

Low Pressure Sootblowing Technology - The State of Development

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ABSTRACT

Sootblowers are vital tools for controlling fireside deposit buildup on heat transfer tubes in recovery boilers. Due to the high consumption of valuable high-pressure steam, extensive research and mill trials have been carried out over the past decade to explore the feasibility of utilizing less expensive low pressure steam for sootblowing. The results show that while it is technically feasible to use low pressure (145 – 200 psig or 10-17 bar g) steam for sootblowing, the sootblowing system must be properly designed, and the nozzles must be well engineered, in order to produce a cleaning power comparable to that of high pressure sootblowers. This paper discusses the underlying principle of low pressure sootblowing technology, the lower limit of steam pressure that can be used for sootblowing, the economic impacts, and the operational experiences at several low-pressure sootblowing installations.

INTRODUCTION

The accumulation of fireside deposits on recovery boiler heat transfer surfaces not only creates an insulating barrier that reduces the boiler thermal efficiency, but may also lead to costly unscheduled shutdowns due to the plugging of gas passages. Deposit accumulation is controlled by sootblowers, that periodically blast deposits off of tube surfaces with high pressure superheated steam. An effective sootblowing operation is vital to ensure continuous boiler operation and to achieve high boiler thermal efficiency.

The higher the pressure of the steam used for sootblowing, the higher the sootblower operating costs. The most expensive steam for sootblowing is taken directly from the boiler final superheater steam outlet. High pressure sootblowing refers to a sootblowing practice that utilizes 400 psig (28 bar g) or higher pressure steam, while low pressure sootblowing refers to an operation with the steam from the turbine extraction at 145 – 200 psig (10 – 13.8 bar g).

A research project was initiated at the University of Toronto in 2001 to examine the feasibility of utilizing low pressure steam for sootblowing. The work included extensive laboratory tests, full scale tests in operating recovery boilers, and numerical modeling [1,2,3]. In the past five years, there have been three new recovery boilers commissioned in the United States; all are equipped with low pressure sootblowers.

This paper briefly reviews the underlying principle of low pressure sootblowing technology, the lower limit of steam pressure that can be used for sootblowing, and discusses the operational experiences at the three low pressure sootblowing installations, and the economic impacts.

BASIC CONCEPT

The deposit removal process involves the impingement of a sootblower jet on a deposit that is attached to one or more boiler tubes. The force of the jet exerted on the deposit (F_{jet}) induces a stress at the deposit/tube contact area ($A_{deposit/tube\ contact\ area}$). The deposit will be removed from the tube if this stress exceeds the adhesion strength of the deposit, i.e.,

$$S_{jet} > S_{adhesion} \quad (1)$$

As expressed in Equation (1) and conceptually shown in Figure 1, the efficacy of the deposit removal process depends on two key parameters, (i) the stress imposed by the jet on the deposit (S_{jet}) and (ii) the strength of deposit adhesion to the tube ($S_{adhesion}$). The magnitude of these parameters determines whether or not deposit accumulation in the boiler bank can be controlled by sootblowers. Although S_{jet} is directly related to the sootblower design, the magnitude of $S_{adhesion}$ is controlled by factors unrelated to the sootblower equipment, such as boiler operating practices and liquor chemistry [4, 5].

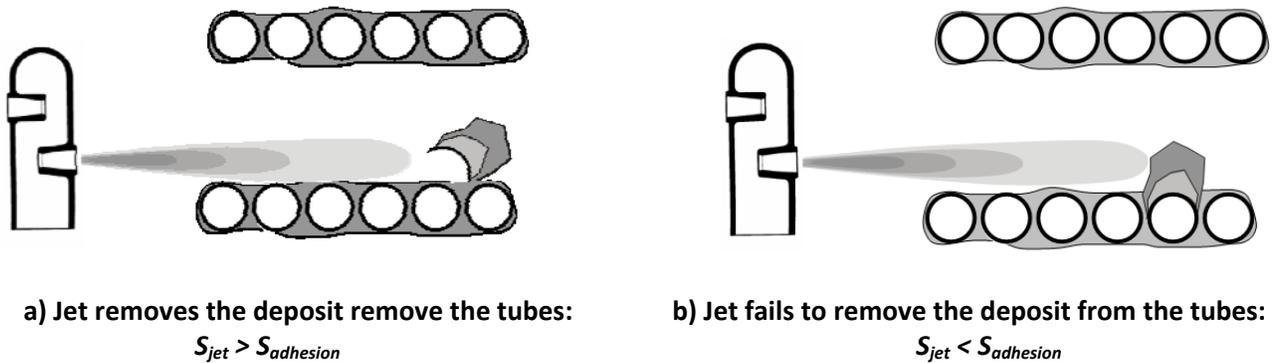


Figure 1. Deposit removal by a sootblower jet

S_{jet} is simply the jet force exerted on a deposit divided by the deposit/tube contact area.

$$S_{Jet} = \frac{F_{Jet}}{A_{deposit / tube \text{ contact area}}} \quad (2)$$

The deposit removal criteria, equation (1), can then be rewritten as

$$F_{Jet} > S_{Adhesion} A_{deposit / tube \text{ contact area}} \quad (3)$$

During the development of low pressure sootblowing technology, the crucial question to be answered was whether it was possible for low pressure steam to generate a comparable F_{jet} to that of a high pressure sootblower. In mathematical form, this can be written as:

$$F_{jet} |_{Low \text{ Pressure}} \stackrel{??}{=} F_{jet} |_{High \text{ Pressure}} \quad (4)$$

$$\int IP_{Jet} * dA |_{Low \text{ Pressure}} \stackrel{??}{=} \int IP_{Jet} * dA |_{High \text{ Pressure}}$$

where:

- IP_{jet} : Jet impact pressure (psi or kN/mm^2)
- A : Deposit /jet contact area (in^2 or mm^2)

Jet impact pressure is the pressure exerted by the jet on a solid object.

Figure 2 shows the jet impact pressure along the y-axis versus deposit/jet contact area on the x-axis. The thick curve represents the impact pressure profile of a low pressure jet, while the thin curve corresponds to the high pressure jet. The Peak Impact Pressure (PIP) is the point located at the peak of each impact pressure curve. As seen in Figure 2, the PIP of the low pressure jet is lower than that of the high pressure jet. However, the areas under each curve ( for high pressure, and  for low pressure), which represent the jet force exerted on a deposit, can be equal. Note that the force exerted on a deposit is the impact pressure multiplied by the deposit/jet contact area, which is essentially the area under the curve.

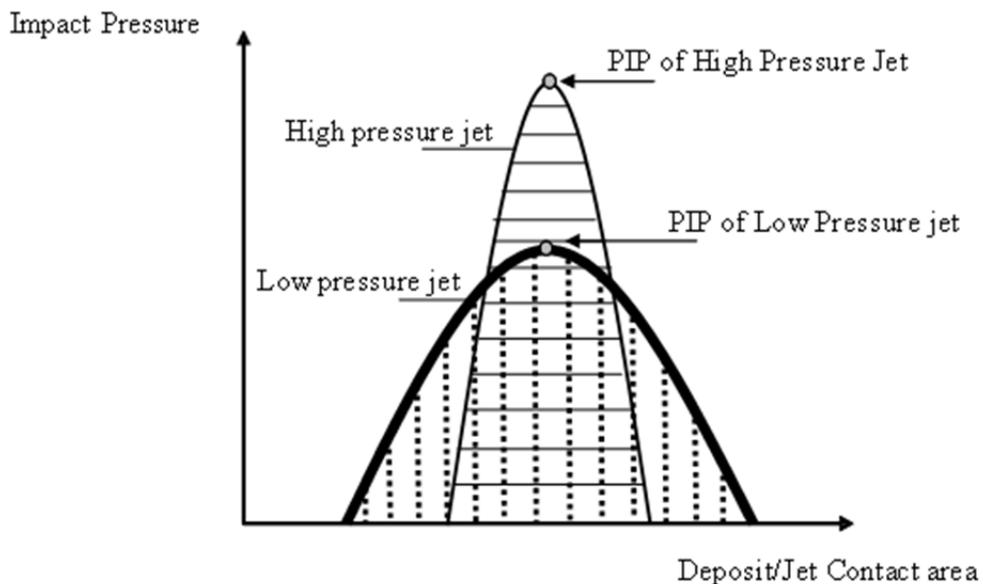


Figure 2. Peak Impact Pressure of high and low pressure jets

The feasibility of utilizing lower pressure steam for sootblowing hinges on the fact that the jet force (i.e., the area under the curve) produced by low pressure steam can be made equal to that of high pressure steam, as depicted in Figure 2.

DESIGN OF LOW PRESSURE SOOTBLOWER

A high pressure sootblower is generally designed to produce a total force of 245 lbf (1090 N). Total force is defined as the force exerted on an artificial flat surface placed right at the nozzle exit (see Appendix A for the detailed theory to calculate the jet total force, for a given nozzle pressure and a sootblower nozzle). In most cases, this design force is only used occasionally to combat heavy fouling or plugging. Under normal boiler operating conditions, the highest total jet force is typically set to 200 lbf (890 N), for sootblowers operating in areas with tenacious deposits, such as superheater sections.

To ensure that a low pressure sootblower has the same capability as a high pressure sootblower for dealing with heavy fouling, the low pressure sootblower must be designed to produce 245 lbf (1090 N) total force. Table 1 shows nozzle pressure (P_{Nozzle}) and steam flow rate requirements for both high and low pressure sootblowers to achieve 245 lbf (1090 N) total jet force.

Table 1. Pressure and flow rate requirements to achieve 245 lbf (1090 N) total jet force

	D_{Throat}	P_{Nozzle}	Steam Flow Rate
Standard High Pressure Sootblower	1" (25.4 mm)	254 psig (17.5 bar g)	19,700 lb/hr (8.9 ton/hr)
11 bar – Low Pressure Sootblower	1.25" (31.75 mm)	164 psig (11.3 bar g)	20,450 lb/hr (9.3 ton/hr)
6 bar – Low Pressure Sootblower	1.75" (44.45 mm)	88 psig (6.1 bar g)	22,900 lb/hr (10.4 ton/hr)

* The steam temperatures for high pressure, low pressure 11 bar, and low pressure 6 bar are assumed to be 572°F (300 °C), 485°F (252 °C), and 470 °F (243 °C), respectively.

P_{Nozzle} is the steam pressure upstream of the two sootblower nozzles, as depicted in Figure 3. P_{Nozzle} and the design of the nozzles are the two main parameters that determine the jet cleaning force and the steam consumption [6].

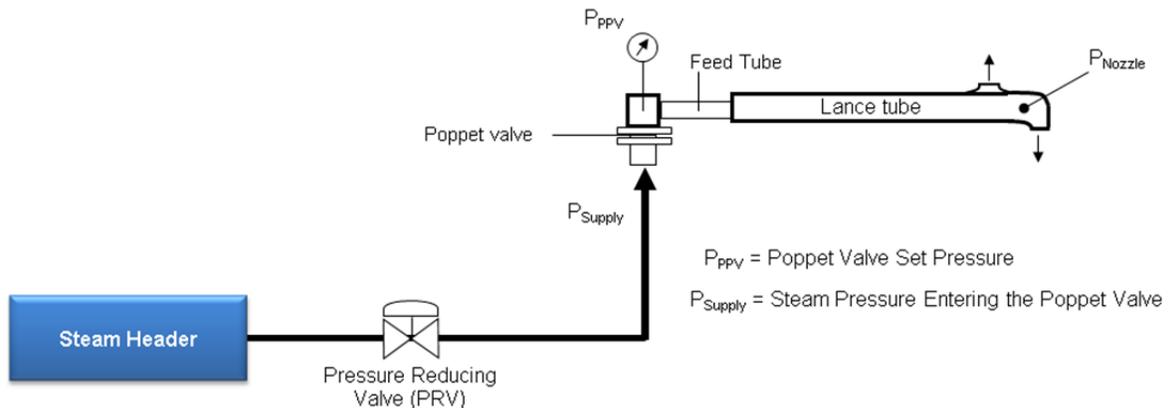


Figure 3. Sootblower Piping Arrangement.

Standard high pressure sootblowers are designed with little consideration to minimize pressure drop. Significant pressure drop in the sootblower is expected to occur especially across the poppet valve and feed tube. To achieve 254 psig (17.5 bar g) nozzle pressure on a 13-23 ft (4 - 7 m) long sootblower, the P_{Supply} should be at least 350 psig. Hence, for a pulp mill that utilizes 400 psig (28 bar g) steam for sootblowing, the pressure drop across the pressure reducing valve (PRV), steam piping, and fittings should not be more than 50 psi (3.5 bar). This is to ensure that the sootblowers will be supplied with 350 psig (24 bar g) steam and thereby deliver the 245 lbf designed total force.

The 11 bar low pressure sootblower requires about 11 bar nozzle pressure to produce 245 lbf total force. Based on thermodynamic principles, the calculated steam flow rate required for an 11 bar low pressure sootblower to produce a total force of 245 lbf is about 750 lb/hr (0.34 ton/hr) more than a high pressure sootblower. This is in reasonable agreement with the results of a full scale test at a kraft mill [7], where an 11 bar low pressure sootblower required 500 lb/hr (0.23 ton/hr) more steam to achieve a cleaning performance comparable to a high pressure sootblower.

Three new recovery boilers commissioned in the United States in recent years are all equipped with 11 bar low pressure sootblowers. The steam used for sootblowing in two of the installations are from 200 psig (13.8 bar g) turbine extraction. To achieve the design force of 245 lbf, the sootblower system was designed with a pressure drop of only 36 psi (2.5 bar) [i.e., 200 psig – 164 psig] from the steam header to the sootblower nozzles. The other installation uses 300 psig (20.7 bar g) turbine extraction for sootblowing, but it is presently evaluating the possibility of using lower steam pressure from its 150 psig turbine extraction for sootblowing.

The lowest design pressure that a sootblowing system can serve a recovery boiler is probably 6 bar. This is based on the fact that the nozzle exit diameter will have to be larger than the boiler tube OD of 2.5" (63.5 mm) in order to achieve the design total force of 245 lbf – see Appendix B for the derivation of the equation and concepts. Although a large diameter jet can produce a powerful total jet force, the actual force that is exerted on a deposit will be much smaller if a significant portion of the jet does not directly impinge on the deposit, as shown in Figure 4.

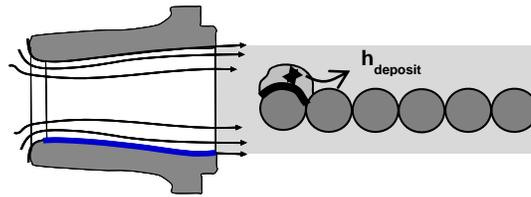


Figure 4. Large Sootblower Jet Impinges on boiler tubes

Therefore, it is desirable to design fully expanded sootblower nozzles with an exit diameter not to exceed the boiler OD of 2.5" (63.5 mm) to ensure that the jet force can be effectively exerted on boiler bank deposits.

As seen in Table 1, a low pressure sootblower operating at 6 bar requires 22,900 lb/hr (10.4 ton/hr) low pressure steam in order to produce 245 lbf design force. The flow rate is 3,400 lb/hr (1.54 ton/hr) higher than the flow rate of a high pressure sootblower. A 6 bar low pressure sootblower can be operated with the steam from 145 psig (10 bar g) turbine extraction, allowing approximately 58 psi (4 bar) total pressure drop in the sootblower piping system, fittings, and the sootblower equipment itself.

ECONOMIC ANALYSIS

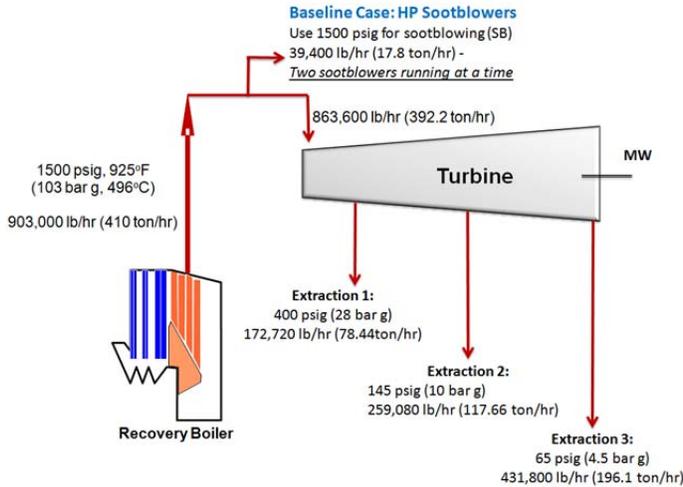
The economic benefits of utilizing low pressure steam for sootblowing come primarily from the fact that the low pressure steam has less intrinsic value (i.e., less potential to generate power) than high pressure steam. In this section, the economic benefit of utilizing 6 bar and 11 bar low pressure sootblowers will be examined and compared with the cost of operating high pressure sootblowers. This analysis is based on the assumption that the 6 bar and 11 bar low pressure sootblowers will consume 3,400 lb/hr (1.54 ton/hr) and 750 lb/hr (0.34 ton/hr) more steam, respectively.

Low Pressure Sootblowers Operating at 6 bar

In this section, two high pressure sootblowing scenarios (Baseline and Case A) are compared to 6 bar low pressure sootblowing (Case B). In the baseline case, the steam for sootblowing is taken from the final superheated steam produced by the boiler. In case A, the steam comes from the first turbine extraction, which typically has a steam pressure equal to or greater than 400 psig (28 bar g). Case B, of the 6 bar low pressure sootblower, utilizes steam taken from the second turbine extraction at 145 psig (10 bar g).

In all cases, it is assumed that the steam turbine is not at full capacity, and that its isentropic efficiency is 90%. The turbine extraction steam demand for mill processes is set to 172,720 lb/hr (78.44 ton/hr) for extraction 1, 259,080 lb/hr (117.66 ton/hr) for extraction 2, and 431,800 lb/hr (196.1 ton/hr) for extraction 3.

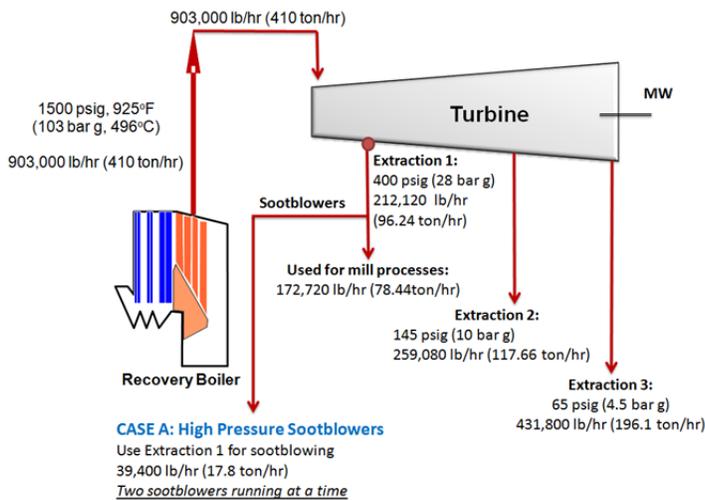
As discussed in the previous section, the standard high pressure sootblower, at a design condition of 245 lbf (1090 N) total jet force, consumes 19,700 lb/hr (8.9 ton/hr) per blower. In this analysis, it is assumed that the recovery boiler requires two sootblowers operating at the same time to control the deposit accumulation. With two sootblowers running at the same time, the total steam consumption is 39,400 lb/hr (17.8 ton/hr).



Baseline Case: HP Sootblowing operated with final SPHR Steam			
	ΔH	Flow Rate	MW Gen
Recovery boiler steam production		410.00 ton/hr	
Sootblower steam consumption		17.80 ton/hr	
28 bar Extraction	325.02 KJ/Kg	78.440 ton/hr	7.08 MW
10 bar Extraction	516.48 KJ/Kg	117.66 ton/hr	16.88 MW
4.5 bar Extraction	633.34 KJ/Kg	196.10 ton/hr	34.50 MW
			58.47 MW

Figure 5. Baseline Case: HP sootblowing using final superheater steam

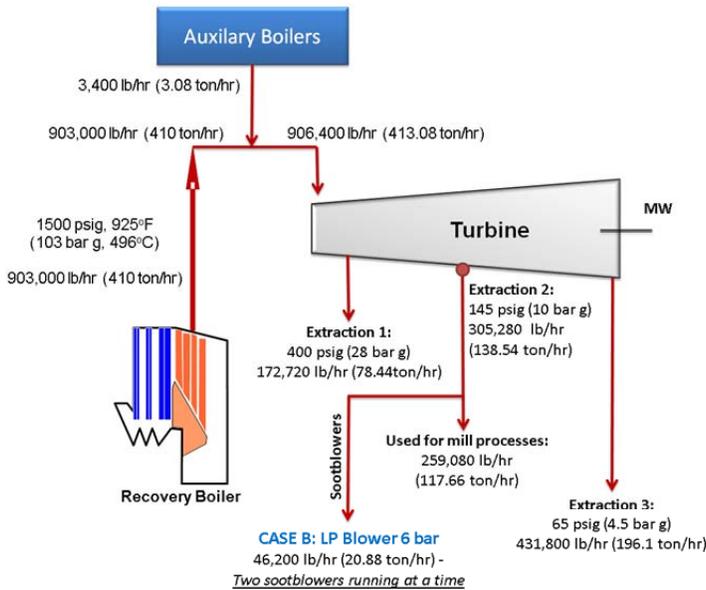
In the baseline case, the high pressure sootblowers consume 39,400 lb/hr (17.8 ton/hr) final superheated steam of 1500 psig (103 bar g). Before the steam is supplied to the sootblower piping line, this high pressure steam passes through a pressure reducing valve and is then directed to a lower pressure header, which is typically at 600 psig (41 bar g). As seen in Figure 5, the baseline case will generate 58.47 MW of power.



CASE A: HP Sootblowing operated with steam from turbine extraction 1			
	ΔH	Flow Rate	MW Gen
Recovery boiler steam production		410.00 ton/hr	
Sootblower steam consumption		17.80 ton/hr	
28 bar Extraction	325.02 KJ/Kg	96.24 ton/hr	8.69 MW
10 bar Extraction	516.48 KJ/Kg	117.66 ton/hr	16.88 MW
4.5 bar Extraction	633.34 KJ/Kg	196.10 ton/hr	34.50 MW
			60.07 MW

Figure 6. CASE A: HP sootblowing, using steam from turbine extraction 1

The high pressure sootblowers in Case A consume the same 39,400 lb/hr (17.8 ton/hr) of steam, but in this case, all final superheated steam is directed to the steam turbine and the sootblowing steam is taken from the first turbine extraction at 400 psig (28 bar). The mill demand for 400 psig (28 bar g) steam remains constant at 172,720 lb/hr (78.44 ton/hr) but the total demand for extraction 1 increases to accommodate the sootblower steam consumption. By directing all of the valuable final superheated steam to the turbine and using 400 psig (28 bar g) extraction for sootblowing, the turbine generates an additional 1.61 MW as compared to the baseline case (60.07 – 58.47 = 1.6 MW).



CASE B: 6 bar LP Sootblower operated with steam from turbine extraction 2			
	ΔH	Flow Rate	MW Gen
Recovery boiler steam production		410.00 ton/hr	
Sootblower steam consumption		17.80 ton/hr	
Additional SB steam consumption (compensated by the auxiliary boilers)	633.34 KJ/Kg	3.08 ton/hr	0.54 MW
28 bar Extraction	325.02 KJ/Kg	78.44 ton/hr	7.08 MW
10 bar Extraction	516.48 KJ/Kg	138.54 ton/hr	19.88 MW
4.5 bar Extraction	633.34 KJ/Kg	193.02 ton/hr	33.96 MW
Total MW Production			61.46 MW
Effective MW Production			61.24 MW

Figure 7. CASE B: Low pressure sootblowing, using 10 bar steam from turbine extraction 2

To achieve the design total force of 245 lbf, the 6 bar low pressure sootblower requires 3,400 lb/hr (1.54 ton/hr) more steam. With two sootblowers operating at the same time, the additional steam flow rate is 6,800 lb/hr (3.08 ton/hr), as reflected in Figure 7. This additional 3.08 ton/hr is not part of the baseline design. Hence, the steam must be taken from the portion that is supposed to be throttled to turbine extraction 3, causing a 3.08 ton/hr steam deficit in the 65 psig (4.5 bar g) extraction line. Since the mill steam demand from turbine extraction 1, 2, and 3 is constant, the steam flow rate exiting turbine extraction 2 must be increased to accommodate the sootblower steam demand, and so the auxiliary boilers must be used to compensate for the deficit in the turbine extraction 3.

The additional steam produced by the auxiliary boiler generates 0.54 MW, bringing the total MW generation by the steam turbine to 61.46 MW. However, there is a fuel cost associated with the MW generation from the auxiliary boiler. If the energy is valued at \$50/MWh and the fuel cost is assumed to be \$6/MMBtu, the net gain of using the auxiliary boiler to generate the additional MW is as follows:

$$\text{Net gain} = \text{Value of the energy} - \text{Auxiliary Fuel Costs} \quad (5)$$

$$= \frac{\$Y}{MWh} - \frac{\$X}{10^6 Btu} \frac{3.415 \times 10^6 Btu}{MWh}$$

where Y is the value of the energy (\$50/MWh) and X is the cost of the auxiliary fuel (\$6 / MMBtu)

$$\text{Net Gain} = \frac{\$50}{MWh} - \frac{\$6}{10^6 Btu} \frac{3.415 \times 10^6 Btu}{MWh} \quad (6)$$

$$= \frac{\$50}{MWh} - \frac{\$20.5}{MWh} = \frac{\$29.5}{MWh}$$

Hence, the 0.54 MW will be valued at \$29.5/MWh, instead of the full energy value of \$50/MWh.

$$0.54 \text{ MW} \frac{\$29.5}{MWh} = \frac{\$15.93}{h} \quad (7)$$

The effective additional MW generation by the auxiliary boiler, taking into account the fuel cost, will be

$$\frac{\$15.93}{h} \div \frac{\$50}{MWh} = 0.32 \text{ MW} \quad (8)$$

In other words, with a fuel cost of \$6/MMBtu, the additional MW generation by the auxiliary boiler becomes 0.32 MW, instead of 0.54 MW.

The effective total MW production by the turbine is

$$MW_{\text{From 28 bar extraction}} + MW_{\text{From 10 bar extraction}} + MW_{\text{From 4.5 bar extraction}} + \text{Effective } MW_{\text{From auxiliary boiler}}$$

$$7.08 \text{ MW} + 19.88 \text{ MW} + 33.96 \text{ MW} + 0.32 \text{ MW} = 61.24 \text{ MW} \quad (9)$$

Hence, by utilizing 6 bar low pressure sootblowers, the net MW gain is 1.17 MW compared to Case A (61.24 - 60.07 = 1.17 MW), and 2.77 MW compared to the baseline case (61.24 - 58.47 = 2.77 MW).

Figure 8 shows the zero net gain chart for the additional steam produced by the auxiliary boiler and the MW generation. Above the straight line, the mill gains additional effective MW; below the straight line, the mill

loses MW. With the value of energy assumed to be \$50/MWh, the mill will have zero MW gain from the auxiliary boiler if the cost of fuel is \$15/MMBtu. This \$15/MMBtu break even cost of fuel corresponds roughly to the cost of #6 fuel oil. Note that this calculation assumes that the turbine has additional capacity available to accept this additional steam from the auxiliary boiler.

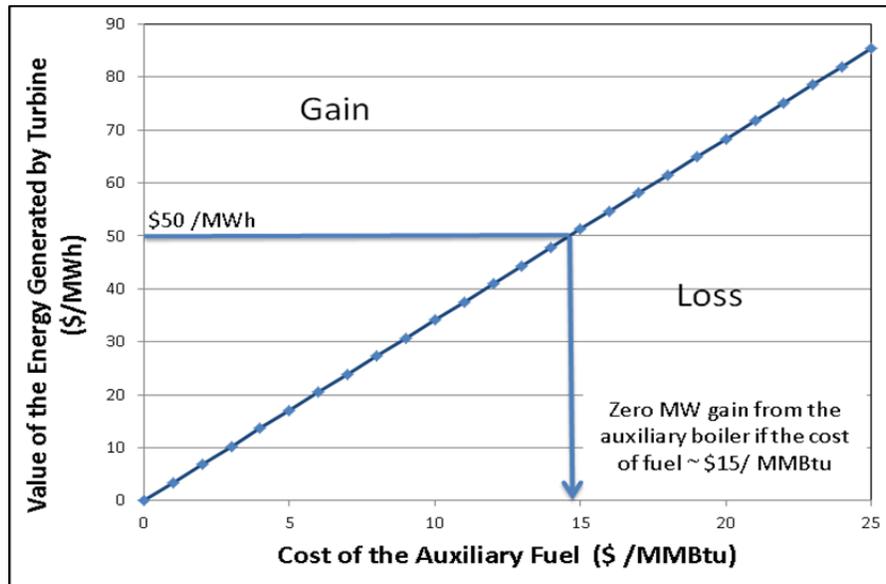


Figure 8. Zero net gain chart for the additional steam produced by the auxiliary boiler

Table 2 summarizes the economic benefits of utilizing 6 bar low pressure sootblowers supplied by 10 bar g turbine extraction. With the cost of energy assumed to be \$50/MWh (€40/MWh), using the 6 bar low pressure sootblowers generates 2.77 MW additional power (worth ≈\$1.2 Million/year) and 1.17 MW (worth ≈\$500K) compared to the baseline case and Case A, respectively.

Table 2. Economic benefits of utilizing 10 bar g turbine extraction for sootblowing

	Baseline Case SB from Final SPHR	CASE A SB from 28 bar extraction	CASE B SB from 10 bar extraction
Type of Sootblower	HP Blower	HP Blower	6 bar LP Blower
Power Generation ==>	58.47 MW	60.07 MW	61.24 MW
Economic benefits due to LP Blower 6 bar	Additional Power Generation	\$ Saving per year assuming \$50/MWh	
Compare to Baseline Case	2.77 MW	\$1,196,640	
Compare to CASE A	1.17 MW	\$505,440	

Low Pressure Sootblowers Operating at 11 bar

The same procedure described above is used to evaluate the economic benefits of utilizing 11 bar low pressure sootblowers, supplied by 13.8 bar g turbine extraction. Table 3 summarizes the results.

Table 3. Economic benefits of utilizing 13.8 bar g turbine extraction for sootblowing

	Baseline Case SB from Final SPHR	CASE A SB from 28 bar extraction	CASE B SB from 13.8 bar extraction
Type of Sootblower	HP Blower	HP Blower	11 bar LP Blower
Power Generation ==>	56.69 MW	58.30 MW	59.02 MW
Economic benefits due to LP Blower 6 bar	Additional Power Generation	\$ Saving per year assuming \$50/MWh	
Compare to Baseline Case	2.33 MW	\$1,006,560	
Compare to CASE A	0.72 MW	\$311,040	

The additional power generation and the \$ savings per year are lower when compared to utilizing 10 bar extraction for 6 bar low pressure sootblowing.

OPERATIONAL EXPERIENCES

Three new recovery boilers have been commissioned in the United States within the past five years, and all are operated with low pressure sootblowers. Table 4 shows the design specifications of the three recovery boilers. To date, there have been no operation interruptions due to heavy fouling or plugging at any of these boilers.

Table 4. Design specification of the three recovery boilers

	Year Commissioned	Capacity (BLDS)	% dry Solid	Steam Conditions	Sootblower
RB - A	2007	6.3 million lb/day (2,860 ton/day) DS	80%	1500 psig, 925 °F, 903 KPPH 103 psig, 496 °F, 410.5 ton/hr	88 LP Blowers (32 SPHR, 34 GB, 24 ECON)
RB - B	2008	6.0 million lb/day (2,720 ton/day) DS	80%	1500 psig, 950 °F, 900 KPPH 103 psig, 510°F, 410 ton/hr	88 LP Blowers (32 SPHR, 34 GB, 24 ECON)
RB - C	2011	3.5 million lb/day (1,588 ton/day) DS	75%	1500 psig, 925°F, 624 KPPH 103 bar g, 496°C, 283.3 ton/hr	76 LP Blowers (36 SPHR, 18 GB, 22 ECON)

Recovery Boiler A

Of the three boilers, RB – A has run at its design capacity for the longest time, and so in this section, the discussion will focus on RB – A. Figure 8 shows a schematic diagram of RB – A and the sootblower locations.

RB-A began operating on January 25th, 2007 with all low pressure sootblowers running at a uniform poppet valve set pressure of 170 psig (11.72 bar g). At this poppet valve set pressure, the sootblowers are run slightly below the design total jet force of 245 lbf. At these conditions, P_{Nozzle} is 158 psig (10.9 bar g), the steam flow rate is 20,200 lb/hr (9.2 ton/hr), and the resulting total jet force is 234 lbf (1041 N).

Online boiler inspection using an infrared camera was carried out on February 8th, 2007 to study the boiler and sootblower operations with low pressure sootblowing system. It was confirmed that the secondary and tertiary superheater platens swung widely, especially when sootblowers S23, S24, S25, S26, S29, S30, S31, and S32 were in operation. It was determined that the poppet valve set pressure of 170 psig (11.72 bar g) was too high, and the poppet valve pressure in the superheater and generating bank were consequently re-set to 150 psig (10.3 bar), and in the economizer to 107 psig (7.4 bar g).

At 150 psig poppet valve pressure, the P_{Nozzle} is 140 psig (9.65 bar g), the steam flow rate is 18,000 lb/hr (8.16 ton/hr), and the resulting total jet force is 205 lbf (913 N). In the economizer where the poppet is set to 107

psig, the P_{Nozzle} is 99.42 psig (6.85 bar g), the steam flow rate is 13,184 lb/hr (5.98 ton/hr), and the resulting total jet force is 142 lbf (630 N).

As seen in Figure 9, the 200 psig (13.8 bar g) steam enters the sootblower piping line from the top of the boiler and is distributed uniformly to the north and south sides of the boiler. The sootblower piping system is not equipped with steam traps; instead, orifices are located at the ends of the sootblower piping lines. The unused steam leaves the boiler through an orifice and is then directed to a 65 psig (4.5 bar g) steam header, and used for various mill processes.

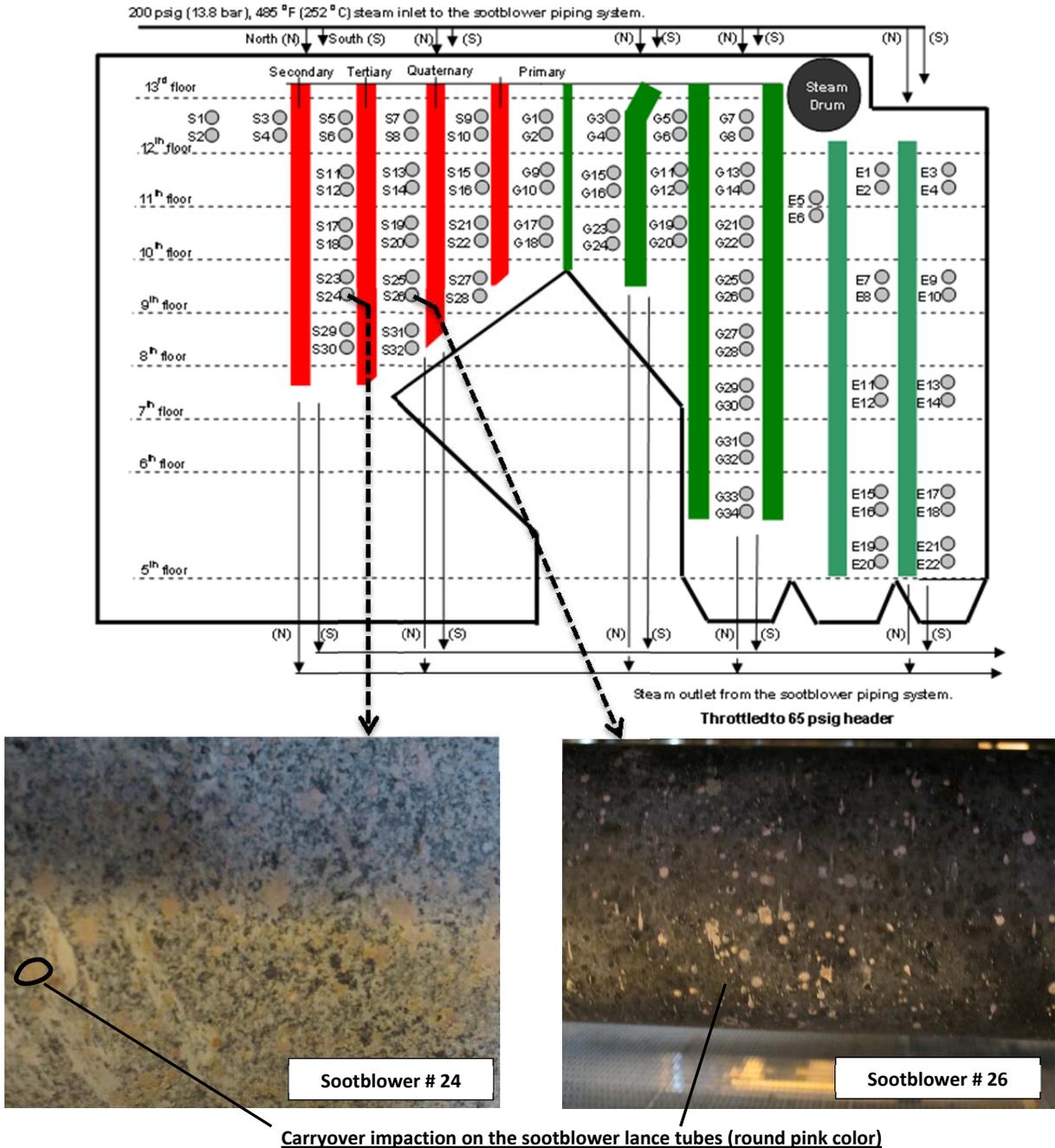


Figure 9. Schematic diagram of RB – A

Superheater Cleanliness

Round pink carryover deposits were observed on sootblower lance tubes in the superheater region. These carryover particles were fused and very hard. The pink color faded and turned gray with time as the deposits were exposed to air and oxidized. The quantity of pink deposits on the lance tube was much less in the generating bank and regions further downstream.

Figure 10 shows the attemperator flow profile and the total steam flow production, as a function of time. This plot can be used to evaluate the performance of the low pressure sootblowers in controlling the deposit accumulation in the superheater. The attemperator flow is the flow of feed water that is mixed with the main steam to control the main steam temperature. When the boiler superheater is clean, the efficiency of the heat transfer to the superheated steam is highly variable main steam temperature. The main steam temperature is controlled and held constant by utilizing the attemperator flow. High attemperator flow thus suggests that the superheater section is clean. The opposite is true when the attemperator flow is low; this suggests that the superheater section is experiencing heavy fouling.

The boiler is designed to produce 900 KPPH steam; the 900 KPPH dotted line indicates the boiler is operating at full load design condition. Figure 10 shows the steam production and the attemperator flow data from December 1st, 2010 to August 3rd, 2012. This period includes 5 month of continuous operation, during which the boiler was run at or above the design load.

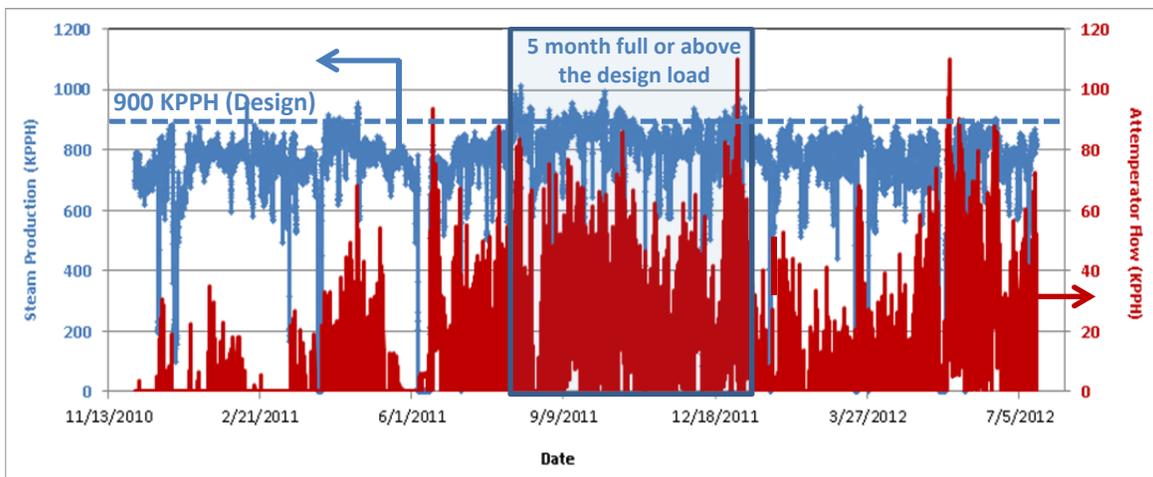


Figure 10. RB – A superheater main steam temperature trend Dec 1, 2010 to Aug 3rd, 2012

The attemperator flow trend follows the steam production profile. During the highlighted 5 month period, there was no indication that the attemperator flow was trending downward at a given steam production. This suggests that the low pressure sootblowers were controlling deposit accumulation in the superheater section.

Generating Bank Cleanliness

The generating bank gas outlet temperature can be used to assess the cleanliness of the generating bank. The dirtier the generating bank, the higher the gas outlet temperature will be. The generating bank gas outlet temperature measured after a water wash, when the boiler is clean, can be considered a benchmark temperature.

Figure 11 illustrates the generating bank gas outlet temperature for the same period as shown in Figure 10. Following the scheduled outage in June, 2011, the temperature gradually increased and then remained

constant at a level below 800 °F. This suggests that the low pressure sootblowers were successfully controlling the deposit accumulation in the generating bank.

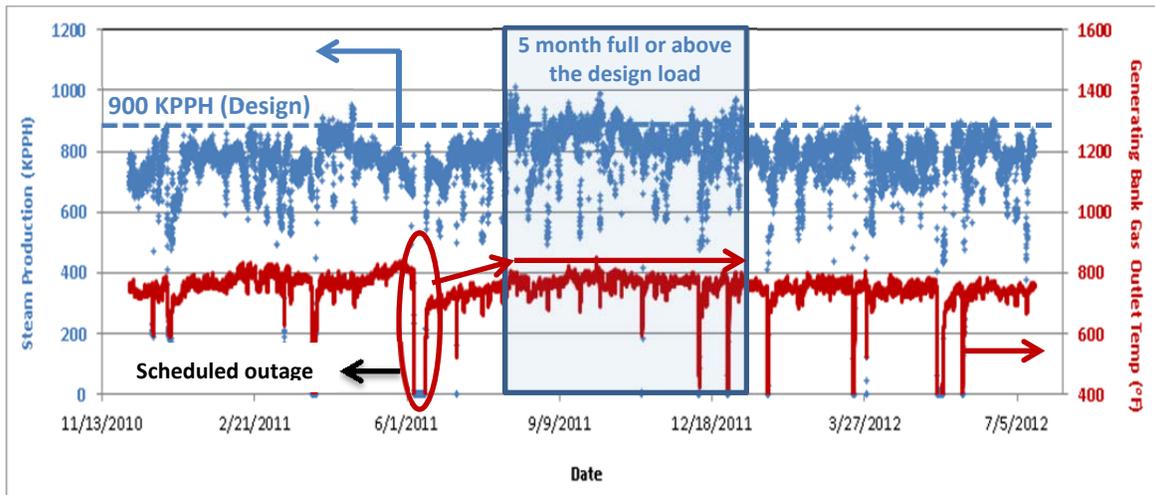


Figure 11. RB – Averaged generating bank gas outlet temperature trend Dec 1, 2010 to Aug 3rd, 2012

Economizer and Overall Cleanliness

The economizer exit gas temperature (EEGT) and ID fan rpm speed trends not only reflect the cleanliness of the economizer, but also the overall cleanliness of the boiler. Figures 12 and 13 show the averaged EEGT and ID fan rpm speed profiles, respectively.

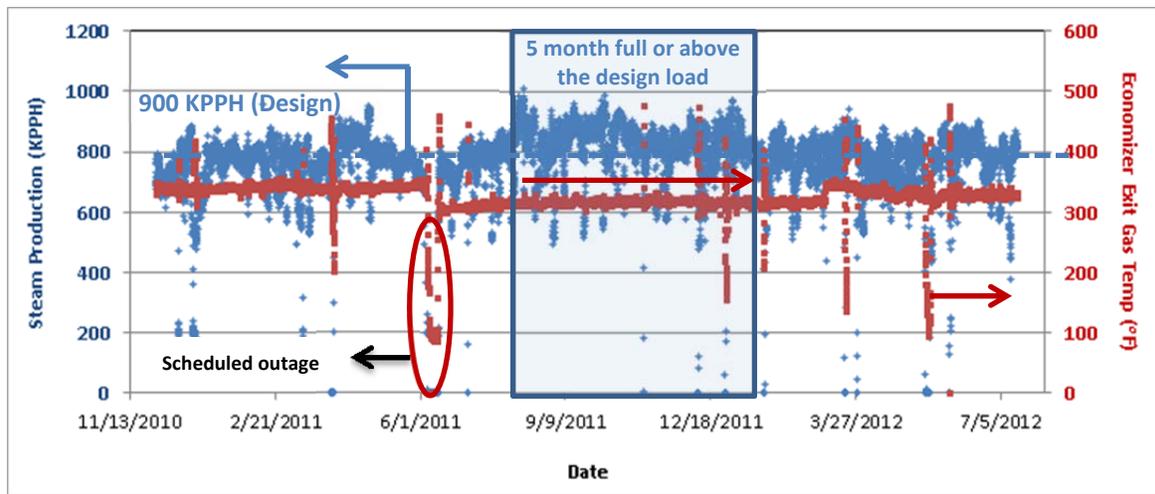


Figure 12. RB – Averaged economizer exit gas temperature trend Dec 1, 2010 to Aug 3rd, 2012

The EEGT following the scheduled outage in June, 2011, slightly increased but remained constant at about 320 °F. This suggests that the low pressure sootblowers were successfully controlling the overall deposit accumulation in the boiler.

The ID fan trend, as shown in Figure 13, follows the trend of the total steam production. No upward trend of ID fan was observed at a given steam production level.

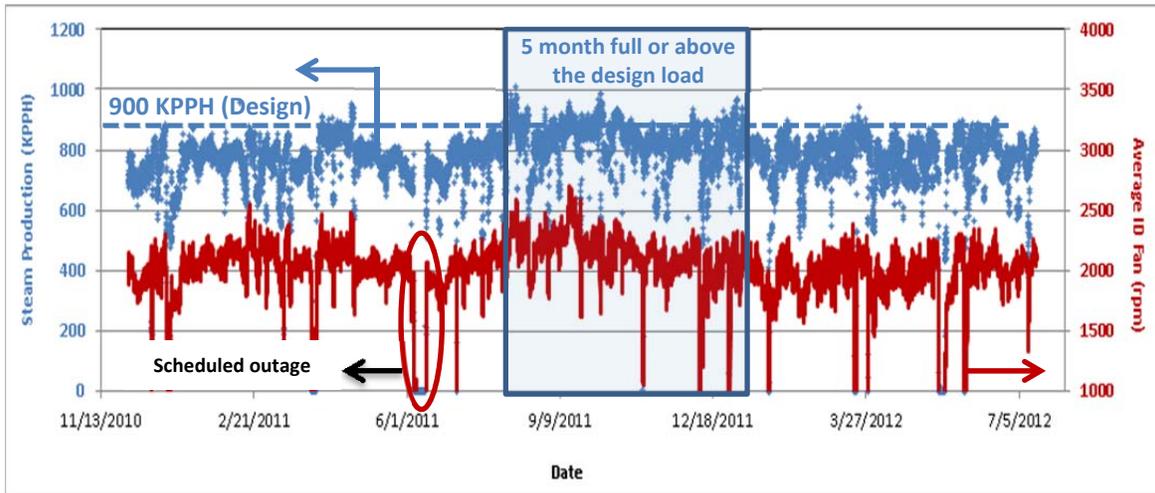


Figure 13. RB – A averaged ID fan rpm speed trend Dec 1, 2010 to Aug 3rd, 2012

Overall, it can be concluded that the low pressure sootblowing operation at RB – A has successfully controlled deposit accumulation since 2007.

Recovery Boiler B

RB – B has a similar design and capacity as RB –A. However, RB – B is rarely run at full load. At the present time, the sootblowers are all run a poppet valve set pressure of 105 psig (7.24 bar g). At this setting, P_{Nozzle} is 97.55 psig (6.7 bar g), the steam flow rate is 12,960 lb/hr (5.8 ton/hr), and the resulting total jet force is 139 lbf (617 N). The boiler has run very well, with no interruption due to fouling or plugging.

Recovery Boiler C

RB – C is considerably smaller than RB - A and RB - B. RB - C was commissioned towards the end of 2011. The sootblower poppet valve pressures are all set similar to those in RB – A (150 psig, or 10.34 bar g). The boiler has never experienced an interruption due to fouling or plugging.

Since low pressure sootblowers operate at a lower pressure than high pressure sootblowers, the life of sootblower components, such as seals and packing, are expected to be longer. The number of replaced sootblower components subject to wear and tear due to steam pressure, such as seals and packing, is plotted in Figure 14. These components of low pressure sootblowers seem to last longer than corresponding parts used on high pressure sootblowers. It can also be observed that RB - A has seen higher number of parts replaced than RB – B, which may be because RB – A is operating poppet valves at pressures that are ≈ 50 psi (3.45 bar) higher than those at RB - B.

However, the conclusion drawn from the data shown in Figure 14 assumes that these recovery boilers are maintained at the same level, i.e., at the same intensity and by maintenance crews with similar expertise; that may not be the case.

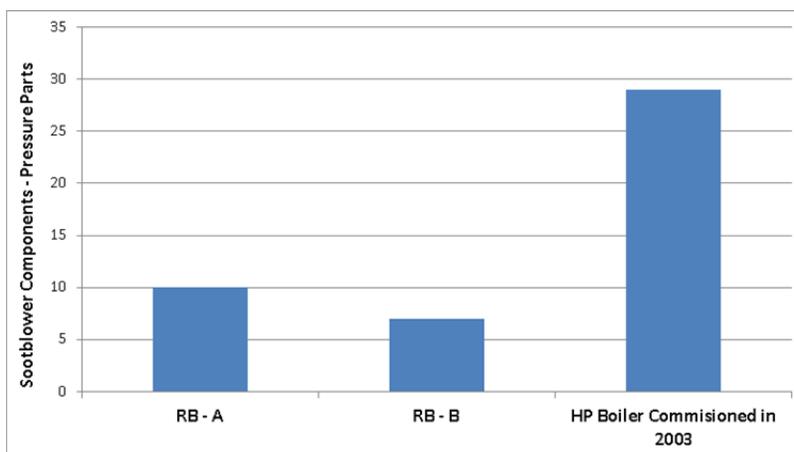


Figure 14. Annual replacement frequency of sootblower pressure parts

SUMMARY

While sootblowing is essential in recovery boilers, its operation can be very costly due to the high consumption of valuable high pressure superheated steam. For the past decade, extensive research and several mill trials have been carried out to explore the feasibility of utilizing less expensive low pressure steam for sootblowing. Results of numerous laboratory experiments, Computational Fluid Dynamic (CFD) simulations, and the mill trials have shown that low pressure sootblowing is technically and practically feasible, provided that the sootblowers are properly designed and that the nozzles are engineered to produce a cleaning power comparable to that of high pressure sootblowers.

6 bar seems to be the lowest feasible low pressure system that could be used in recovery boilers. This argument comes from the fact that, in order to achieve the design total force of 245 lbf with a nozzle pressure lower than 6 bar, one must use a nozzle exit diameter larger than the boiler tube OD of 2.5" (63.5 mm). A sootblower jet that is greater in diameter than the boiler tube OD is unlikely to be effective at removing deposits.

To achieve the same design cleaning force of 245 lbf, 11 bar and 6 bar low pressure sootblowers consume 750 lb/hr (0.34 ton/hr) and 3,400 lb/hr (1.54 ton/hr) more steam, respectively, than a corresponding high pressure sootblower. The results of an economic analysis suggest that by utilizing low pressure sootblowers, pulp mills can generate additional MW power. This is true even if the low pressure sootblower may consume slightly more steam than high pressure sootblowers.

Finally, since their commissioning, all three recovery boilers with low pressure sootblowers have run well with no interruption due to fouling or plugging.

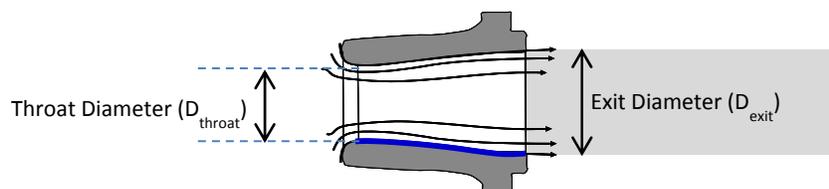
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APPENDIX A – JET TOTAL FORCE CALCULATION

A 3-D Computational Fluid Dynamic (CFD) simulation, with the model validated by laboratory experiments, is by far the best method to calculate jet total force. However, for quick estimation, Equation A1.1, based on a 1-D isentropic flow assumption, can be used:



$$\frac{A_{Exit}}{A_{Throat}} = \frac{1}{M} \left[\left(\frac{2}{k+1} \right) \left(1 + \frac{k-1}{2} M^2 \right) \right]^{\frac{k+1}{2(k-1)}}$$

$$\frac{D_{Exit}^2}{D_{Throat}^2} = \frac{1}{M} \left[\left(\frac{2}{k+1} \right) \left(1 + \frac{k-1}{2} M^2 \right) \right]^{\frac{k+1}{2(k-1)}} \quad (A1.1)$$

M is the jet Mach number at the nozzle exit, and k is the specific heat ratio of steam (k = 1.329)

For a given D_{throat} and D_{exit}, Equation A1.1 can be used to calculate the jet Mach number M at the nozzle exit.

If the D_{throat} and D_{exit} are 1.75" and 2.35", respectively, M will be

$$\frac{2.35^2}{1.75^2} = \frac{1}{M} \left[\left(\frac{2}{1.329 + 1} \right) \left(1 + \frac{1.329 - 1}{2} M^2 \right) \right]^{\frac{1.329 + 1}{2(1.329 - 1)}}$$

$$M = 2.04$$

For a given nozzle pressure (P_{Nozzle}), the jet pressure at the nozzle exit (P_{Exit}) can be calculated using Equation A1.2 (based on 1-D isentropic flow assumption):

$$\frac{P_{Nozzle}}{P_{Exit}} = \left(1 + \frac{k-1}{2} M^2 \right)^{\frac{k}{k-1}} \quad (A1.2)$$

If P_{Nozzle} is 104.7 psia (7.2 bar a), the jet pressure at the exit (P_{Exit}) will be:

$$\frac{104.7}{P_{Exit}} = \left(1 + \frac{1.329 - 1}{2} 2.04^2 \right)^{\frac{1.329}{1.329 - 1}}$$

$$P_{Exit} = \frac{104.7}{8.22} = 12.8 \text{ psia}$$

The jet impact pressure at the nozzle exit (IP_{exit}) can be computed using Rayleigh's pitot formula:

$$IP_{Exit} = P_{Exit} \left[\frac{k+1}{2} M^2 \right]^{\frac{k}{k-1}} \left[\frac{k+1}{2kM^2 - (k-1)} \right]^{\frac{1}{k-1}} \quad (A1.3)$$

With P_{Exit} 12.8 psia and $M = 2.04$, the jet impact pressure at the nozzle exit (IP_{exit}) can be calculated using Equation A1.3:

$$IP_{exit} = 72 \text{ psia (57 psig)}$$

Hence, the jet total force (F_{Jet}) can be calculated by integrating IP_{exit} over the nozzle exit area:

$$F_{Jet} = \int IP_{Exit} dA_{Exit} = 57 \text{ psig} \frac{1}{4} \pi 2.35^2 = 247 \text{ lbf (1100N)} \quad (A1.4)$$

APPENDIX B – NOZZLE EXIT DIAMETER CALCULATION

In this section, we calculate the nozzle exit diameter (D_{Exit}) for a given nozzle pressure (P_{Nozzle}) to achieve a design total force of 245 lbf. As in Appendix A, a 1-D isentropic flow assumption is used for a quick estimation.

Equation A1.4 can be used to calculate the IP_{Exit} required to generate 245 lbf:

$$IP_{Exit} = \frac{F_{Jet}}{\frac{1}{4} \pi D_{Exit}^2} = \frac{245 \text{ lbf}}{\frac{1}{4} \pi D_{Exit}^2} = \frac{312}{D_{Exit}^2} \quad (A2.1)$$

IP_{Exit} in Equation A2.1 and P_{Exit} from Equation A1.2 can be substituted into Equation A1.3 to obtain an equation for D_{Exit} as a function of Mach number (M):

$$\frac{312}{D_{Exit}^2} = \frac{P_{Nozzle}}{\left(1 + \frac{k-1}{2}M^2\right)^{\frac{k}{k-1}}} \left[\frac{k+1}{2}M^2\right]^{\frac{k}{k-1}} \left[\frac{k+1}{2kM^2 - (k-1)}\right]^{\frac{1}{k-1}} \quad (A2.2)$$

$$D_{Exit} = \sqrt{\frac{312\left(1 + \frac{k-1}{2}M^2\right)^{\frac{k}{k-1}}}{P_{Nozzle}\left[\frac{k+1}{2}M^2\right]^{\frac{k}{k-1}} \left[\frac{k+1}{2kM^2 - (k-1)}\right]^{\frac{1}{k-1}}}} \quad (A2.3)$$

The nozzle semi-divergence angle is generally set to 7° to prevent flow separation inside the nozzle [10]. With the length of the nozzle (L) limited to 2" to fit within a typical 4" OD lance, D_{throat} can be written in term of D_{Exit} as follows:

$$D_{Throat} = D_{Exit} - 2 * L * \tan(7^\circ) \quad (A2.4)$$

Note that for older sootblower design, the lance OD is typically smaller (i.e., 3.5"). In this case the design of L will be lower than 2".

D_{throat} from A2.4 can be substituted into Equation A1.1 to arrive at a second equation for D_{exit} as a function of M:

$$\frac{D_{Exit}^2}{(D_{Exit} - 2 * L * \tan(7^\circ))^2} = \frac{1}{M} \left[\left(\frac{2}{k+1} \right) \left(1 + \frac{k-1}{2} M^2 \right) \right]^{\frac{k+1}{2(k-1)}} \quad (A2.5)$$

Equations A2.3 and A2.5 have two unknowns D_{exit} and M. These two equations with two unknowns can be solved for D_{exit} and M.

If $P_{Nozzle} = 4.5$ bar g (65.3 psig), Equations A2.3 and A2.5 yield $D_{exit} = 2.6''$ and $M = 1.85$. This value of D_{exit} , required to achieve the design total force of 245 lbf, is greater than the boiler tube OD of 2.5" (63.5 mm).