Refiner tram and its contribution to pulping and pulp movement in high consistency refiners

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Abstract: Lubrication theory considering the pulp a lubricant between the rotor and stator can explain how the closing force is supported. It requires a small angle between the two discs or an appropriate shape of the refiner bars. Calculating the closing force (P) and dividing by the shear force (F) gives a non-dimensional ratio that shows good agreement with measured values from the refiner motor load and hydraulic pressure while accounting for the steam pressure. The exact value of the pulp's apparent viscosity is not needed.

The major priority of engineers who design refiners is to create a machine that can transmit 20 to 30 megawatts of power to fast-flowing wood chips or pulp safely and in precise control. On the other hand, pulp processing engineers are concerned with the control parameters, such as case pressures, feed rates and chip conditions, that lead to consistencies, temperatures and above all, useful pulp quality at acceptable energy use rates. Although a number of authors have created models for the radial flow of chips and pulp through the refiner, relatively few have looked at the instantaneous relation between the local machine geometry and pulp interaction. May and Miles considered the pulp to be held on the stator while the rotor bars impart the energy to the pulp. A uniform value for viscosity had to be assumed [1,2]. Allison et al. developed more dynamic models using gray box theory to account for measured values [3]. Temperature measurements by several authors indicate steam pressure develops in the refining zone and the integrated value balances a portion of the hydraulic closing pressure [4].

By considering the refiner to be a thrust bearing, with the pulp-steam-water mixture acting as the lubricant, it is possible to apply lubrication theory to explain several control parameter relationships observed by operators. It also allows some insight into plate design and refiner geometry. In this analysis, two models are considered: one assumes the rotor and stator discs remain perfectly parallel but allows for the rounding of the bar leading edges; the second has the rotor and stator not parallel or out-of-tram by a small angle. In TMP refiners the plates are ground with a small taper which is essentially incorporated in the tram angle used here. Originally this ground taper was to compensate for disc turn-back at operating rpm. Due to flexing of the rotor disc, the point of smallest gap can move around the rotor disc so that the plates tend to wear evenly. This author developed a lubrication model based on the rounding of bar leading edges that accounted for the rise in closing pressure needed as the plates wore in use [5,6]. A key measure is the pressure force (P) closing the gap divided by the shear force (F) generated by the motor turning the disc. However, further improvement of this model shows new plates running parallel cannot generate the measured dimensionless P/F ratio, but a very small tram angle between the discs can. For several different sizes of pressurized refiners, this ratio was shown to be in the range of 2 to 5 after subtracting the steam pressure contribution.

Theoretical

The Navier-Stokes equations are written in cylindrical coordinates with the center located at the center of rotation of the refiner. The disc rotation is characterized by \( V_\theta \) while \( V_z \) gives the pulp velocity in the radial direction. The movement of pulp in the z or axial direction is negligible. In a pump, the \( V_\theta^2/r \) creates the pressure gradient dP/dr from the radial momentum equation. It is assumed in most formulations that this is the source of pressure to balance the net closing force after any steam pressure is accounted for. However, if the pulp is mostly on the stator, and therefore moving at a much lower tangential velocity, the contribution to pressure is much smaller. Also, it is worth noting that Kingsbury thrust bearings rely on pivoted plates that provide small pinch or squeeze angles that slide over the lubricant. The centrifugal force therefore does not provide sufficient pressure gradient to support the thrust load, but lubrication theory does.

The average gap is given by \( h \). In the tangential equation, the inertial terms are given by:

\[
\rho V_\theta \frac{\partial V_\theta}{\partial \theta} + \rho V_z \frac{\partial V_z}{\partial r} = \frac{\mu}{h^2} \left[ \frac{\partial^2 V_z}{\partial r^2} + \frac{1}{r} \frac{\partial V_z}{\partial r} \right]
\]

and the viscous terms by:

\[
\frac{\partial}{\partial z} \left[ \mu \left( \frac{\partial V_z}{\partial z} + \frac{2\theta}{2z} \text{ smaller terms} \right) \right]
\]

Dividing the inertial terms by the viscous terms in dimensionless form gives

\[
\frac{\rho V_z^2}{R \theta} + \frac{\rho V_z}{R} \frac{V_z}{\mu} \frac{1}{h^2}
\]
where the terms are now scaled by nominal values. If inertial terms are small enough to be neglected when compared to viscous terms, the pressure gradient is balanced by the viscous terms. For this to be true, the reduced Reynolds number \( \text{Re}^* \) must be small. This occurs when:

\[
\text{Re}^* \sim \frac{\rho (h/R)}{H} \left( \frac{V_n}{V_r} \right) R \theta \ll 1
\]

For a typical refiner, \( V_n = 150 \text{ m/s} \) for the tram angle model or \( R = 3 \text{ mm} \) for the bar shape model. The local gap separation \( h = 0.5 \text{ mm} \). \( \mu = 1 \text{ Pa.s} \). This value for viscosity is about 10 times that of lubricating oil and is taken from values for highly sheared (fluidized) flows of semi solids. Using these values, the reduced Reynolds number is on the order of 0.06 or less, so that the lubrication theory can be used for either case developed below. The ratio of thrust to drag force \( (P/F) \) can now be derived. The boundary conditions for the tangential direction \( r \) and the distance into the gap, \( z \), for tangential velocity \( v \) and the pressure \( P_m \) are given below.

When:

\[
\begin{align*}
 z = 0 & \quad v_0 = V_0 \quad r_0 < R \pi \quad P_m = P_0 \\
 z = h & \quad v_0 = 0 \quad r_0 > 0 \quad P_m = P_0
\end{align*}
\]

For both the tram angle and bar geometry models:

- Continuity is satisfied by the criterion that the flow across any section be constant, i.e.,

\[
Q = \int_0^{h/2} v_0 dz = \text{constant}
\]

The tangential momentum equation becomes:

\[
\frac{dP_m}{d\theta} = \frac{\mu d^2 v_0}{d^2 z}
\]

and a solution for velocity \( v_0 \) is:

\[
v_0 = V_0 (1 - z^2 / h^2) - (h^2 / 2 \mu) \int_0^r \frac{dP_m}{d\theta} (z / h)(1 - z^2 / h)
\]

Substituting into equation 1 gives:

\[
Q = \frac{h V_o}{2 - h^2 / 12 \mu (1/r)} \frac{dP_m}{d\theta}
\]

Pressure is given by:

\[
P_m(r, \theta) = 6 \rho V_0 \int_0^{r^2_\mu} (1 / h^2) \frac{dP_m}{d\theta} - 12 \mu Q \int_0^{r^2_\mu} (1 / h^3) \frac{dP_m}{d\theta}
\]

Setting \( P_0 = 0 \) (pulp is not under pressure when on the "opening side" of the out-of-tram refining zone nor when the refiner bar is over a groove) and using the boundary conditions shown earlier gives:

\[
Q = V_0 \int_0^{r^2_\mu} (1 / h^2) \frac{dP_m}{d\theta} \int_0^{r^2_\mu} (1 / h^3) \frac{dP_m}{d\theta}
\]

Inserting \( Q \) into equation 5, we obtain:

\[
P_m(r, \theta) = 6 \rho V_0 b_1 - 12 \mu Q b_2
\]

Where:

\[
b_1 = \int_0^{r^2_\mu} (1 / h^2) \frac{dP_m}{d\theta} \quad b_2 = \int_0^{r^2_\mu} (1 / h^3) \frac{dP_m}{d\theta}
\]

Here \( h \) is a function of \( r_0 \) and represents the disc or bar surface contour. Two cases were considered:

- **Case 1:** The discs operate at a small tram angle and the bar profiles are considered second order terms.

For case 1, the refining zone is defined by \( h = h_0 + R \tan \alpha - r \tan \theta \cos \phi \) where \( h_0 \) is the minimum gap and the geometry is shown in Fig. 1. For simplicity let \( h = c + d \cos \theta \).

When integrated using tables of integrals that lead to confluent hypergeometric functions, it was easier to use computational methods to determine the value for \( P \).

Using temperature measurements the saturated steam pressure contribution to the closing force was subtracted before calculating the \( P/F \) ratio.

The shear force is given by:

\[
F = \mu V_0 \int_0^{r^2_\mu} \right( \frac{dP_m}{d\theta} / h - 3 H / h^2 \right) d\theta
\]

\[
F = \mu V_0 \left( 4 \right) Y_3 + 3 H Y_1 [Y_2 - Y_3]
\]

Numeric values are inserted into the formulae and the following results produced.

- **Figure 2** shows the results for a minimum gap of 0.05 mm ranging up to a maximum gap of 0.35 mm. When the minimum gap equals the maximum gap, \( P/F \) is zero because the tram angle is zero.

Using temperature measurements the saturated steam pressure contribution to the closing force was subtracted before calculating the \( p/f \) ratio.

The pressure is determined from equation 5, using equations 9 and 10 above.

Integrating the pressure for the zone from \( \theta = 0 \) to \( \pi \) where the pulp supports the thrust load gives the total thrust force \( P \) on the pulp. Although the integration of \( b_1 \) and \( b_2 \) can be accomplished using tables of integrals that lead to confluent hypergeometric functions, it was easier to use computational methods to determine the value for \( P \).

**Case 2:** The discs operate at zero tram angle between them and the bar profiles are considered for new and worn leading edges.

- For case 1, the refining zone is defined by \( h = h_0 + R \tan \alpha - r \tan \theta \cos \phi \) where \( h_0 \) is the minimum gap and the geometry is shown in Fig. 1. For simplicity let \( h = c + d \cos \theta \).

When integrated using tables of integrals that lead to confluent hypergeometric functions, it was easier to use computational methods to determine the value for \( P \).

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profile in the outer refining zone is usually fixed so that \( r \) may be held constant.

Figure 5 graphs the \( P/F \) ratio for parallel plates with new, sharp or worn bar leading edges.

A transformation of \( s = \exp(-\beta r) \) and conversion of the integration limits changes the first integral to:

\[
b_2 = -\beta \int_1^5 \frac{1}{s(a + Bs)^2} \, ds \tag{15}
\]

The solution is given by:

\[
b_1 = -\frac{1}{\beta a^2} \left[ -\frac{a}{a + Bs} - \ln \frac{(a + Bs)}{s} \right] \tag{16}
\]

\[
b_2 = -\frac{1}{\beta a^2} \left[ \frac{a^2 + aBs + B^2s^2}{2(a + Bs)^2} \right. + \ln s + \ln(a + Bs) \right] \tag{17}
\]

where \( s \) is now replaced by \( \exp(-\beta r) \); \( \ln \) is the natural logarithm.

Using these integrals, the pressure \( P_m \) can be calculated and the total thrust is given by:

\[
P = \int_0^1 P_m(r\theta) \, d\theta \tag{18}
\]

Where \( l \) is the bar width.

The shear force is calculated for this case as above and the following results are found.

Note that the new plates cannot develop sufficient \( P/F \) ratio even at a minimal gap while the severely worn profile can develop the needed \( P/F \) ratio at relatively small gaps.

**DISCUSSION**

There is much speculation on the condition of the pulp as it passes through the refiner. Cinematic studies generally suggest that the pulp moves in flocs rather than a continuum. This analysis assumes only that the \( P/F \) ratio averaged over all the flocs at any instant is attributable to the bearing model and does not depend on a centrifugal force to create a hydrostatic pressure.

The theory developed above can be confirmed by three refiner measurements. The motor load gives the shear force \( F \). The closing pressure \( P \) is determined from the gap closing mechanism that is either hydraulic or by using strain gauges if mechanical. The gap \( h \) must be measured in real time. This author originally made gap measurements using a system consisting of a Hall effect gap probe feeding a high-speed (100 kHz) analog recorder that showed the gap varied from 0.1 to 0.9 mm with each revolution. Current digital equipment must have a sample frequency of 3000 Hz to capture this variation on an 1800 rpm refiner.
Temperature measurements are generally used to calculate the local steam pressure, assuming saturation conditions. However, the frictional effect of shearing the pulp may contribute to this temperature, so that the steam, at lower temperature, may pass around the pulp flocs at much higher velocities than the radial velocity of the pulp itself.

The tram angle model allows pulp radial velocities to be much lower than required for the parallel disc model and provides a greater area for steam escape as well.

**CONCLUSIONS**

For new plates at startup, a small tram angle is needed to develop the measured P/F ratios. Only if the plates wear severely can the tram angle be reduced, although the radial velocities will be higher than predicted or measured. Thus the refiner is stabilized by the slightly out-of-tram condition and refiner designers should allow this to occur, but without instabilities such as oscillations or ringing of the disc. This could make the feeding of chips and pulp more uniform, to reduce motor load variation that is significant, on the order of 40-50%, when viewed in real time with high resolution.

By carefully monitoring the tram angle versus pulp properties, an optimal tram angle could be found that preserves fiber length but does not increase the shive level.

**LITERATURE**


Résumé: La théorie en matière de lubrification qui considère que la pâte est un lubrifiant entre le rotor et le stator peut expliquer comment la force de fermeture est soutenue. Elle exige que l’angle entre les deux disques soit petit et que les barres du raffineur soient de forme appropriée. Calculer la force de fermeture (P) et la diviser par la force de cisaillement (F) donne un rapport non dimensionnel en bonne corrélation avec les valeurs mesurées de la charge du moteur du raffineur et de la pression hydraulique, tout en tenant compte de la pression de vapeur. La valeur exacte de la viscosité apparente de la pâte n’est pas nécessaire.


Keywords: REFINERS, MACHINE DESIGN, TRAMMING, LUBRICATION, PULPS.