p_i = \frac{p_{i-1} \cdot h_w^3 + p_{i+1} \cdot h_e^3 + 12 \cdot \eta \cdot U \cdot s_x \cdot (h_w - h_e)}{h_w^3 + h_e^3} \qquad \{14\}$$

We use the approximations:

$$h_w = \frac{h_{i-1} + h_i}{2};\ h_e = \frac{h_i + h_{i+1}}{2};\ p_w = \frac{p_{i-1} + p_i}{2};\ p_e = \frac{p_i + p_{i+1}}{2} \qquad \{15\}$$

The pressure boundary conditions are:

$p(x = x_{min} = -l_1) = p(x = x_{max} = \alpha_W \cdot R + l_1) = p_a.$

The length l_1 has to be chosen so that no visible pressure gradient occurs at the boundaries. For given values of velocity U, viscosity η, and grid point distances s_x the pressure distribution can be calculated iteratively by a relaxation method if the gap width h(x) is known.

Gap width h(x) and web deflection u(x)

Within the region 3 contact between the web and the roll is assumed. That means

$$h(l_1 \le x \le \alpha_W \cdot R - l_1) = h_{eq,\,min} \qquad \{16\}$$

We define $h(x) = h_0(x) + u(x)$. In this $h_0(x)$ represents the (theoretical) gap width between smooth surfaces without any deflection. For the different regions we find

region 1: $$h_0(x_{min} \le x \le 0) = R \cdot \left[1 - \sqrt{1 - \left(\frac{x}{R}\right)^2}\right] \qquad \{17\}$$

regions 2, 3, and 4: $$h_0(0 \le x \le R \cdot \alpha_W) = 0 \qquad \{18\}$$

region 5: $$h_0(R \cdot \alpha_W \le x \le x_{max}) = R \cdot \left[1 - \sqrt{1 - \left(\frac{x - R \cdot \alpha_W}{R}\right)^2}\right] \qquad \{19\}$$

These definitions mean that in region 3, where contact is assumed, $u(x) = const = h_{eq,min}$. If there is no contact between web and roll we find the equations for the web deflection u by the equilibrium of forces acting on the web. This is the case at least in parts of regions 1 and 5. We find (see figure 4):

$$p_i \cdot s_x + T_e \cdot \sin\alpha_e = p_a \cdot s_x + T_w \cdot \sin\alpha_w \qquad \{20\}$$

As contact is excluded here (i.e. there is no friction within one element) $T_e = T_w = T_i$. Though that does not mean that the web tension is constant over the whole wrap!

Approximately we can set:

$$\sin\alpha_{w,\,e} \approx \alpha_{w,\,e}$$

$$(p_i - p_a) \cdot s_x = T \cdot (\alpha_w - \alpha_e) \qquad \{20a\}$$

$$\left(\frac{du}{dx}\right)_w = \tan\alpha_w \approx \alpha_w \approx \frac{u_i - u_{i-1}}{s_x} \; ; \quad \left(\frac{du}{dx}\right)_e = \tan\alpha_e \approx \alpha_e \approx \frac{u_{i+1} - u_i}{s_x} \qquad \{21\}$$

$$u_i = \frac{u_{i-1} + u_{i+1}}{2} + \frac{s_x^2}{2} \cdot \frac{p_i - p_a}{T_i} \qquad \{22\}$$

As boundary conditions at $x_{min} = -l_1$ and $x_{max} = \alpha_W \cdot R + l_1$ we define: du/dx = 0.
With $p_i = p_a$ at the boundaries we get:

$u(i=0) = u(i=1)$; $u(i=i_{max}) = u(i=i_{max}-1)$

Deflection also may occur in the regions 2 and 4. As the deflection is here counted from the roll surface, i.e. from a bent surface, we can modify eqn. 21 and write (see figure 5):

$$\alpha_w \approx \tan\alpha_w = \left(\frac{du}{dx}\right)_w \approx \frac{u_i - \left(u_{i-1} - \frac{s_x^2}{2 \cdot R}\right)}{s_x} = \frac{u_i - u_{i-1}}{s_x} + \frac{s_x}{2 \cdot R}$$

$$\alpha_e \approx \tan\alpha_e = \left(\frac{du}{dx}\right)_e \approx \frac{\left(u_{i+1} - \frac{s_x^2}{2 \cdot R}\right) - u_i}{s_x} = \frac{u_{i+1} - u_i}{s_x} - \frac{s_x}{2 \cdot R} \qquad \{23\}$$

This leads to

$$u_i = \frac{u_{i-1} + u_{i+1}}{2} + \frac{s_x^2}{2} \cdot \left(\frac{p_i - p_a}{T_i} - \frac{1}{R}\right) \qquad \{24\}$$

At the boundaries between regions 1 and 2 and between regions 4 and 5 a modified form of eqn. 24 is used with 0.5/R instead of 1/R.
During the calculation process h is not allowed to fall below the minimum equivalent gap width $h_{eq,min}$. If the calculation yields $u_i < h_{eq,min} - h_{0,i}$, we set $u_i = h_{eq,min} - h_{0,i}$.

Web tension T(x)

In region 3 and in parts of the other regions the gap width is equal to $h_{eq,min}$. Here contact may cause a friction force and thus a variation of the web tension.
The forces per length unit acting on a web element with the length s_x are (see figure 6):
- the web tensions T_i and T_{i+1} at the two element boundaries,
- the pressure force caused by the ambient pressure p_a,
- the pressure force caused by the fluid pressure $p = (p_i+p_{i+1})/2$
- the pressure force caused by the contact pressure p_C between the two surfaces, and
- the friction force caused by the contact pressure p_C and by a relative movement of the two surfaces.

The equilibrium of forces in circumferential direction yields:

$$\frac{F_R}{b} + T_{i+1} - T_i = 0$$

The friction force is positive if the surface velocity of the roll is greater than that of the web. That is the case for M > 0 (the roll drives the web).
In radial direction we get

$$\left(\frac{p_i + p_{i+1}}{2} + p_C - p_a\right) \cdot s_x = \frac{s_x}{2 \cdot R} \cdot (T_i + T_{i+1}) \qquad \{25\}$$

The absolute value of the friction force is assumed to be

$$\frac{|F_R|}{b} = \mu \cdot p_C \cdot s_x \qquad \{26\}$$

For the two cases M > 0, M < 0 we can write:

$$\frac{F_R}{b} = \mathrm{sgn}M \cdot \mu \cdot p_C \cdot s_x \qquad \{26a\}$$

This leads to

$$T_i \cdot \left(1 - \mathrm{sgn}M \cdot \frac{\mu \cdot s_x}{2 \cdot R}\right) \qquad \{27\}$$

$$= T_{i+1} \cdot \left(1 + \mathrm{sgn}M \cdot \frac{\mu \cdot s_x}{2 \cdot R}\right) - \mathrm{sgn}M \cdot \mu \cdot s_x \cdot \left(\frac{p_i + p_{i+1}}{2} - p_a\right)$$

Due to the various cases, including the occurence of partial sliding, describing the algorithm completely would lead too far. The following simple case may serve as an example.

The web drives the roll (sgnM = -1). In this case the higher web tension which is assumed to be known is located at the end of the wrap, looking in the direction of velocity. Thus we know $T_{i=imax}$ and can calculate backwards.
If there is no contact ($h_i > h_{eq,min}$) we set $T_i = T_{i+1}$.
If there is contact ($h_i = h_{eq,min}$) we use the following formula which is, in this case, adequate to eqn. 27:

$$T_i = \frac{T_{i+1} \cdot \left(1 - \frac{\mu \cdot s_x}{2 \cdot R}\right) + \mu \cdot s_x \cdot \left(\frac{p_i + p_{i+1}}{2} - p_a\right)}{1 + \frac{\mu \cdot s_x}{2 \cdot R}} \qquad \{27a\}$$

Results

We will not try here to present results of general validity. The chosen data (see figure 7) partially correspond to our test rig.
The simple Eytelwein formula (eqn. 4) yields, in this case:

$$\left|\frac{M}{b}\right|_{max} = 0.3312\frac{Nm}{cm}$$

The question is: How does the velocity U influence the transferable torque?
From the extended Eytelwein formula (eqn. 9) we get:

$$\left|\frac{M}{b}\right|_{max} = \left(0.3312 - 1.285 \cdot 10^{-3} \cdot \frac{U}{m/min}\right)\frac{Nm}{cm}$$

That means: At a speed of about 258 m/min the torque will vanish.
Results for the computed maximum torque are shown in figure 7.
The maximum transferable torque depends slightly on the torque direction (see fig. 7): If the roll drives the web it is higher than in the opposite case. This is valid as well for compressible as for incompressible flow.
The reasons for this result are the assumption of a given T_{high} and the fact that web deflection is beneficial to pressure generation. In the case „roll drives web" the web tension in the mounting area is high ($T_1 = T_{high}$). Compared to the opposite case „web drives roll" with $T_1 < T_{high}$ the maximum hydrodynamic pressure generated in the mounting area is smaller (see figure 8).

Taking into account compressibility leads to a remarkebly lower torque in both cases. The reason for this is the influence of compressibility on the pressure drop in the contact zone (see figure 8). Instead of a linear pressure drop in an incompressible flow we get a pressure which is nearly constant over a wide range of the contact zone.
We proved this surprising result thoroughly and found that, indeed, in a gap of constant width, when the walls are moving with a high velocity, the pressure gradient is, due to compessibility, nearly zero over a wide range. That means compressibility has to be taken into account in such cases.

DEVICE REDUCING THE AIR ENTRAINMENT

Based on theoretical considerations and on observations during tests we found a device reducing the air entrainment. We call it the „gap throttle effect". The basic idea can be explained by means of a simple model, the so called viscosity pump. In the theory of hydrodynamic lubrication this model is often used to explain the generation of hydrodynamic pressure which is higher than the ambient pressure. Figure 9 shows a modified model, generating a pressure below ambient pressure.
Both tapes are moved with the velocity U. The gap width between the tapes (which are here considered as to be inflexible across the length L!) is h. Assuming the roll radiuses are very small we only take in account the length L. A foil with the thickness s < h which is fixed outside the gap is placed in the middle of the gap within a part L_1 of the complete length L. The gap is filled with a fluid of uniform viscosity. Transversely to the direction of movement the walls are infinitely long, i.e. there is no side flow. The flow resulting from the moving tapes and from the generated pressure gradient is assumed to be laminar, inertialess, and incompressible. At both boundaries there is the ambient pressure.
We will only give the result here: The minimum pressure p_{min} occurs at $x = L_1$. For the absolute value of the difference between the ambient and the minimum pressure we get:

$$|p_{min} - p_a| = \frac{\eta \cdot U \cdot L}{h^2} \cdot \frac{1 + \frac{s}{h}}{\frac{1}{6 \cdot \left(1 - \frac{L_1}{L}\right)} + \frac{\left(1 - \frac{s}{h}\right)^3}{24 \cdot \frac{L_1}{L}}} \qquad \{28\}$$

There are two reasons why the foil reduces the air entrainment: The gap width at the leading edge is reduced, and the fluid adheres to the foil walls. Thus even a foil with a thickness s << h has an effect. For the borderline case $s \to 0$ we find:

$$|p_{min} - p_a| = \frac{\eta \cdot U \cdot L}{h^2} \cdot \frac{24 \cdot \frac{L_1}{L} \cdot \left(1 - \frac{L_1}{L}\right)}{1 + 3 \cdot \frac{L_1}{L}} \qquad \{28a\}$$

The maximum here occurs at $L_1/L = 1/3$. For this length ratio we get

$$|p_{min} - p_a|_{max} = \frac{8}{3} \cdot \frac{\eta \cdot U \cdot L}{h^2} \qquad \{28b\}$$

The following example with data similar to those chosen before may give an idea of the

magnitude of order of the gap throttle effect:
data: η = 0.018 mPas; U = 150 m/min; L = 2 cm; h = 20 μm
result (eqn. 28b): $p_{min} - p_a$ = - 60 mbar
This is, compared to a pressure resulting from web tension, e.g.
p(T=4 N/cm, R=17.75 cm) = 22.5 mbar,
a considerable value.

The computer programm was modified so that the effect of a gap throttle foil can be calculated. It was assumed that the gap throttle foil is perfectly smooth and that the roughnesses of web and roll are equal. That means that the gap widths on both sides of the gap throttle foil are equal and that, in the case of contact, they are equal to the half of $h_{eq,min}$.
It should be mentioned: due to the flexibility of the web that was really happens is much more complicated than it could be shown at the example of the viscosity pump. But the results show that the velocity up to which a torque can be transferred (as well as the maximum transferable torque at a given velocity) can be increased distinctly by a gap throttle foil (see fig. 10). The reason for this is the generation of a subatmospheric pressure by the foil (see fig. 11). For optimizing the effect the length of the foil was varied in these calculations from 20 mm at low velocities up to 24 mm at the highest velocity, counted from $x=-l_1=-20$ mm.

For all calculations presented here the minimum equivalent gap width was assumed to be known. The problem to predict a value suitable to the kind and size of roughness will be an object of our further investigations.

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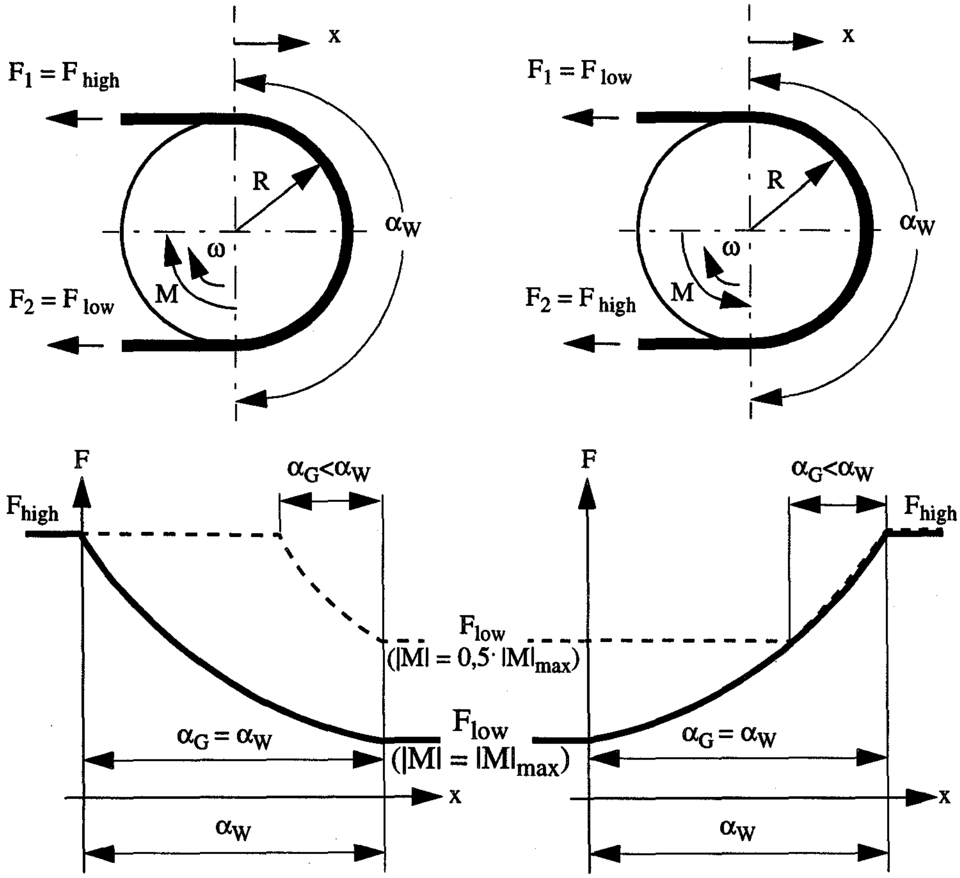

fig. 1 Torque on wrapped rolls

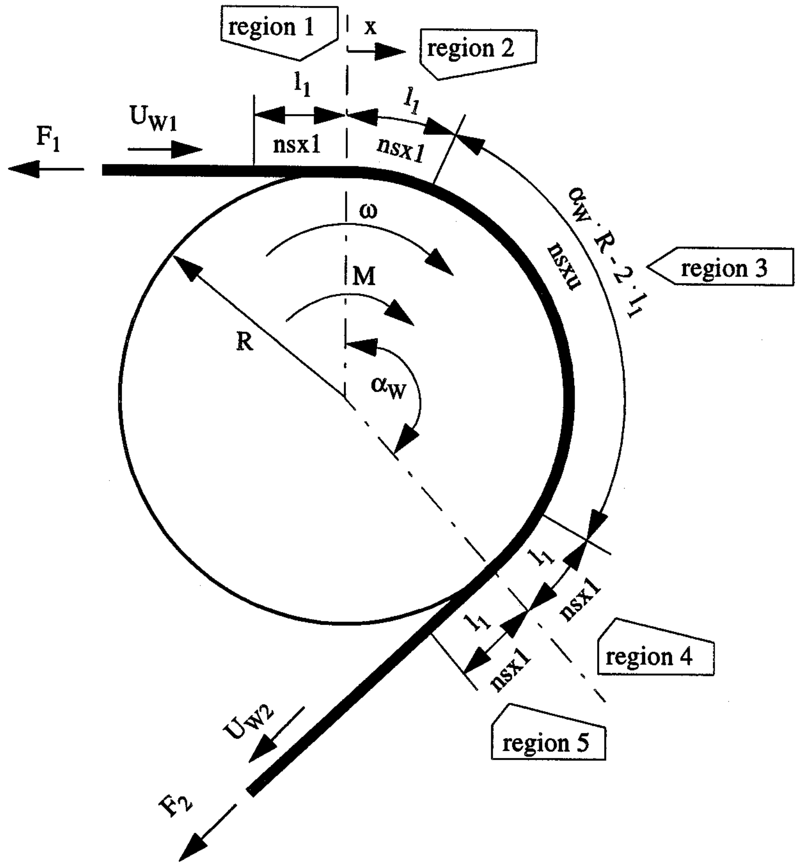

fig. 2 Regions for calculating

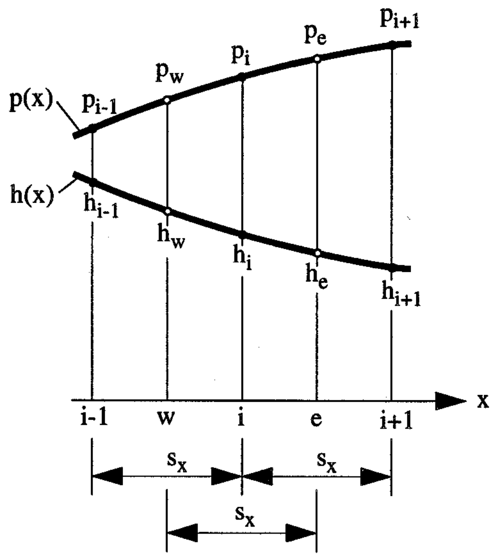

fig. 3 Notation of finite difference method, pressure and gap width

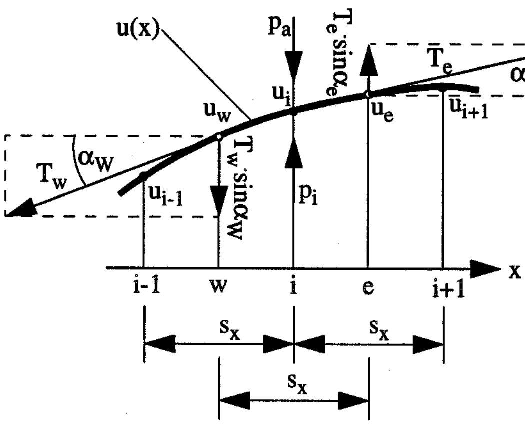

fig. 4 Notation of finite difference method, tension, regions 1 and 5

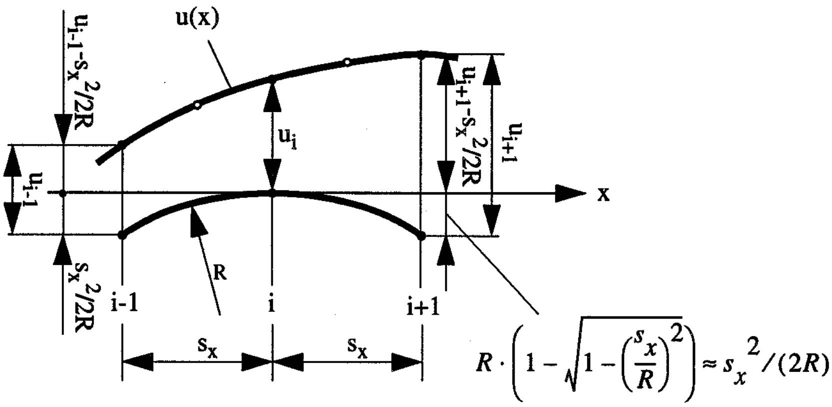

fig. 5 Notation of finite difference method, regions 2 and 4

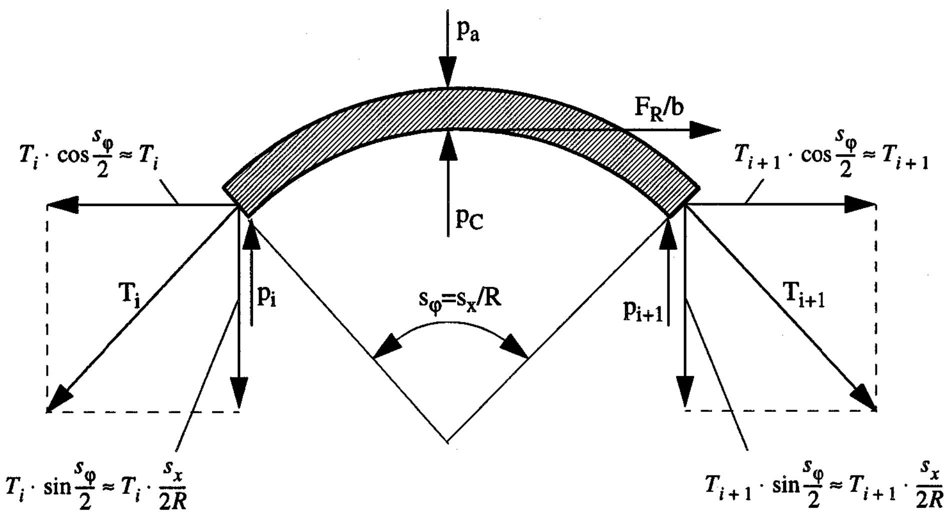

fig. 6 Forces on the web in the contact region

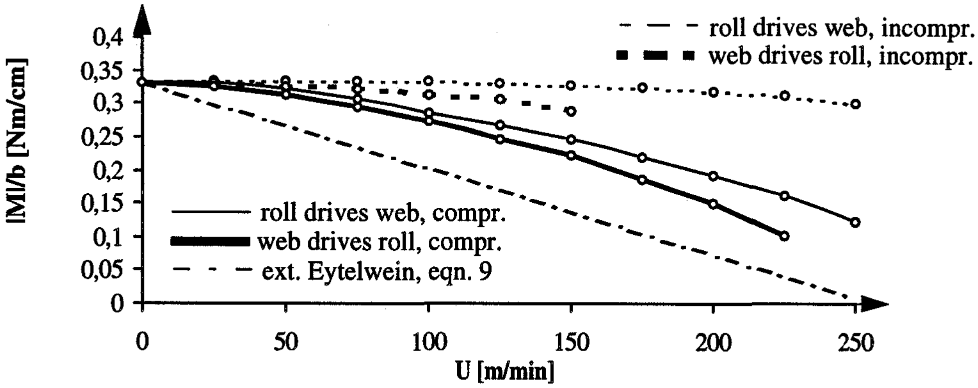

fig. 7 Torque versus web velocity, R = 17.75 cm, T_{high} = 4 N/cm, η = 0.018 mPas, h_{eq} = 20 µm, α_W = 180°, µ = 0.2, l_1 = 20 mm, nsx1 = 20, nsxu = 92

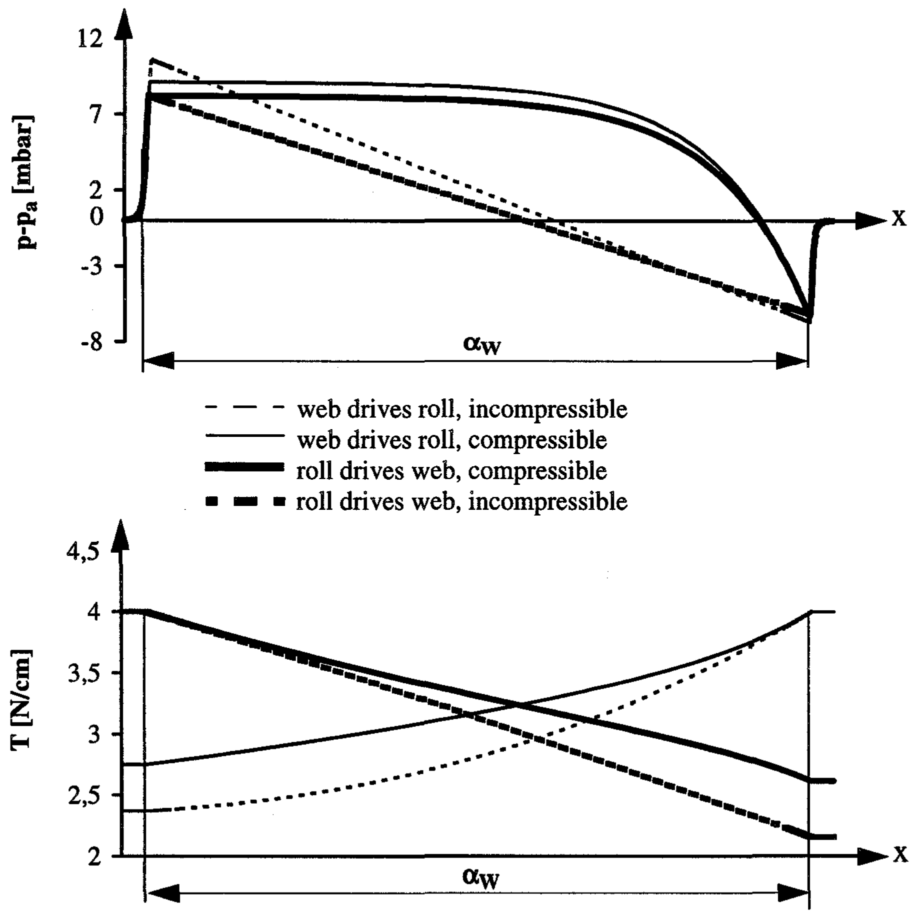

fig. 8 Hydrodynamic pressure and web tension on a roll, data: see figure 7, U = 150 m/min

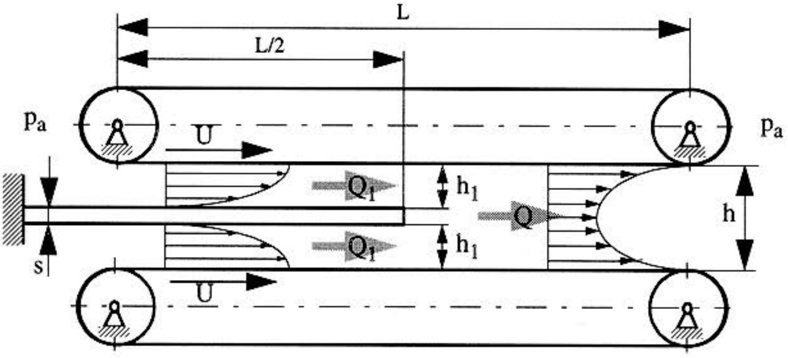

fig. 9 Model of a viscosity pump generating a subambient pressure

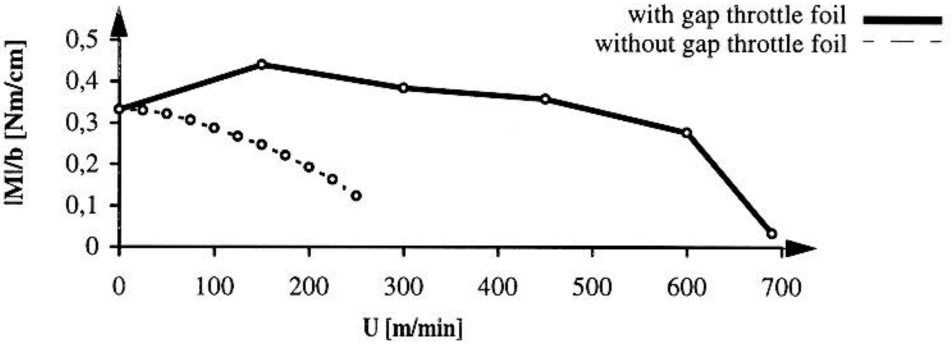

fig. 10 Torque versus web velocity, data: see figure 7, gap throttle foil 20 µm thick

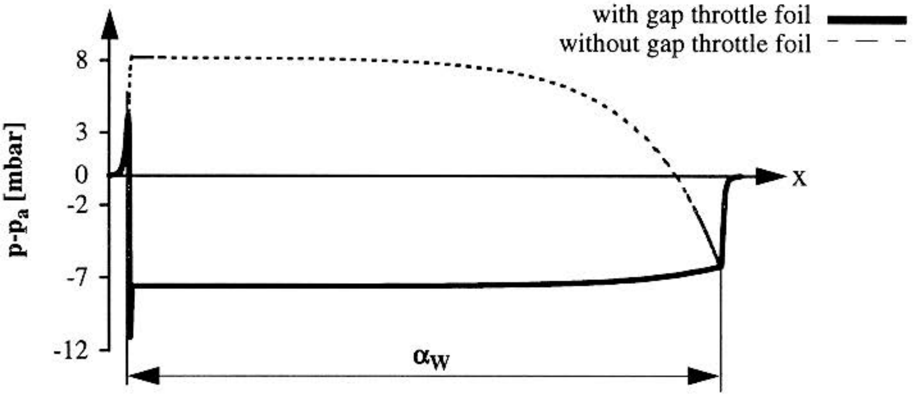

fig. 11 Hydrodynamic pressure, data: see figure 7, U = 150 m/min

D. Schüler, E. G. Welp And O. Kopp
Closed Solid-State-Fluid Mechanical Model for Calculating the Transferable Torque on Wrapped Rolls
6/9/99 Session 4 2:40 - 3:05 p.m.

Question – David Pfeiffer, JDP Innovations Inc.
I have a simple question. Is the Eytelwein equation a German name for something or is it a German name for a person?

Answer – D. Schüler, Ruhr University
Do you ask is he was a German person? I think so. It is a name but I can't tell you when he lived. I have heard a different name for this equation here I think, the capstan or belt equation.

Question – Keith Good, Oklahoma State University
In the earlier discussions we saw that traction, which would relate to transferable torque, would be effected by asperities. In the discussion that I saw just given, it looked like the transferable torque was a function of a viscosity of a fluid, and I saw no interaction of any asperities involved. Would you like to comment on that at all?

Answer – D. Schüler, Ruhr University
I'll try to answer your question. The asperities, the roughness, effect the expression H_C. The velocity at which the transferable torque becomes zero creates a gap with H_C. This gap dictates when contact occurs. It depends on roughness. If we have a higher roughness, this will have a high value. And if we have smooth surface, it will be zero. That is the point which we have to determine the equivalent gap width which is suitable for this case.

Question – Hammond Udalt, Salisbury University
The foil bearing theory is just varied when we have a very small *rub/web* angle?

Answer - D. Schüler , Ruhr University
No, I do not think so. I think that it is not dependent on the *rub/web?* angle. It may be 180 degrees, or more or less, but not less than 5 degrees, but at larger angle it will hold.

DISCUSSION I
6/7/99 Session 1 2:15 - 2:45 p.m.

Jim Dobbs, 3M:
We have had a lot of discussion on the load cell vs. dancer control and one of the things it seems to me is you can make a model of a system but you have to implement it in reality. And if you are going to pick a load cell system and you are going to drive that winding or unwinding roll, you need to match the disturbance. We have not had a lot of discussion about the whole mechanical drive train, the roll, the structure. We had one question that went unanswered as to whether the roll can take the control systems acceleration/deceleration. That is a problem which we observe. We can ask for a control system to do all that but we disregard whether the web or the wound roll will suffer from the control action that is taken.

Keith Good, Oklahoma State University:
I think that's good point and Al Forrest talked a little bit this morning about what some of the shortcomings are of winding models. One of them that he didn't discuss is that we, from a modeling point of view, we are really pretty much in a steady state world. Jan Olsen has done some work in centrifugal winding and that sort of thing but what about roll structure during tremendous accel and decel and that sort of thing? I think that is a definite need from the winding point of view.

PeteWerner, Rockwell Automation:
I am probably going to create a lot of enemies here but relative to the same subject that gentleman brought up is the comparison of force-transducers to dancers. I guess one of the things that I have always tried to point out is that there is really no true comparisons between - -- there is no comparison between dancers and load cell systems. A dancer has two very separate systems: one sets tension in a linear mode and then through a separate optionally used position feedback, we can work into a servo system to keep the dancer at a particular tension – a particular position zone so it's two systems. In fact the dancer is effectively a mechanical to a electrical speed integrator, the speed coming into it and the speed exiting it. However, a tension transducer can only sense tension so when you try to compare the two, it's a relatively inappropriate comparison. What you need to do is compare dancers to other systems that will set tension. For example, program torque, winders and unwinders, etc.... So it's really a somewhat illegitimate comparison and I apologize if I upset anybody with that.

Keith Good, Oklahoma State University:
Would any of the people who spoke about tension control like to speak about the illegitimacy of their comparison?

Jerry Evans, MagPower:
I would agree with the gentleman from Rockwell Automation. The dancer itself is an integrator too so its primarily in a speed control device and you do have to set the tension alone on it and that's one of the big misunderstanding about the people who use dancers. The other point I would like to say, just something about the, and maybe the man from Rockwell could help me here, when it gets to the practical world, of how much you can do to accelerate and decelerate that roll, don't we in the range of 10 horsepower or so start to run out of the torque to inertia ratio of drive motors. What I am saying is that it requires a tremendous amount of torque to change the velocity of the roll and what I am

saying is to do that you need a bigger motor. But you get to a certain point where the motor torque to inertia ration is working against you.

Brian Boulter, Rockwell Automation:
The idea of a paranormal inertia is the time it takes to accelerate the given inertia with rated motor torque. So if you have a paranormal inertia of one second, that tells you that with rated motor torque you can accelerate that inertia in one second so it's a pretty responsive system. If you have a paranormal inertia of 200 seconds like a Yankee dryer that's telling you the motor is very small compared to the inertia loads and eccentricity compensation in that case s not even possible. So the best way to look at it is at a ratio of these paranormal inertias. The paranormal inertia at full roll might be 2 seconds and on an empty roll it might be a second resulting in a 4 to 1 ratio. Maybe you can do eccentricity compensation with that because you will get enough torque out of the motor to handle that kind of paranormal inertia. If you have a paper machine roll that has paranormal inertia of 100 seconds on a full roll compared to one second for the empty core there is no chance of doing eccentricity compensation without destroying something whether it's a coupling, a gearbox or the drive itself. So each application is different. You need to look at the ratio of the paranormal inertia. It's always good to think per unit units. It takes things that are very complicated and makes them simple to deal with. If I tell you a paranormal inertia of a roll was 200 seconds you immediately know that even if it is 500,000 pound feet squared or 50 ounce inch squared, still the motor is very small compared to the load so you just look at the rated motor torque in a paranormal inertia.

Jim Ries, DuPont:
I think in comparing the two systems it depends on where you draw your black box and if you only draw it around the mechanical part, yes they are totally different; they operate and function differently. But most drive manufactures will tell you to take that load cell signal and run it through a PI and what do they do there? They add that integrator. They want that integrator in there. So as a matter of fact you can take the signal out of a tension out of a load cell and do a double integration and it will look exactly the same as a dancer. So it depends on where you draw your box.

Dan Carlson, 3M:
I was just going to clarify that the timed accelerate is one good measure. But if you use the torque to inertia ratio you can compare a line that goes 0 to 100 fpm versus a line that goes 0 to 5000. You guys do this paranormal inertia. I believe that I get a better comparison if I calculate the torque to inertia ratio. Because then I can compare, that takes into account the speed of the line. And so if I wanted to compare the frequency response and two different lines I'd have to adjust for the equipment speed. Do you see what I mean?

Brian Boulter, Rockwell Automation:
Either concept is okay. It's just that the tradition at Reliance has been to use a paranormal inertia. If you are flexible you can use anything. The whole idea is to be able in your mind to get rid of engineering units and just look at the big picture without getting bogged down with these engineering units. However you do it is your choice. We do it with paranormal units.

DISCUSSION II
6/8/99 Session 2/3 2:15 - 3:05 p.m.

Discussion on Slitting:

Discussion:
Dr. Lu stated in his paper in Session 2 that the definition of slitting was controlled crack propagation. I think this may apply to some webs, but in precision slitting hopefully we shear the web without any cracks of propagation on the line due to web factors.

Discussion:
In production we have slit separation problems and overlapping section problems in winding. So proper lateral position control of of each slit web after slitting would solve our problems. The runnability sensitivity to tension after slitting is a quality issue.

Discussion:
If the slitting process causes a burr or anything like that as you go into the coating process it can work the rolls and cause an edge build up and roll defects will result in the winding process. So the slitting problems can cause problems in coaters and winders.

Discussion:
You used the phrase process this morning and sort of isolated it from web management handling. So I just threw the phrase process as a slitting function (lets use that word here). Its probably one of the most common function beyond unwinding and rewinding. Just for sake of interest here how many are doing some kind of slitting function? Look at this it's enormous. In fact there was a survey done a few years ago, I think by *Paper Film and Foil Magazine* that said that slitting was probably the most common function beyond winding and unwinding. So how many of you here have a problem with slitting? So its pretty common; they don't want to talk about it. The process of slitting sometimes is wrongly chosen, people will adopt a process that is the simplest cause they don't understand it well. They'll end up sticking a razor blade in the sheet cause that's simple, or they might use a crusher score slitting system cause that's simple. Anyone who tries shear slitting is confused as it appears to be a complex function and there is a great deal of misunderstanding; when they start to make errors the errors compound and everybody gets frustrated and starts hammering and beating on the equipment. Understanding the basic principles of shear slitting is really worthwhile. That is something that needs to be examined more deeply.

Discussion on Fiber Cores:

Bob Lucas, Beloit Corporation:
Core problems related to differential winding and difficulty with cores.

Differential duplex winders: We as machinery suppliers do not make them and never have. I am moderately familiar with them. Their supplied by a wide suppliers here in north America and elsewhere. But primarily there used for winding small rolls in converting plants. But to explain what a differential duplex winder is in general, if you take a roll of paper or film and you pull it through a center process of a winding application you have a web that passes over a drum and you may slit it into many rolls and these rolls are wound on alternate sides of the drum, where each roll has separate supports. Now there are a lot of winders today for each roll that is wound on a duplex winder, has truly a separate support stand to support each and every roll. But this type of winder I was speaking of all the rolls on one side of the drum are supported on a shaft which may have intermediate support, but they all have a common shaft. Which raises some interesting questions cause you can have rolls of slightly different diameter on different parts of the profile which would generate different rpm's and things of that sort. But the idea here is to supply a center wind torque to be able to handle all these individual rolls. So what in essence what we have is each that each one of these shafts has a solid shaft that is keyed or splinned on which we have paper tubes, cores and maybe some spacer cores; but on the ends of each core there is a disk that is keyed into the shaft. We have a shaft rotating at a higher rpm and the nominal speed of the winding roll. By virtue of the cylinder that provides a pressure or force on this stack of tubes that goes across the entire length of the shaft; we can have the ends of each core acts as a clutch base. So the torque that you can transmit to the individual rolls on this shaft is a function of the thrust force that you have on the end of the shaft. That sounds very simple and straight forward; but life is not so simple. The problem is when you start to wind rolls that are much heavier in weight where now you have to recognize the inner surface or the core also is part of a clutch surface, so it is engaging on the shaft and as the roll of weight increases you have a simple $f=\mu n$ that's part of the roll weight and what ever friction force is existing on the ID of the core and then you end up transmitting more torque than you want. As you get rolls that are progressively larger than 36" in diameter, good roll quality becomes more difficult to achieve.

Now depending on the size of the roll some machinery suppliers have provided means of having needle bearings or similar devices that can create more of an antifriction device so that the cores are riding on bearings rather than riding on the shaft, but that means the shaft is being cut down in stiffness so you end up with some other compromises. But this is a design of winder that has been around for close to, well I've been around for 40 years and it wasn't a new design of winder when I started working with winders. The center wind may be applied with a hydraulic motor or it can be applied with a electric drive. So there are all sorts of creation on the scene depending on who the supplier is. Having said that does somebody have a question they want some help in clarifying? Duane Smith might have some other details to add to this, we can get more involved in details than trying to explain the winder, that is a simple clear explanation. I hope I was reasonably clear!

Mike McAlaer, Macro Engineering:
One of the problems we have is when one of our customer wind on paper cores with too much tension, resulting in radial collapse of the core. Often times the manufactures want to save ever penny and buy a thinner core than what may be needed. One thing I'd like to see down the road is some more research on what is the optimum size of the core versus winding tension, and/or the interface with the airshaft. We have a number of different types of air shafts whose engagement to the core could be categorized as continuous strip, leaf type, lump type and what effect does that have on the core collapsing. These are some issues that might be of interest to someone.

Discussion:
I'd like to bring up a topic, I don't have a lot of information on it; but for a lot of mills that wind paper have problems disposing of core material. Does the industry handle that kind of problem and is there a solution towards the landfill disposal or use for the cores as a fuel?

David Rhodes, Sonoco:
We sell a lot of our fiber cores even in metals industries. After winding at pretty high stresses we have a return program with our customers and we sort through them and reuse them. The paper core are certainly recyclable so if you can get them back to a recycle center they can be repulped and reused. In fact that is where the majority of our furnish for paper.

Karl Reid, Oklahoma State University:
I ask the Sonoco team, we have software that requires certain parameters. We have certain standards of measurement in the industry that are accepted as was discussed in a paper this afternoon. Are we at a position or are we still not there yet in the ability to request of a core manufacturur a certain core design or a certain set of properties?

David Rhodes, Sonoco:
For all the parameters we discussed this morning: radial crush, id stiffness, flat crush, u_c, we can provide that information on any core we produce. We have computer models that we use such that we can predict those values. We can do that at different moisture contents. So that information is available.

Karl Reid, Oklahoma State University:
I was struck by the high sensitivity of the cores to moisture. Has anybody impregnated cores with a material to eliminate the problems of hygroscopic effects or are there other methods being used for that purpose? Are we simply at a point where we design for worst case and accept the fact that the strength of the core will degrade with the change of moisture?

David Rhodes, Sonoco:
For today I'd say that we accept that it is going to change and part of the design requirements for the tubes are that we include a safety factor knowing that the moisture content will reach a certain percentage. Paper is a hygroscopic material and is just the nature of paper, your not going to change that physical property of paper. We do things in our process to minimize hygroscopic effects as much as possible. For certain applications we do have composite tubes that we make that are very stable. But they are typical for reuse application and much to expensive for the majority of applications that paper cores are used for. So we certainly don't have an answer for that today, I wish we did.

Comment:
I think there are web strength chemicals and things that can be impregnated into the tube to decrease moisture effects on core strength. That could adversely effect the recycle ability of those tubes. It is hard to repulp a plastic impregnated core.

Gary Homan, Westvaco:
Considering the comments that were made about recycling cores. Is there such a thing that there is only a certain number of times that a core can be recycled where your going to start increasing the core actually collapsing. I know some of our customers to save money will have a tendency if they are winding in-house, particularly taking a larger parent roll and cutting it up into smaller rolls where they will continue to reuse cores time after time. Are they going to end up hurting themselves after a period of time?

Discussion on Web Mechanics, Dynamics and Tension Control:

John Shelton, Oklahoma State University:
An unusual paper on a subject that we haven't had before was presented by Jerry Brown, in Session 3. Propagation of tension in a slender moving web was his topic. He was talking about tension not instantly being applied from roller A to roller B. But like pulses in air it pulses down there at the velocity of sound and he pointed out that this is usually not anything that you would notice today. But we have to build for the future and I think that was an interesting subject for keeping in mind. Webs, whatever their characteristics may be, have a low velocity of sound and speeds are in excess of 4000 m/min in rewinders in the paper industry.

We had two papers on accumulators. Mine was rather restricted, but something that hasn't been covered in literature and that is the period of acceleration required for converting the kinetic energy of accelerating and decelerating rollers in an accumulator into equations that help us predict the variation in web tension in an accumulator. The other paper was a computer simulation as well as testing on a accumulator. Francoise Ono who has presented other papers here as well as her colleagues, had a paper on centering webs with a crowned roller in a steel mill. Lateral behavior of a steel strip is a tremendous problem when you have such a long span, low strain rates, and an imperfect web. The steel usually has significant camber to it and it is a continual battle to try to keep steel strip on the rollers in the machine.

DISCUSSION III
6/9/99 Session 4 2:15 - 3:05 p.m.

Karl Reid, Oklahoma State University:
We've had a lot of discussion about slitting during this meeting and I'd like for Bob Lucas of Beloit Corporation to lead off and say a few words right now.

Bob Lucas, Beloit Corporation:
I didn't mean to get wound up in the center of this but it seems that everyone is interested in slitting. There was a surprising show of hands as to how many people were slitting as part of their web handling process. There is some uncertainty as to whether slitting should be considered as part of the web handling or is it an adjunct to web handling; Karl asked me to get your opinion on that. How do you view slitting? Do you view that as a disassociated process or function?

First time I thought about it, I was momentarily speechless. I considered slitting part of the process of winding. We can start breaking it down; you know web traction is part of the process, spreading is also part of the process, so I considered it an integral part of a slitter rewinding process. Whether one wants to get tied up with semantics to whether it is something that falls within the category of web handling, I'm not sure.

This was a view, from a discussion I had yesterday, on how we view the web. Up to now I think we've all been thinking the web is a single band as a single strip of material. But the moment we slit it, it becomes subsets of that original web. Those subsets may or may not have the same characteristics as the parent web. That creates problems. How many of you have problems after you slit with the individual ribbons, individual strips and would like to understand how to control those individual strips. That's one of those issues when I do the seminar in slitting its always an extremely interesting topic. That I think is another one of those facets of this whole topic that bears evaluation.

Karl Reid, Oklahoma State University:
Our decision as a Center, and with the counsel of our advisory board, has been to treat slitting in the context of the last problem you talked about. As you slit the web and your rewinding the individual strips for runnability in the rewinding process is a logical area for us to be considering as part of our portfolio of research topics. Slitting, as a process itself, is something that we are engaged in, but its not main stream to our primary activity as web handling center. I would like to thank Tidland for making a nice donation of equipment. Most of the equipment that we had in the laboratories until we were able to purchase a high speed web line was donated by our sponsors. So almost everything you saw in the laboratory came from one of our sponsors organizations including the latest slitting machine that came from Tidland and we're indebted to Fife, 3M, Reliance now Rockwell Automation and Beloit. Many others have donated film and paper that have made it possible for us to be engaged with web handling research. DuPont and Mitsubishi and many others have contributed web. That is really an important part of the strength of our Center, that we have been able to put together nice laboratories. What you saw yesterday is really in transition. We have just moved in there so we really have a lot of work yet to do. We just wanted you to see our facilities in the new location. We know that you'll see a lot of different laboratories when you come back next time. We'd also like to say thank you to the six members of the advisory board that worked with us at the beginning to specify the requirements of the high speed web line. It took only ten years to

get this far. But there were interesting obstacles in the along the way, that we won't go into, but we're appreciative of those people who helped us and stayed with us during that time why we were trying to make it happen.

What did we learn during the conference? Did we meet our objective in our first call for papers and statement of papers presented and we said we wanted to make connections with researchers and practitioners around the world. We had ten countries represented, we had about sixty or seventy companies represented, and we made new friends. I was impressed with the number of new people that were here. We have quit a few people here from the web sponsoring companies and some of those were new as well. But we appreciate all of you coming. I'd like to tell you a couple things I learned especially from the keynoters. How Force taught us to customers do not like to see prunes. I think he means a bad roll. He also told us about something I haven't heard about before and that was Suto stress. Then of course Dave McDonald told us what Suto Stress really is when you have three teenagers. But I have had a good time meeting you; I know Keith and Pat and the rest of our staff have enjoyed working with you. I'd like to see if there is anybody else that has a need to say something, a problem. JDP Innovations has no more time on the program.

Bob Lucas, Beloit Corporation:
Karl you raised a question concerning slitters. Were you intending to ask to inquire to get a vote from us as to whether we thought that was appropriate area of study? Or what?

Karl Reid, Oklahoma State University:
No I don't think so. I think we've had adequate discussion on our board. I think that however, that as we look to 2001 IWEB, or call it IWEB 01, if we make it that far and the computer doesn't die. I think we'll see more papers on slitting than we did this time because its obviously a more important area for each of you. I assume that some of you in your evaluation forms did note yesterday with your show of hands that that is an important area to you.

Keith Good, Oklahoma State University:
We had one paper on wrinkling this time. I think it’s a major problem in the web handling industry. We spent a whole morning on winding, we spent a whole afternoon on tension control, yet, wrinkling results in a lot of our losses. To you potential authors please try to submit some abstracts on wrinkling next time, I would appreciate it. Thank you.

Ron Swanson, 3M:
Something I noticed at the IWEB; there seems to be a big gap between theory and tension control and where we're at in industry. You know most of the web lines, maybe 99% of the control out there is PI control. There's a big gap there…

Karl Reid, Oklahoma State University:
Between what we do and what we're talking about.

Ron Swanson, 3M:
Yes, and I'm not sure exactly how to bridge that gap.

Karl Reid, Oklahoma State University:
I think that's a good comment to lead us in the future with and we need to find ways of bridging the gap, filling the gap. I think in the case of OSU researchers, a new web line will make a difference for us in that regard.

Bob Lucas, Beloit Corporation:
Where do you draw the line between a research report and a commercial to the point of view that the people talk about the equipment that they have up and running and working and what not. There's a question that has to be addressed here, cause it is raised to understand what's happening, but how do you do that without a presentation appearing like a commercial?

Karl Reid, Oklahoma State University:
I'll let Keith answer that. But let me say one comment up front, that in general in conference's like this there are concerns going in about having commercialization become too much a part; and at the same time we look for fundamental value and look for things that will be of value to you.

Keith Good, Oklahoma State University:
I have been at conferences before where someone has a new product, this relates to web handling. They pronounce that their product is the best. You get very little or no indication why! They just tell you it's the best. As I review abstracts for this conference I search for commercialism issues. All of the authors will tell you that the letters of acceptance contained a clause on commercialism. I specified that this conference is not a platform for commercialism. There is nothing wrong with presenting new hardware or even software that can solve a web handling problem, but if presented at this conference there had better be sound engineering analysis and testing that proves that in fact the hardware or software can solve the problem. We reserve the right to disqualify any abstract or paper based upon commercialism issues.