ABSTRACT

Under typical recovery boiler sootblower operating conditions, the emerging steam jet is severely under expanded, i.e. the nozzle exit pressure is substantially greater than that of the surrounding atmosphere. This results in a shock wave being set up downstream of the nozzle which deprives the jet of valuable mechanical energy for deposit removal. New nozzle designs provide full expansion under current operating conditions in recovery boilers.

Energy delivered by jets emerging from full expansion nozzles are substantially greater than those obtained from conventional nozzles. Alternately, it is possible to achieve energy levels comparable to what can be achieved by a conventional nozzle while consuming less steam. The latter feature of the full expansion nozzle has been utilized in two recovery boilers leading to steam savings of $30 to $40k/month.

1. INTRODUCTION

The burning of black liquor in a recovery furnace is one of many stages associated with the regeneration of spent pulping chemicals. By burning "black liquor" a by product of pulping, "green liquor" and steam are produced. Since pulp is the valued product, economical advantage from chemical recovery exceeds that of the energy produced. As a fuel, black liquor is the sixth most important fuel in industry. However, it has a low heating value when compared to conventional fuel. Typically this value is around 14MJ/kg (6000 Btu/lb) of dry solids and, the corresponding steam production is about 3 to 4 t steam/t dry solids (3 to 4 lb steam/lb dry solids) [1]. The steam produced from the boiler is used as both process steam, as well as for the generation of electricity. Most mills today have to supplement their steam and power requirements by burning alternate fuel in a power boiler as well as purchasing power from the grid.

As a result of burning liquor, a significant quantity of ash is released. A significant fraction of this ash is deposited in the convective pass while the remainder is captured in the hoppers and the electro static precipitator. The accumulation of fireside deposits in a recovery boiler drastically reduces its heat transfer efficiency. This results in higher flue gas temperatures which accelerate fouling, leading to plugging of the downstream flue gas passages [2]. As a consequence of the plugging, the boiler has to be shut down and water washed [3]. The unscheduled shutdown of a boiler interrupts production and incurs severe financial costs to the mill. In order to maintain the uninterrupted operation of the boiler, it is vital to operate sootblowers on a continuous basis to prevent excessive fouling. Removal of deposits depend to a great extent on the type of deposit and the ability of the sootblower to transmit forces to the deposit, that exceed its mechanical strength.

For a given boiler, the deposit strength varies from location to location in the boiler and is directly related to the flue gas temperature [2,4]. In the superheater, deposits primarily consist of carryover type particles that for most part consist of smelt and/or partially burned black liquor particles. On the other hand, in the generating bank and economizer the type of deposit consists primarily of fume which is formed by condensed volatile material. This type of deposit is very light and easily removed. Typically, sootblowers in a recovery boiler use high pressure superheated steam as a cleaning medium. This steam is delivered to nozzles at the end of a lance tube as shown in Figure 1. The steam expands through the nozzle to produce a high speed jet which is then directed at the deposits in order to remove them from the boiler tubes.

Figure 1: Schematic of a Sootblower Jet

Recent studies on sootblower optimization have shown that for typical sootblower operating pressures in a recovery boiler, the steam reaches the speed of sound at the nozzle throat while the jet emerging from the nozzles is severely underexpanded, i.e. the steam pressure at the nozzle exit \( p_e \) is substantially greater than that of the surrounding atmosphere \( p_\infty \)[5]. This results in a normal shock wave being setup just downstream of the nozzle which deprives the jet of valuable energy for deposit removal. This drawback can be overcome by extending the current nozzle to provide sufficient expansion so that the nozzle exit pressure is in equilibrium with the surrounding atmosphere.
However, physical dimensions of the boiler wall openings and sootblower lanes limit the length to which a nozzle can be extended.

This limitation has been overcome through a novel design in the Contoured Full Expansion or "CFE" nozzle. The wall of this nozzle is uniquely contoured to rapidly expand the gas beyond the throat followed by a second contour that redirects the gas towards the nozzle axis. A cross sectional view through the lance tip for both a conventional (High Impact) and new nozzle of fixed throat diameter is shown here in Figure 2. In order to fit the new nozzle in a conventional lance tube, the nozzles are offset by the helix of the sootblower.

![Figure 2: Illustration of a Lance Tip Fitted with a Conventional and Full Expansion Nozzle](Image)

Experience has shown that the cleaning potential for a sootblower correlates with its jet Peak Impact Pressure "PIP"). This is the centerline stagnation pressure for the jet and is measured by a pitot tube at various distances downstream of the nozzle exit. Through dimensional analysis for the flow in the lance, nozzle and jet, PIP can be summarized as a function of,

\[
PIP = F \left(\frac{x}{r_e}, \frac{M_e}{\rho_e}, \frac{p_e}{\rho_e \Delta p}, \frac{d}{l} \right)
\]

where the dimension d & l are indicated here in Figure 3.

![Figure 3: Characteristic Lengths for Modelling the Flow at the Lance Tip](Image)

It has been estimated that the North American kraft pulp industry could save about $100 million/year in energy costs by making modest improvements to the thermal efficiency of recovery furnaces. One of the main tools used in maintaining this efficiency is the sootblower. Sootblower steam consumption in a recovery boiler generally ranges from 6 to 10% of the steam produced. Past attempts to reduce this quantity while maintaining thermal efficiency has been associated with elaborate control strategies. This has been tied to monitoring and maintaining the heat transfer rates of the various convective sections of the boiler. Recent developments in sootblower nozzle technology [6,7] has shown that steam consumption can be reduced with no significant impact on the thermal efficiency of the boiler[8].

2 NOZZLE PERFORMANCE

A comparison of the PIP response for the quarter scale conventional as well as for the new full and quarter scale nozzles as a function of lance pressure is shown in Figure 4.
The above result also shows that for a conventional nozzle, an increase in lance pressure produces a PIP that eventually levels off at about 303kPa (44 psig). In fact for this nozzle increasing the lance pressure by 17%, from 2068 kPa (300 psig) to 2413 kPa (350 psig) only results in a 10% change of PIP. Alternatively for the new nozzle, a similar increase in lance pressure means a 37% increase in PIP. Therefore, for a conventional nozzle, increasing the blowing pressure does not significantly improve the PIP for the jet, while for the new nozzle, the increase in PIP is substantial. The deficiency at high operating pressures seen in a conventional nozzle has been shown to be attributed to the normal shock wave that is setup downstream of the nozzle exit [5]. Increasing the blowing pressure results in an increase in the mass flow through the nozzle as well as the degree of underexpansion \( \frac{p_e}{p_{\infty}} \gg 1 \). The latter causes the shock wave to grow in size and strength resulting in diminished returns. Hence, it can be concluded that during severe plugging of the boiler, increasing the blowing pressure for a conventional nozzle will not improve it’s cleaning potential.

The PIP advantage from the new nozzle over that of a conventional nozzle is realized, for blowing pressure in excess of about 1379 kPa (200 psig) as indicated in Figure 4. For a conventional nozzle operating at 2413 kPa (350 psig) the PIP delivered at a location 76 cm (30”) from the nozzle is about 303 kPa (44 psig). To deliver this amount of PIP with the new nozzle the required blowing pressure is only about 1724 kPa (250 psig). This amounts to a 29% reduction in blowing pressure for the new nozzle. The mass flux for a choked nozzle, i.e. nozzle with sonic velocities at the throat is

\[
\frac{\omega}{A_r} \propto \frac{p_o^2}{\sqrt{T_o}}
\]

According to equation (2), the 29% reduction in pressure means a 29% less flow of gas through the nozzle. Therefore, for a sootblower operating with a conventional nozzle at high blowing pressures, the new nozzle provides a means for better cleaning with no increase in mass flow or equivalent cleaning and a saving of the blowing medium through reduced nozzle pressure.

3. CASE I

3.1 BOILER PARAMETERS

The concept of improved cleaning for the same blowing pressure or equivalent cleaning while saving steam was adopted as a new sootblowing strategy by the Boise Cascade Wallula Mill, Wallula, Washington. The opportunity for sootblower steam savings was originally identified by the mill Operations and Maintenance groups. Once identified, a multi-phase implementation strategy was developed to limit financial risk. The initial phase was to install 28 new lance tube assemblies in the No. 3 Recovery Boiler, followed by a nozzle performance evaluation period. Once acceptable performance was verified, the second and third phases were to proceed with retrofit of spare lances with new nozzles for the remainder of the boiler.

The No. 3 Recovery Boiler used for this test is a two drum C-E unit with tangentially fired air system, a two stage superheater and a long flow economizer. A schematic of the convective pass for this boiler is given here in Figure 5. The swirl created by tangential firing leads to non-uniform fouling at the screen tube area of the superheater.

![Figure 5: Schematic of the Upper Furnace. New Nozzle & Conventional Nozzle Locations](image)
In the past, the boiler had to be shut down every six months for a water wash, to clear the deposit accumulation at this location which leads to a loss of final steam temperatures.

Of the 28 lance tubes replaced with the new nozzle, eight were replaced at the screen tube area of the superheater to achieve better cleaning and twenty were replaced in the economizer for steam savings (see Figure 5). The impact of this soot blowing strategy on the boiler operation has been monitored over two cycles; cycle 1(10/93 to 4/94) where the sootblowers operated as usual with the conventional nozzles, and cycle 2(4/94 to 10/94) with the adjusted poppet valve pressures and the new sootblower nozzles.

The liquor flow rate for the No. 3 Recovery Boiler is shown above in Figure 6. The unshaded area represents cycle 1 that lasted 169 days from water wash to water wash while, the shaded area represents cycle 2 that lasted 173 days. The final steam temperature over this period is given in Figure 7. During these two cycles, the liquor was reduced to <50% of the nominal mean flow to the boiler; cycle 1 on day 29 & 114 and cycle 2 day 66. With the exception of these interruptions the load to the boiler averaged around 1298t/day(3Mlb/day). The heating value for the liquor during this period was about 13MJ/kg(5580Btu/lb).

After 28 days into cycle 1, the final steam temperature has dropped from an initial value of about 390C(734F) by about 30C(86F). The loss of liquor due to low inventory on this day resulted in a boiler thermal shock, causing deposits to be shed from the superheater[9]. Upon the return to a full liquor firing load, the final steam temperature measured at the exit to the 2nd stage superheater, immediately climbed back to the post water wash value of about 390C(734F). However, in a matter of a few days, the superheater is once again fouled, leading to the degradation of the final steam temperature. A similar event occurred during cycle 2 on day 66. Here the liquor flow was reduced to perform an inspection of the boiler, to evaluate the performance of the new nozzles. As in the previous case, upon return to the nominal liquor flow a similar spike in the steam temperature is realized. However, in this case the presence of the new nozzles in the screen tube area meant that the final steam temperature was 10C(50F) above the value reached in cycle 1 and was maintained at this level for a longer period of time than was possible with the conventional nozzles.

Water/steam side temperature and mass flow through the economizer was monitored during the two cycles described above. This information was used to calculate a daily heat absorption rate Q, for the economizer and the two superheaters over the same period of time. These values have been normalized by the total heat absorbed Q_{TOTAL} and plotted here in Figure 8 as Q/Q_{TOTAL} where, Q=Q_{SH} for the superheater and Q=Q_{ECON} for the economizer. In the case of both cycles, the fraction of heat absorbed in the economizer increases, while the corresponding fraction for the superheater decreases with time. This is attributed to fouling, which starts at the superheater leading to an increased heat load on the economizer and a reduction of the final steam temperature(see Figure 7). Unchecked this phenomenon accelerates fouling causing a rise in flue gas temperatures in the generating bank and economizer, eventually resulting in the pluggage of the boiler.

![Figure 6: Feedwater and Liquor Flow rates for Cycles 1 and 2. For cycle 2, 28 sootblowers were fitted with Full Expansion Nozzles.](image)

![Figure 7: Comparison of final Steam Temperature for Cycles 1 and 2.](image)
Figure 8: Normalized Heat Absorption Rate for the Economizer and Superheater

Comparing the superheater results shown in Figure 8 for cycles 1 & 2, it is seen that a greater fraction of the heat is being absorbed during cycle 2. Since the input heat load to the boiler is practically unchanged, this increase can be attributed to the performance of the new nozzles at the entrance to the superheater. This is illustrated by the higher and more stable final steam temperature seen in Figure 7, for cycle 2. Since the final steam comes off the 2nd stage superheater, the eight new nozzles in its vicinity, significantly affects this temperature. The better heat transfer in the superheater given by the higher $\frac{Q_{SH}}{Q_{TOTAL}}$, lowers the flue gas temperature entering the generating bank and the economizer. This lowers the heat load to the economizer indicated by the lower ratio of $\frac{Q_{ECON}}{Q_{TOTAL}}$ for cycle 2 in Figure 8.

The availability of both steam and gas side temperature measurements for the economizer inlet and exit enables one to compute an overall heat transfer coefficient $\nu_{ECON}$, as follows

$$\nu_{ECON} = \frac{Q_{ECON}}{A_{ECON} F [LMTD]_{ECON}}$$

where $[LMTD]_{ECON}$ is the Logarithmic Mean Temperature, $A_{ECON}$ the surface area and F a factor based on the tube arrangement for the economizer [10]. The thermal resistance is the reciprocal of the overall heat transfer coefficient($\nu$). A fouling index R, that uses the thermal resistance has been proposed[10], where

$$R = \frac{1}{\nu_{fouled}} - \frac{1}{\nu_{clean}}$$

Since $A_{ECON}$ and F are constant, a plot of the simplified fouling index $FI_{ECON}$, based on the above principle and defined as

$$FI_{ECON} = \frac{[LMTD]_{ECON}^{clean}}{Q_{ECON}^{fouled}} - \frac{[LMTD]_{ECON}^{fouled}}{Q_{ECON}^{clean}}$$

has been calculated for the two cycles and plotted here in Figure 9.

Figure 9: Economizer Fouling Index

The aim here is that in the economizer the new nozzles would save steam, while maintaining the effective cleaning from the past. Figure 9 indicates that the fouling index, used in this case as a measure of effective cleaning in the economizer, shows no significant changes between cycles 1 & 2. A visual inspection of the economizer showed, that after 66 days (6/94) of operating with the new nozzles and using less steam per sootblower, the tubes were bare. At this point the steam flow was further reduced and the boiler was operated until the annual outage and water wash in October 1994. Once again, visual inspections showed practically no significant change in the fouling of the economizer. Due to the unavailability of gas side temperatures in the superheater region it was not possible to develop a fouling index similar to that described for the economizer.

The ID fan speed over the period of time corresponding to this test has been plotted in Figure 10.
The unrestricted flue gas passages following a water wash offers minimum resistance to the flue gas path, hence the ID fan speed required for a fixed mass of gas in the boiler is minimal. As time progressed, the fouling of the boiler poses greater restrictions for the movement of the flue gas, resulting in increased power consumption (RPM) by the fan. For cycle 1, Figure 10 shows that once full load was established, the fan RPM steadily rose from about 550 to 630 RPM over a period of about 120 days. With the new nozzles (cycle 2), the fan had a more stable speed averaging around 580 RPM over a similar length of time. This implies that the overall flow restriction to the flue gas or fouling of the boiler, should have been lower during cycle 2 in comparison to cycle 1.

3.2 STEAM SAVINGS

In April, 1994, eight screen tube sootblower nozzles and twenty economizer sootblower nozzles were upgraded. The steam flow was then reduced to 9.5 t/hr (20.5 klb/hr) for a 20% steam savings. After two months of operation, the boiler was taken off liquor for a visual inspection. Increased cleaning was observed in both the screen tube area and economizer. The economizer sootblowers were then reset to 8.4 t/hr (18.5 klb/hr) for an additional 10% steam reduction. The screen tube sootblowers were left as previously set.

The boiler was then operated through October, 1995, and taken down for an annual outage. After six months of operation with the new sootblower nozzles, visual inspection indicated increased cleaning in the screen tube area and equivalent cleaning in the economizer. As discussed earlier, the visual indications are quantitatively supported. Both of the original project goals were met concluding nozzle performance verification. Based on these savings, the mill is currently upgrading the remaining nozzles in the No. 3 Recovery boiler and all the nozzles in No. 2 Recovery Boiler.

Figure 11 shows two sets of data for sootblower steam consumption on the No. 3 Recovery boiler. In one set of data, 32 sootblowers (24 in the generating bank and 8 in the superheater screen tube area, GB & SH) are represented. The other data set shows 20 economizer sootblowers (ECON). The base line consumption was 11.4 t/hr (25.1 klb/hr) for GB & SH, and 11.5 t/hr (25.4 klb/hr) for ECON. The base line consists of a two-year average from 1/93 to 3/94. In October/95 the header pressure was dropped from 4MPa(580 psi) to 3.3MPa(480psi). While this was done, the poppet valves remain unchanged. This resulted in a flow rate around 7t/hr(15klb/hr). The boiler ran for the usual 6 months with this reduced flow and with no change in the boiler load. Subsequently the header pressure was put back to its original value of 4MPa(580 psi) in 6/96.

Steam savings were calculated by determining the amount of natural gas saved by reducing the load on one of the power boilers.

Figure 12 shows the financial impact of steam saving over ten months of operating with full expansion nozzles.
The following assumptions were made: (1) $0.25/MJ ($2.72/MMBTU) natural gas, (2) 70% boiler thermal efficiency, (3) enthalpy change of 2814 kJ/kg (1210 BTU/lb), which leads to (4) $10.36/1000 kg ($4.70/1000 lbs) of steam. Measured savings from 4/94 to 2/95 are $300,000 as shown by the cumulative value in Figure 12. The monthly savings are all about $30,000/month. When the project is complete, estimated annual savings are $700,000/yr for both boilers.

4. CASE II

4.1 BOILER PARAMETERS

The boiler discussed in case I is not a severely plugging unit. The run times between water washes is typically 6 months at a firing rate of about 1298 t/day (3Mlb/day) dry solids. In order to determine if the steam savings seen here is applicable to a heavily loaded unit, a second site was selected. The boiler selected was the unit at the Georgia-Pacific mill, Leaf River, Mississippi. A schematic of the recovery boiler is shown in Figure 13. This unit is a single drum, Gortaverken unit operating at about 2802 t/day dry solids (6Mlb/day).

![Figure 13: Schematic of Upper Furnace. 52 New Nozzle Location at the end of the Phase I.](image)

The run time on this unit is about 90 to 100 days. At this time the boiler is taken off line to be water washed. Today, 92 sootblowers operate 4 at a time. Saturated steam enters the primary superheater at the beginning of the convective pass and then passes on to the secondary superheater (closest to the generating bank). Sootblowing steam is extracted at the inlet to this superheater. Following this the steam makes a final pass through the tertiary superheater before exiting the boiler at 8.6Mpa(1250 psi). Sootblower steam consumption accounted for 18% of the steam produced.

Due to additional steam demand in the mill, a steam savings program was proposed. The mill was willing to accept the proposal on the condition that the run time was not compromised and that 100% of the risk be taken by Bergemann. Based on the fact that the full expansion nozzle could deliver the same PIP as a conventional nozzle while using less steam, the challenge was accepted. The project was broken into two phases. Phase I: 52 of 92 blowers in the most critical areas based on maximum usage were identified and replaced with new lance tubes fitted with the full expansion nozzle. The old lances (52) were sent back to the shop to be retrofitted with the new nozzles and returned to the mill. Phase II: The retrofitted lances were now installed in the remaining locations and the balance 12 lances were put in stock. Phase I was done during an outage and Phase II was done on the run. The installation was all completed by middle of 12/95. Following this the boiler was water washed and put back online on 12/15/96.

Historical records for this boiler show that fouling of the secondary superheater leads to an increase in flue gas temperatures in the generating section which eventually results in a pluggage of this section. Therefore, when the heat transfer rate reaches a critical value, the boiler is water washed. At present loads the cycle runs for about 90 days from water wash to water wash.

![Figure 14: Decay of heat transfer Rate-Secondary Superheater](image)

Figure 14 above shows the heat transfer coefficient for the secondary superheater as a % of the rate for a clean superheater.
The initial values >100% are a result of the ramping up of the boiler during startup. The thick line indicates the same ratio after the nozzles were changed. This period corresponds to the time of 12/95 to 3/96. In both these cases, the degradation of the heat transfer rate, i.e. increase of fouling is similar. The boiler ran for about 100 days before it was water washed in 4/96. This entire run was done with less steam consumption at all sootblowers. A set of curves for the tertiary superheater during the same period is shown in Figure 15. Once again, the pre and post CFE nozzle era shows no change in the cleaning efficiency as measured by the thermal efficiency of this surface.

![Figure 15: Decay of Heat Transfer Rate-Tertiary Superheater](image)

4.2 STEAM SAVINGS

The poppet valve at all sootblower locations were fully open, giving pressure readings of 2068kPa (300 psig). The nozzle size was 28mm (11/8 "). Steam flow for 4 blowers was measured at 51t/hr (112klb/hr). A daily average for 30 days is shown here in Figure 16. Also shown here is the new flow rate after the nozzles were changed out with the 25mm (1") full expansion nozzles. At this time the poppet valves were not turned down. Due to the smaller nozzle size the valve pressure increased to 2413kPa (350 psig). The mean flow is now around 40 t/hr (89klb/hr). This is ~21% reduction from the previous value of 51 t/hr (112klb/hr).

The dollar value of steam was determined by the cost to generate steam in the power boiler. Based on the fuel cost of $11.5/ton of bark, the steam cost was estimated at $2.36/klb of steam. At this cost the savings as a % reduction of sootblower steam is shown here in Figure 18.

![Figure 16: Sootblower Steam Consumption](image)

![Figure 17: Financial Impact on Steam Savings/Year. Monthly savings are estimated at around 40,000$.](image)

To date with a 21% reduction, the annual savings is estimated at about $480,000.00. At this reduced flow rate to the sootblowers the boiler ran its entire 90 day cycle which ended at the end of 3/95.

Since then a second cycle of similar length was completed between 4/96 through 6/96. Visual inspection showed that the flue gas passages looked no different from the past. Further, the economizer was reasonably clean. The heat transfer coefficient ratio for the economizer is shown here in Figure 18. These results are for the months of 12/95 through 3/96.
Figure 18: Decay of heat Transfer Rate-Economizers

During the period of 12/95, the economizer section still had conventional 28mm (1 1/8") nozzles installed. The heat transfer rate was around 98% of the value at the beginning of the cycle. Comparing this with the values from 1/96 through 3/96 where the new nozzles were in use, there appears to be no significant difference in its thermal efficiency. This is very similar to the observations from case I (Figure 9). Therefore, the mill has agreed to make a further reduction of 10% in the economizer sootblower steam consumption. Due to the less frequent use of these blowers in comparison to the remainder of the boiler, the net effect would be less than 30%. It is expected that the total savings would be around 25%, leading to an annual savings of $571,000.00.

5. IMPLICATIONS

The increased PIP delivering capability of the new nozzle suggests, the potential for improved cleaning over sootblowers using conventional nozzles in a recovery boiler. For a 2.5cm (1") nozzle, this benefit can be realized at operating pressures exceeding about 1379kPa (200psig).

Laboratory measurements for PIP suggest that the cleaning equivalent of a conventional nozzle can be reached with reduced consumption of the cleaning fluid. Once again, for the nozzles tested here, a savings of about 30% is expected at blowing pressures of 2413kPa (350psig). Measurements of PIP indicate, that there may be little merit in operating at pressures in excess of about 2068kPa (300psig) with a conventional nozzle.

Boiler parameters, indicate that the cleaner superheater in cycle 2, improves the overall boiler performance. This evidence strongly supports the view that a clean superheater can reduce the cleaning demand in the rest of the boiler. Further reduction in cleaning load and improvements in boiler thermal efficiency are expected once the complete change out of all sootblower nozzles are made.

For case I, it turns out that the boiler was able to operate for it’s normal 6 month run time with over 50% reduction in sootblower steam flow. Therefore, it is probable that in reality sootblower steam usage in recovery boilers are in excess of what is required.

In addition to the present savings and projected savings based on the cost to generate steam, a further savings will be realized through reduced wear and tear of the sootblowers and associated maintenance costs. The increase in sootblower motor amps at high blowing pressures during the retraction of the lance, is indicative of the extra work/load done by/on the sootblower.

6. CONCLUSIONS

The dimensional analysis for the flow in the lance, nozzle and jet provided the key parameters required to scale this problem. The good agreement between the quarter and full scale results have been obtained. The increased PIP measurements for the new nozzle have verified the gain to be achieved by fully expanding the gas prior to leaving the nozzle.

Both visual inspection and improved boiler performance parameters have confirmed that the increase in PIP delivered by the full expansion nozzle, does improve the cleaning capability of the sootblower. In this case the improved cleaning was achieved, with at least 20% less steam consumption. The steam saving potential for the new nozzle, together with the lower flue gas temperature due to the cleaner superheater has enabled a 30% reduction of the economizer blowing pressure while the boiler operated at full load.

It has been shown that steam savings is possible for recovery boilers operating at both modest and heavy loads. These studies show that there is a fair amount of latitude for trying to optimize the operating costs and thermal efficiency of present recovery boilers.

In both Case I where there is high fuel costs but moderate to low steam usage and, in Case II where there is low fuel costs but high steam usage the net result was a savings of about $30k to 40k per month. Naturally, in a location where both fuel costs and steam usage are high, tremendous savings could be realized by taking advantage of recent developments in sootblower technology.

The present study has looked at steam from the point of view of fuel cost. An opportunity lies in making gains in areas where fuel costs are low but electricity is at a premium. This will come by having to make lesser amounts of extraction form turbines. The author is not aware of such a study having been done to date.
The new nozzle together with judicious operating conditions, provide an excellent opportunity for optimizing the sootblower operation to achieve both better thermal efficiency while realizing financial savings through reduced steam consumption and a potential for lower maintenance costs.

7. ACKNOWLEDGEMENT

The author likes to thank Mark Easterwood, Alan Umemoto, Bill Gallacher & Jim Hanson from the Wallula mill and John Lacher & Ken Petrie from the Leaf River Mill for their efforts in gathering data for this study.

8. NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>d</td>
<td>Diameter</td>
</tr>
<tr>
<td>l</td>
<td>Distance from nozzle entrance to inner wall of lance tube (Figure 3)</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
</tr>
<tr>
<td>r</td>
<td>Radius</td>
</tr>
<tr>
<td>x</td>
<td>Axial distance along the jet center line</td>
</tr>
<tr>
<td>A</td>
<td>Area</td>
</tr>
<tr>
<td>F</td>
<td>Shape factor based on tube orientation</td>
</tr>
<tr>
<td>FLTD</td>
<td>Log-mean temperature difference</td>
</tr>
<tr>
<td>M</td>
<td>Mach number</td>
</tr>
<tr>
<td>PIP</td>
<td>Peak Impact Pressure</td>
</tr>
<tr>
<td>Q</td>
<td>Heat absorbed</td>
</tr>
<tr>
<td>R</td>
<td>Fouling index, equation (11)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>U</td>
<td>Axial velocity of the gas</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Gas density</td>
</tr>
<tr>
<td>( \overline{\rho} )</td>
<td>Overall heat transfer coefficient</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>Mass flow rate of gas</td>
</tr>
<tr>
<td>( p_{\text{lance}} )</td>
<td>Lance-nozzle pressure</td>
</tr>
<tr>
<td>o1</td>
<td>Upstream of Nozzle (in lance)</td>
</tr>
<tr>
<td>o2</td>
<td>At nozzle</td>
</tr>
<tr>
<td>e</td>
<td>Exit of the nozzle</td>
</tr>
<tr>
<td>x</td>
<td>Center line location in the jet</td>
</tr>
<tr>
<td>ECON</td>
<td>Economizer</td>
</tr>
<tr>
<td>GB</td>
<td>Generating bank</td>
</tr>
<tr>
<td>SH</td>
<td>Superheater</td>
</tr>
<tr>
<td>T</td>
<td>Throat</td>
</tr>
<tr>
<td>( \infty )</td>
<td>Ambient conditions</td>
</tr>
</tbody>
</table>

9. REFERENCES


(10th Latin American Recovery Congress, Concepción, Chile, August 26-30, 1996)