Q. **Reference: Improve Boiler Load Capacity – Units 1, 2 and 3, Holyrood, June 1, 2018, Appendix A, B&W Engineering Study Report, Page A14.**

"The current fuel oil atomizing temperature (approx. 187 F) is lower than required for optimal combustion. It is recommended to increase firing temperature to 220-225 F to ensure proper combustion with the current range of oil viscosities."

Has operating at these lower temperatures contributed to the abnormal ash deposits encountered in recent years and does Hydro intend to follow the recommendation to operate at the higher temperature?

A. Of the three units, only Unit 3 has had this minor differential in fuel oil atomizing temperature, yet the hard ash exists on all units. Therefore, the implication of this minor difference is not deemed material to the ash build up. The resulting change in Unit 3 atomization temperature is about 5 degrees Celsius, and the current temperature has been in use for 15 years. Hydro receives advice every season on improving its operating parameters and intends to adjust the operating temperature this season.

For information, the referred statement from the B&W report has been removed in Revision 3 of the report Performance Study Unit Capacity Limitations, which was issued on June 8, 2018. This revision to the report was made to address a misunderstanding regarding the atomization temperatures maintained at Holyrood. The new statement, on page 14 of 90, is as follows:

> *"The current fuel oil atomizing temperature (approx. 187 F) is at times lower than required for optimal combustion. It is*

recommended to increase firing temperature to achieve target oil viscosities as discussed in Section 6.1."

The updated report is attached as NP-NLH-005, Attachment 1.

Thermal Power Department

Technical Services

Engineering Study Report

Customer:	**Newfoundland and Labrador Hydro (NLH)**
	Holyrood Units #1, #2, #3
Subject:	**Performance Study**
	Unit Capacity Limitations
Ref No:	B&W Project 312C
	Rev 03, June 8 / 2018
Prepared By:	Brian Jordan P. Eng Project Engineer
Reviewed By:	Malcolm Mackenzie, P. Eng.

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1 INTRODUCTION

The three oil fired units at Newfoundland and Labrador Hydro's Holyrood station are currently not capable of generating their rated megawatt outputs. Newfoundland and Labrador Hydro (NLH) requested B&W to perform this engineering study to identify the causes of the current limitations and make recommendations to return the units to full load capability. The B&W proposal for this study was B&W reference TP001082 issued on 21 November 2017. A two stage approach was proposed. The first stage identifies the causes of load limitations and the second stage focuses on the steam generator heating surface effectiveness. This report summarizes the results of both stages.

The Unit #1 and #2 boilers at Holyrood are Combustion Engineering (CE) units built in the late 1960's. The Unit #3 boiler was provided by Babcock & Wilcox Canada (B&W) in 1979. All three boilers are pressurized (i.e. forced draft fans only). The turbine-generator sets for all three units were supplied by Hitachi Ltd. The three units were originally rated at 150 MW (Gross). Units #1 and #2 were up-rated to 174.2 MW in 1988 and 1989 respectively.

The maximum unit load for Units #1 and #2 was limited to 133 and 125 MW (gross) respectively by furnace pressure per the January / February 2018 operating data considered in this study. The maximum load for Unit #3 was limited by FD fan capacity to128 MW per January 2018 operating data.

The load limitation for Unit #1 and #2 is maximum furnace pressure thus this study focuses on the factors which affect furnace pressure for these units. The load limiting factor for Unit #3 is FD fan capacity so the focus is on fan capacity.

The common fuel oil supply system is also considered with respect to issues that affect boiler performance.

2 EXECUTIVE SUMMARY

Recent losses in the capacity of the three Holyrood units are primarily a result of:

i) Increases in air and flue gas pressure drops across the cold end boiler heating surfaces (economizers and air heaters) due to oil firing deposits (fouling) on these surfaces. These deposits form predominantly during periods of low load and startup when the heating surfaces are cold and combustion efficiency is low.

ii) Degradation of unit heat rate which increases the required heat input per MW. These increases lead to increased furnace pressure and FD fan loading in turn.

Units #1 and #2 are currently load limited by the maximum allowable furnace pressure. Unit #3 is load limited by the FD fans.

Reductions in maximum load capability for Units #1 and #2 have been present since 2015/2016. The reduction in maximum load for Unit #3 occurred relatively quickly in the Oct 2017-Jan 2018 time period.

Due to excessive deposition, all three units experience increased draft losses. The air heaters on all three units are affected. Units 1&2 are equipped with extended surface (finned) economizers which also experience increased draft losses. Replacing or cleaning of fouled heat transfer surfaces to ‘as new’ condition (if possible) will restore the design maximum unit load capability.

If unit load capability is restored by cleaning and/or replacing heat transfer surfaces, reoccurrence of unit de-rates caused by fouling can be prevented by:

- Ensuring air heater Average Cold End Temperatures (ACET) are maintained above 212 F (100 C) at all times.

- Reinstating use of the fuel MgO dosing system

- Increasing the fuel oil atomizing temperature to ensure proper atomization and combustion.

- Ensuring sootblowing steam is dry

The key findings of this study are outlined below.

2.1 Units #1 and #2

The maximum output of Units #1 and #2 is currently limited by the maximum allowable furnace pressure. Maximum furnace pressure is established by the boiler manufacturer according to the structural design of the boiler and furnace. Unit #1 was limited to 133 MW on Jan 18, 2018 at a furnace pressure of 17.9” wg. Unit #2 was limited to 125 MW on Feb 2, 2018 at a furnace pressure of 19.9” wg. Loads of 170 MW were last achieved in Jan 2015 and Oct 2016 for Units #1 and #2 respectively.

The operating furnace pressures are significantly higher than design primarily due to the combination of:

a) Higher than design air heater and economizer pressure drop due to fouling of the heating surfaces
b) Higher than design unit heat rate due to reduced boiler efficiency and increased Turbine Generator (T-G) heat rate.
c) Higher than design air flows

The potential increases in unit load as limited by furnace pressure that would occur if the above issues are corrected are illustrated in Figures #1 and #2. The gains associated with restoring economizer / air heater pressure drops are based on new heating surfaces or surfaces restored to ''as new' condition and are thus best case scenarios.

Unit heat rate could be restored by restoring T-G efficiency, correcting lower than design hot reheat steam temperatures, and restoring air heater / economizer heat transfer efficiency (Boiler efficiency).

The higher than design air flows are due to underestimation of the combustion air quantity as indicated by the OEM boiler supplier data sheets (Appendix 8.1) and are therefore not considered 'correctable'.

Figure 1 Unit #1 Potential MW Output Increases

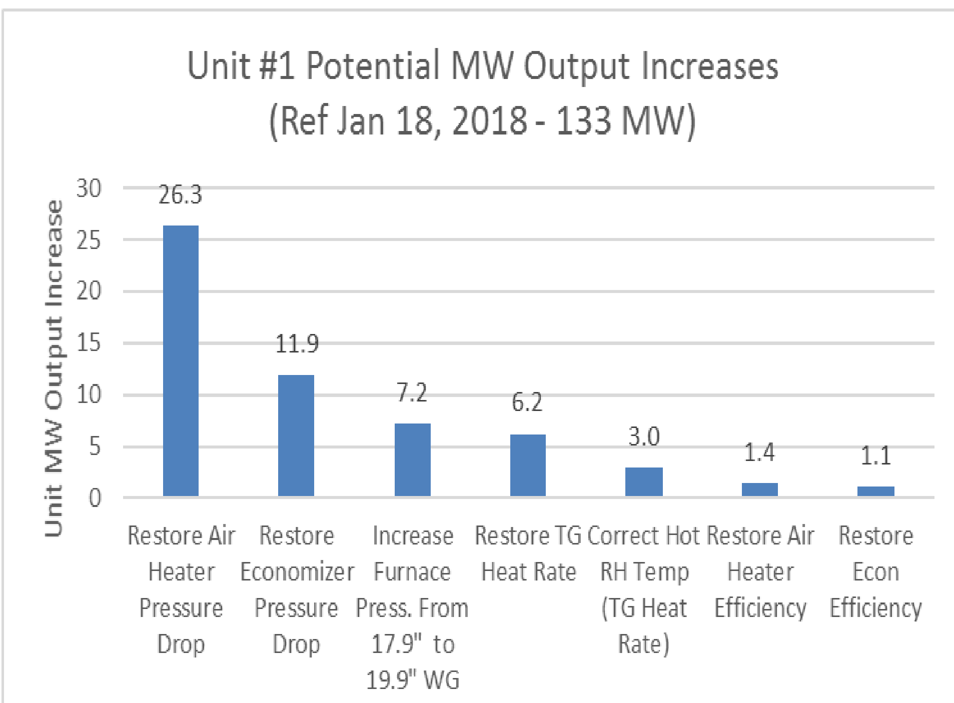
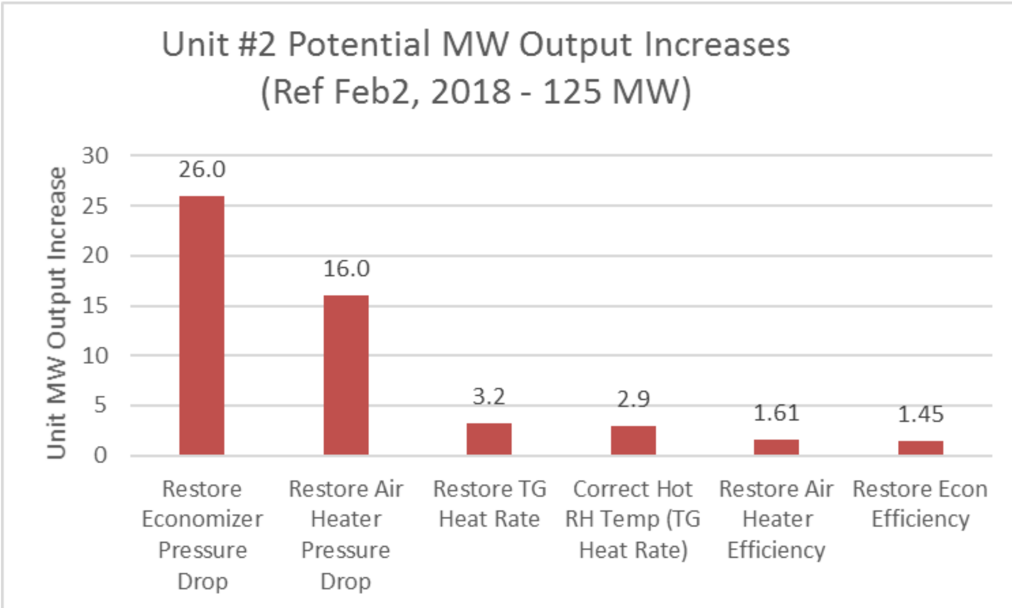


Figure 2 Unit #2 Potential MW Output Increases



If both air heater and economizer pressure drops are restored, the full rated 174.2 MW will be achievable on both units without exceeding the current 20” WG furnace pressure alarm point limit. According to site reports, cleaning of these heating surfaces has proven very difficult in the past. Unless more effective methods can be employed such as chemical cleaning the most effective means of reducing furnace pressure would be to replace the fouled air heater elements. Replacement of economizer surfaces would very likely not be economically viable.

Less significant increases in maximum unit load capability are possible by restoring turbine / generator (T-G) heat rate.and/or restoring the heat transfer effectiveness of the boiler heating surfaces. Results of the ‘Stage 2’ study indicate poor heat transfer effectiveness of the air heaters and economizers. It is important to note that if pressure drops as above are restored by surface cleaning or replacement, a significant portion of the MW gains from increased boiler efficiency will also be realized along with the associated fuel savings.

Unit # 1 was operating at a furnace pressure of 17.9” wg on January 18, 2018, reportedly load limited by furnace pressure. The reason for this lower operating pressure at that time is unknown. If the maximum operating furnace pressure is increased to 19.9” as per Unit #2, an increase in maximum load of 7.2 MW would be realized.

The reheaters on both units are underperforming significantly. While the cause of poor air heater and economizer performance is clearly fouling as evidenced by high pressure drops, the cause of poor reheater heat transfer performance is not known and should be investigated. Sootblower usage patterns and blowing pressures may need to be adjusted to improve effectiveness. Poor reheater heat transfer effectiveness reduces unit efficiency (and MW output) on four fronts:

a) Low hot reheat temperature (increased T-G heat rate)
b) High burner tilts (less furnace effectiveness – loss of boiler efficiency)
c) High superheat sprayflows (increased T-G heat rate).
d) Increase in stack temperature. (loss of boiler efficiency)

Of the above, item a) is the most significant.

2.2 Unit #3

Unit #3 is load limited by the current capability of the FD fans. Maximum load dropped from 150 MW in October 2017 to 128 MW on January 4, 2018 as air heater pressure drop increased. The pressure drop increased most significantly during lower load operation (less than 100 MW) and when air heater Average Cold End Temperature was less than 100 C (212 F).

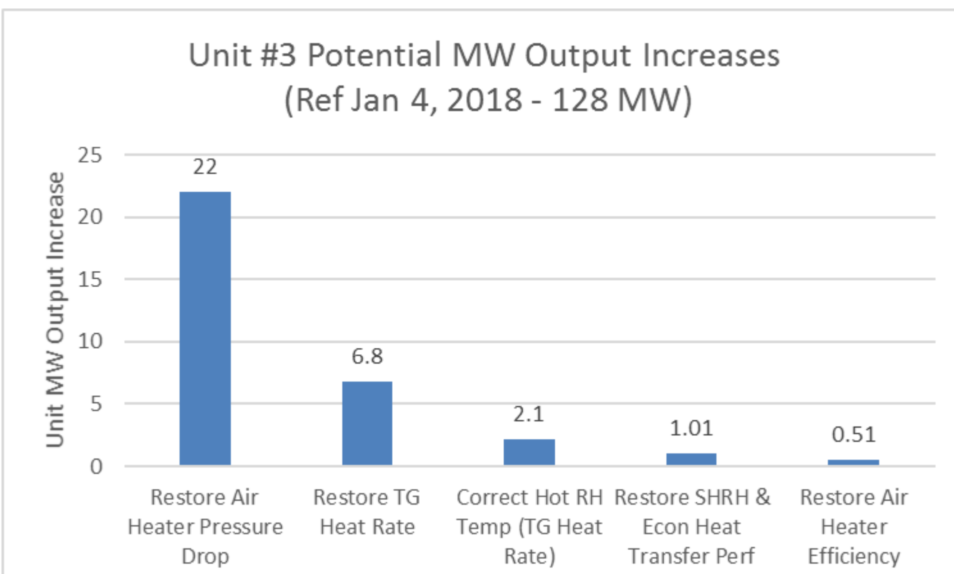
The fan VIV’s have been restricted to 54/70% open on the east/west fans respectively due to inlet ducting vibration which occurs at higher openings under some operating conditions.

Without this restriction, full load operation would have been attainable on January 4, 2018 when load was limited to 133 MW. An inspection and test program should be implemented to determine how the full fan capacity can be restored.

The required FD fan duty is higher than design primarily due to higher than expected air heater pressure drop and higher than design Unit heat rate (lower than design unit efficiency). Full load operation would be restored given the current FD fan VIV restrictions if the fouled hot end air heater elements are replaced with the proposed “ARVOS” elements and if the existing cold end baskets are clean and in good condition.

With reference to the Jan 4, 2018 operating point (128 MW), increases in unit load capability as per Figure #3 would be possible for fixed fuel input. The largest contributor to unit efficiency reductions is turbine – generator inefficiencies. The largest boiler related contributor to the increase in unit heat rate is low hot reheat temperature.

Figure 3 Unit #3 Potential MW Output Increases



The heat transfer effectiveness of the Unit #3 superheater and reheater declined significantly during the time period from Oct 2017 to Jan 2018. These surfaces should be inspected for cleanliness to determine the cause of this decline. Sootblowing patterns and/or blowing pressures may need to be revised to improve cleanliness.

2.3 Fuel Related Issues (Common Units 1,2,3)

The quality of fuel oil has improved significantly in recent years. A significant reduction in fuel oil Vanadium and Sulphur content occurred in 2006. These improvements would be expected to reduce the tendency towards boiler cold end (air heater and economizer) fouling and boiler corrosion. From a combustion standpoint, the currently utilized fuels are very close to the original Unit #3 design fuel.

The current fuel oil atomizing temperature (approx. 187 F) is at times lower than required for optimal combustion. It is recommended to increase firing temperature to achieve target oilviscosities as discuused in Section 6.1. The MgO additive system was taken out of service in 2014 and reductions in unit load capability for Units #1 and #2 started to occur in 2015-2016 and Unit #3 in late 2017. This system should be placed back into service and the oil dosed at a rate of 1 lb. MgO per lb. V2O in the fuel oil.

The Unit #3 air heater fouled rapidly between Oct 2017 to Jan 2018. During this time period, air heater pressure drop increased most notably during periods of both low load operation and low ACET. When ACET was maintained above 212 F there was no significant increase in pressure drop. It is recommended that air heater ACET is maintained at a minimum of 212 F for all three units.

For Unit #3, the combination of low load operation (possibly poor combustion due to low atomizing temperatures), the lack of MgO additives, and low ACET is the most likely cause air heater fouling that occurred between October 2017 and January 2018.

Fouling in the Unit #1 and Unit #2 air heaters and economizers occurred between 2015-16 and 2018. The operating conditions during which this fouling occurred is unknown. It is most likely that the economizer fouling occurred start-up operation and the air heater during low load and/or start-up operation.

3 CONCLUSIONS AND RECOMMENDATIONS

The conclusions and recommendations of this study are summarized below:

3.1 Units #1 and #2

3.1.1 Conclusions

a) The current maximum achievable load of Units #1 and #2 is limited by furnace pressure due to the combination of the following factors:
    i. The draft loss across boiler surfaces is higher than design, most notably the economizer and air heater
    ii. Unit efficiency is lower than design
    iii. The calculated fuel air flow requirements (per unit fuel flow) are higher than original design
b) The air heater and economizer pressure drops have increased significantly between the 2015/16 and 2018
c) Pressure drops across the superheater and reheater are significantly higher than design but are not a major contributor to higher than design furnace pressure.
d) Reheater heat absorption is lower than design as evidenced by lower than design hot reheat steam temperatures. Low hot reheat temperatures are leading to an up to 1.5% increase in TG heat rate.
e) The current largest contributors to higher than design furnace pressures and unit derating are:
    i. For Unit #1, high air heater pressure drop

ii. For Unit #2, high economizer pressure drop

f) Restoring the air heater and/or economizer pressure drops to original design would increase maximum load as limited by furnace pressure per the following table: (Note that restoring both components results in increase above that of individual components- if just one component is restored, furnace pressure is still limited by restriction in the other)

MAXIMUM LOAD AS LIMITED BY FURNACE PRESSURE			
		Unit #1	Unit #2
Maximum Load Per 2018 Data	MW	133	125
Increase Maximum Furnace Pressure up to 19.9" WG (Unit #1)	MW	140	125
Restore Design Air Heater Pressure Drop	MW	159	141
Restore Design Economizer Pressure Drop	MW	145	151
Restore Both Economizer and Air Heater	MW	175	175

g) Improved heat transfer and boiler efficiency will follow restoration of heating surface cleanliness. FD fan power consumption will also be reduced.

h) Alternate methods of economizer / boiler surface cleaning such as explosives or acoustic shock – blast methods could be considered if it is not possible to clean these surfaces by conventional means.

i) Maximum boiler load as limited by furnace pressure may be increased if modifications/repairs to the turbine/generator set are made to improve heat rate.

j) It may be possible to increase the current furnace pressure alarm and trip points. The original boiler supplier could advise if this is possible.

k) The heat transfer performance of the economizer and air heater on both units is significantly lower than design, reducing boiler efficiency significantly

l) The heat transfer performance of the reheater heating surfaces is significantly lower than design, reducing Turbine-Generator efficiency significantly and boiler efficiency.

m) Removal of air heater heating surfaces is not recommended due to the negative effect on combustion efficiency and structural limitations of downstream flues/stack.

n) Partial removal of economizer heating surfaces to reduce pressure drop should be considered as a last resort only due to negative effect on downstream boiler structure and boiler performance.
o) Increasing the maximum furnace pressure of Unit #1 to 19.9” as per Unit #2 operation will account for 7.2 MW of additional unit output.

3.1.2 Recommendations

a) Reduce the pressure drop across the air heaters and/or economizers by cleaning and/or replacement of heating surfaces. Prioritize this work as follows:
    1) Unit # 1 air heater
    2) Unit #2 Economizer
    3) Unit #2 Air Heater
    4) Unit #1 Economizer
b) If economizer and boiler surfaces cannot be cleaned by ‘conventional’ methods investigate alternative methods such as explosive or acoustic shock-blasting
c) Ensure that the steam supply to economizers and air heater sootblowers is dry
d) Determine if the current furnace pressure alarm/trip setpoints can be increased. (By original boiler supplier)
e) Inspect the reheaters to determine the cause of low reheater heat transfer performance.
f) Use burner tilts within manufacturers recommended range as required to increase hot reheat temperatures
g) Consider turbine – generator – condenser upgrades which would improve heat rate.
h) Consider increasing the maximum furnace operating pressure of Unit #1 to 19.9” wg
i) Consider increasing the furnace pressure alarm pressures.

3.2 UNIT #3

3.2.1 Conclusions

a) The current maximum achievable load Unit #3 is limited by the capacity of the FD fans due to the combination of the following factors:
    i. The FD fans capacity are currently not operated at their maximum capacity
    ii. Air heater leakage rates are up to 3 times higher than design
    iii. Air heater pressure drops are 3 to 4 times higher than design
    iv. Unit heat rate approximately is approximately 10% higher than design due to lower than design boiler efficiency and higher than design Turbine Generator Heat Rate
    v. Operating excess air to burners approximately 2% higher than design

b) If the existing FD fan capacity was unrestricted, the full 150 MW unit output could have been attained for the January 4, 2018 operating conditions when maximum load was 128 MW.
c) Replacing the air heater hot end baskets will restore the unit full load capability of 150 MW if the cold end baskets to be re-used are clean and in good condition.
d) The combustion air flow requirement of the fuel oil currently utilized at site is very close to design on a lb/btu input basis.
e) The calculated fuel flows based on unit PI data and the measured fuel flow are both significantly higher than expected confirming that unit efficiency is lower than design. The calculated and measured fuel oil flows are within 3% of each other.
f) Removal of air heater heating surfaces is not recommended due to the negative effect on combustion efficiency and structural limitations of downstream flues/stack.

3.2.2 Recommendations

a) Establish if the current operating restrictions placed on the FD fans can be removed.
    i. Perform an operating test with increased FD fan VIV position and RPM at high load to determine current operating limitations (duct vibration?)

ii. Inspect the FD fan internals, instrumentation, inlet/outlet ducts and correct any anomalies which may lead to operating problems.
iii. Perform an FD fan test after inspections, including inlet/outlet pressure measurements and inlet airflow measurements.

b) Refurbish the air heater seals to reduce leakage and FD fan power consumption.
c) Clean or replace air heater heating elements which are leading to the high pressure drop and load limitations.
d) Ensure that the steam supply to economizer and air heater sootblowers is dry.
e) Consider turbine – generator - condenser upgrades / repairs which would improve TG heat rate

3.3 Fuel Related Issues (Common Units 1,2,3)

3.3.1 Conclusions

a) From a combustion and heating value standpoint, the fuel oil currently utilized is very close to the original Unit #3 design fuel.
b) Fuel oil Sulphur and Vanadium content have been reduced significantly since 2009.
c) Fouling of the Holyrood units leading to reduced maximum load capability has occurred between 2015 and 2018, following the discontinuation of fuel oil MgO injection.
d) The unit #3 operating conditions between October 2017 and January 2018 show increasing air heater pressure drop occurs at reduced loads, and when air heater ACET drops below 212 F.
e) Atomizing fuel oil temperatures must be sufficient to ensure proper atomization / combustion of the range of fuels currently burned (Up to 200 SFS @ 122 F)

3.3.2 Recommendations

a) Recommission the fuel oil MgO injection system and inject MgO into the fuel oil supply at a rate of 1 lb. MGO per lb. V2O in fuel oil.
b) Maintain a minimum air heater ACET of 212 F
c) Maintain atomizing oil atomization as follows for fuel oil viscosities up to 200 SFS@122 ^{o}F

a. Units #1 and #2 that temperature required to achieve 100 SSU or in the absence of viscosity data 230 ^{o}F
b. Unit #3: that temperature required to achieve 135 SSU or in the absence of viscosity data 225 ^{o}F

4 UNITS #1 and #2

4.1 Unit Description and History

The Unit #1 and #2 boilers were supplied by Combustion Engineering Canada in 1969. The boilers supply main and reheated steam at a design 1000 F to Hitachi steam turbines. Air is supplied by two Forced Draft fans through steam coil air heaters and regenerative air heaters to tilting tangentially fired burners in the furnace. Products of combustion leaving the furnace pass through a parallel flow secondary superheater, followed by a counter flow reheater, primary superheater, and finned tube economizer before entering two Ljungström regenerative air heaters.

The units were uprated to deliver 174.2 MW in 1987. Four rows of primary superheater were removed and tube material upgrades were made to the secondary superheater as part of the uprate. The unit was originally designed to control steam temperatures with the combination of flue gas recirculation and burner tilts. The gas recirculation fans have been removed from service.

Neither unit has been capable of operating at loads above 170 MW in recent years. The most recent time period that operating data was available for 170 MW was February 2015 for Unit #1 and October 2016 for Unit #2. The maximum load achievable is currently limited by maximum furnace pressure which has an alarm setpoint of 20” wg. The units will trip if furnace reaches 25” wg. Operators currently maintain furnace pressure below the 20” wg alarm point.

4.2 Basis of Study

This study is based on information provided by NLH as outlined below.

4.2.1 Fuel

NLH supplied a spreadsheet summary of the analysis of fuel oil deliveries to Holyrood between 1997 and 2017. Heating value, density, and trace element composition was included in this spreadsheet. A discussion of the fuel characteristics is included in a following section of this report.

4.2.2 Base Heat Balance Information

The expected original design plant operating information for the uprated unit was supplied by NLH as follows:

- Alstom letter to NF Power “Boiler Predicted Performance Data for Boiler #1 & 2” dated Aug 03, 2000. This document is the predicted boiler performance in the “Uprated” condition
- Turbine heat balance conditions as outlined in document “TIR# 10236-893A, UPRATE” Dated 8/5/88.
- The original Combustion Engineering ‘Contract Data Sheet’ (Contract 68119)

These documents are included in Appendices 8.1 and 8.2 for reference.

4.2.3 Unit Operating Data

B&W requested historical operating data representative of unit operation which was not restricted by furnace pressure and current restricted operating data. In response, ‘PI’ plant historian data was provided by Newfoundland and Labrador Hydro (NLH) in spreadsheet form for the two units at two time periods as outlined in Table 1.

Table 1 Units #1 and #2 Operating Data Conditions

	Unit 1		Unit 2	
Date	Jan 18, 2018	Feb 9, 2015	Oct 18, 2016	Feb 2, 2018
Unit Output MW	133	169	170	125
Operating Condition	Load Limited by Furnace Pressure @ 17.9" WG	Not Restricted	Not Restricted	Load Limited by Furnace Pressure @ 19.9" WG

It is not known why furnace pressure was limited to 17.9" wg on Unit #1 in January 2018. On possibility is that unstable furnace pressures may have led operators to reduce load to keep furnace pressure out of alarm.

4.2.4 Unit Physical Arrangement

NLH provided boiler general arrangement drawings defining the boiler heat transfer surface arrangement.

4.2.5 Heat and Mass Balance Calculations

B&W Single Heat and Material Balance Program – P08475 was used to calculate flue gas flow, flue gas analysis, combustion air flow, and boiler efficiency based on the fuel analysis, and the operating steam/water, and the air/gas boundary conditions.

4.2.6 Boiler Surface Heat Transfer Effectiveness Calculations

The boiler convective component heat transfer effectiveness (Kf) calculations were performed using B&W's proprietary convective surface heat transfer program "P140". The inputs to this program are the FEGT, the flue gas flow / composition from P08475, and the boiler tube bank heating surface geometry.

The thermal performance of the boiler heat transfer components (superheater, reheater, and economizer) heating surfaces is characterized by B&W as 'Kf' factors. Kf is calculated by P140 based on the operating data, (component outlet gas temperature and calculated flue gas flow, steam or water inlet and outlet conditions and flow). The component Kf factor is the ratio of 'test' gas side heat transfer conductance to 'expected' gas side conductance:

$$Kf = Ug_{test} / Ug_{\exp}$$

The tube bank geometry and flue gas flow are known. P140 calculates the expected gas side heat transfer conductance Ug_{exp} (Btu/hr/ft^2/^{o}F) on this basis using the standard Kf. For oil fired units, the expected Kf is 1.0 for superheater, reheater, and economizer surfaces. The overall component heat absorption is calculated from the measured steam or water inlet/outlet conditions (enthalpies) from which a test gas side conductance is determined (Ug_{test}). For oil firing Kf less than 1.0 indicates the heating surfaces are absorbing less heat than expected due to fouling, gas bypassing, unexpected gas flow patterns, etc.

The flue gas temperatures throughout the boiler are calculated by heat balance starting with the measured temperature at the economizer outlet.

4.2.7 Furnace Heat Transfer Effectiveness Calculations

The actual Furnace Exit Gas Temperature (FEGT) is calculated by heat balance around the convective heating surfaces. The difference in temperature between the calculated FEGT and the FEGT as predicted by Alstom is an indication of relative furnace effectiveness. An actual FEGT higher than the expected FEGT indicates underperforming (dirty) furnace surfaces (or higher than expected burner tilts).

4.2.8 Air Heater Heat Transfer Effectiveness Calculations

B&W relies on air heater vendors calculations to predict thermal performance of regenerative air heaters. Air heater heat transfer effectiveness Kf values are thus calculated based on the ratio of the actual heat transfer to the air heater vendors heat transfer adjusted to the actual operating conditions. The 'base' Kf factor to match air heater vendor predicted performance is set to 1.0 thus a calculated Kf value of less than 1.0 indicates heating surfaces are under performing. For Units #1 and #2 the base performance operating condition was taken from the Alstom August 2000 predicted performance data 'MCR' load case.

4.3 DISCUSSION OF RESULTS – Units #1 and #2

Both Units #1 and #2 are currently limited by the maximum allowable furnace pressure. Furnace pressure is a function of the flow resistance (geometry, cleanliness) of the downstream boiler components and the flue gas flow through these components. Flue gas temperature is also a factor, (higher temperatures = higher resistance at a given mass flow) but this effect is small relative to resistance and flue gas flow and is not considered in this study.

4.3.1 Review of Operating Data

The 'PI' system operating data used in this study analysis is generated by the plant permanent instrumentation. It is adequate for detecting trends but not always accurate for measuring 'bulk flow' parameters such as flue gas and air temperatures in large ducts were temperature stratification is expected. As such, the analysis which is based on plant instruments can be considered accurate from a relative standpoint (i.e. to illustrate trends) only. Evaluation of absolute plant performance requires calibrated instruments and air/gas temperature grids in large flues and ducts.

In general, the most accurate plant instruments are those indicating the conditions of major unit inputs/outputs (i.e. fuel flow, MW output), and the 'terminal point' connections between boiler and turbine cycle (i.e. feedwater flow, steam temperatures and pressures). Steam flow as indicated by HP turbine pressure is not considered as accurate as feedwater flow thus steam flow was calculated from the measured (*feedwater flow – blowdown flow – Aux steam flow*). Reheat steam flow was calculated based on *(calculated steam flow - HP turbine 'leakages' - #6 feedwater heater steam flow)*. The HP turbine leakages were taken from the Hitachi 1988 turbine heat balances, and the #6 feedwater flow is calculated by heat balance around the heater based on operating data.

The heat transfer effectiveness analysis (Kf study) requires steam and water – side enthalpies in and out of each boiler component. For units with superheat attemperators, attemperator water flow and attemperator inlet steam temperature are required to determine the heat absorption of the primary and secondary superheater. The measured attemperator steam outlet temperature is prone to reading low due and is not considered accurate. The Units #1 and #2 attemperator inlet steam temperatures are not available, thus only total superheater surface effectiveness (Kf) can be evaluated.

The effect on calculated Kf values of the above factors can be significant. The accuracy of the calculated Kf values would not be expected to be better than +/- 0.1.

4.3.2 Unit Heat Rate

The resistance (Pressure drop) of boiler components and thus the furnace pressure is proportional to the square of flue gas flow. The required flue gas flow is a function of the required unit MW output, the unit efficiency, the fuel theoretical air flow requirements, and the excess air required for complete combustion. Unit efficiency is the combination of Turbine-Generator (TG) efficiency (Heat Rate) and boiler efficiency. These parameters are shown in Table 2.

Table 2 Units #1 and #2 Heat Rate Effect on Furnace Pressure

UNIT HEAT RATE EFFECT ON FURNACE PRESSURE						
		Design	UNIT #1		UNIT #2	
Date		Uprate 2000	Feb9, 2015	Jan18, 2018	Oct18,2016	Feb2,2018
Unit Output	MW	174.2	169.5	132.6	170	125.2
TG Heat Rate	Btu/kWhr	7982	8541	8540	8156	8377
Boiler Efficiency	%	90.01* 88.06**	85.1	86.51	85.14	86.07
Unit Heat Rate	Btu/kWhr	9053	10037	9871	9579	9733
Fuel Theoretical Air	Lb/10,000 btu	6.865	7.407	7.407	7.407	7.407
Excess Air	%	5	5.4	8.1	3.6	7.3

*Design boiler efficiency per Alstom data based on 18,600 btu/lb fuel, steam coil and oil heating steam provided by external supply.

**Efficiency based on 18,450 Btu/lb fuel, steam coil and oil heating steam provided by unit (For direct comparison to B&W calculations – this study)

Significant observations from Table 2:

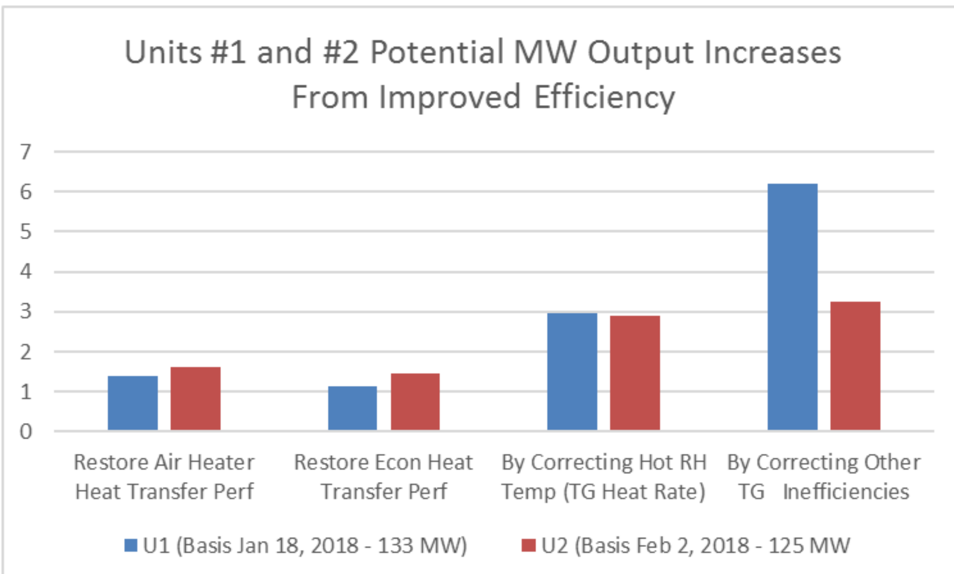
- The TG heat rates are both higher than design.
  - Unit #1 approximately 7% higher
  - Unit #2 approximately 2-5% higher

- Boiler efficiency is approximately 4% lower than design with heat credits (aux steam from ‘outside’), approximately 2% lower than design without heat credits.

- The theoretical combustion air used by Alstom is inconsistent with the fuel analysis. The airflows reported by Alstom in the updated expected performance are not consistent with the combustion airflow required for heavy fuel oil. Per the Alstom 2000 uprate letter data sheet, the MCR theoretical airflow used was 6.73 lb air per 10,000 btu input i.e. (air heater outlet airflow / excess air) / (fuel flow * 18,600 Btu/lb) / 10,000. Heavy fuel oils typically require theoretical combustion air 7.4 to 7.6 lb. per 10,000 Btu input. My calculations are based on a theoretical air requirement of 7.35 lb per 10,000 Btu thus my calculated airflows are higher than the Alstom airflows. This additional airflow contributes to higher furnace pressure.

Figure 3 illustrates the MW gains that would be expected for a fixed firing rate if the original design T-G heat rate and boiler efficiency were restored (ref the 2018 operating data)

Figure 4 Units #1 and #2 Potential MW Output Increases from Improved Efficiency



A significant portion of the increased T-G heat rate is due to lower than design hot reheat steam temperatures. Unit #2 was operating at 898 F at the turbine in February 2018 leading to a T-G heat rate increase of approximately 1.5%. The reheaters on Units #1 and 2 should be inspected to identify the cause of the performance shortfall.

The net effect of the increased unit heat rate, the higher theoretical air, and change in excess air is an increase in unit flue gas flow for a given unit MW output. The increased flue gas flows by themselves are responsible for a significant increase in furnace pressure (reference original design draft losses). The MCR expected furnace pressure per the Alstom data is 11.3" wg @ 174.2 MW. The increased flue gas flow associated with increased unit heat rates alone increases expected furnace pressure to 13.9" wg for Unit #1 and 12.7" wg for Unit #2.

4.3.3 Fuel Oil Flow

The measured and calculated fuel oil flow in relation to expected oil flow provide an indication of unit heat rate. Table 3 illustrates these quantities for the two units and test times. The Expected / Calculated Oil Flows are consistently above 1.0, which is an indication of higher than design unit heat rate.

Table 3 Fuel Oil Flow Calculated/Expected Units #1 and #2

Fuel Oil Flow Calculated/Expected Units #1 and #2					
Unit		1		2	
Date		Feb 9 2015	Jan 18 2018	Oct 18, 2016	Feb 2, 2018
Unit Output	MW	170	133	170	125
Expected Oil Flow (18,450 btu/lb HHV, HR and Blr Efficiency)	Lbs/hr	82176	64175	82383	60487
Calculated Oil Flow	Lbs/hr	92392	71110	88436	66170
Calculated/Expected Oil Flow	-	1.12	1.11	1.07	1.09
Plant Measured Oil Flow	Lbs/hr	90628	68157	90466	65602
Oil HHV	Btu/lb	17,193 - 18,702 (2015-2017 Deliveries)			

4.3.4 Restore Unit Output by Reducing Flue Gas Pressure Drops

Furnace pressure is driven by the pressure drops of the 'downstream' boiler components. These are the superheater/reheater, economizer, air heater, flues to stack. The predicted and actual pressure drops (i.e. the furnace pressure) for Units #1 and #2 are illustrated in Figures 4 and 5.

Figure 5 Unit #1 Furnace Pressure Design Vs Actual

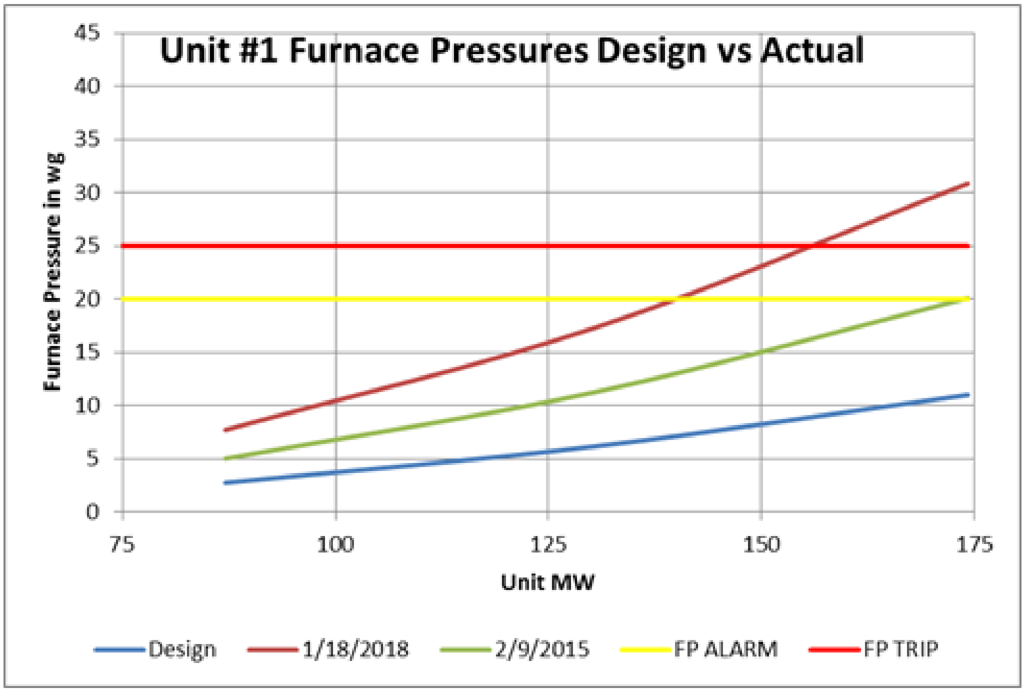
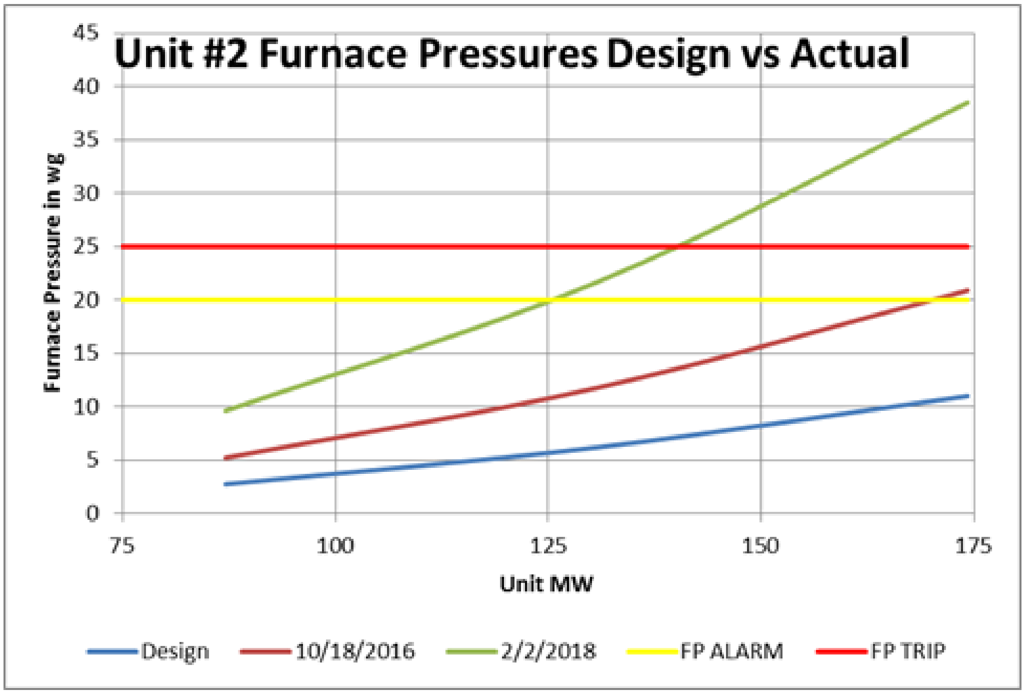


Figure 6 Unit #2 Furnace Pressure Design Vs Actual



The original pressure drops were almost doubled at the times when full load was nearly (170 MW) achieved in the 2015/2016 data with furnace pressures approaching the 20” wg alarm. Between that time and 2018, pressure drops increased even further, predominantly due to increases in economizer and regenerative air heater pressure drops. As these pressure drops increased, unit load was restricted in step. It is not known if the pressure drop increases were gradual or associated with particular operating scenarios. A review of all operating data between 2015-2016 and current would be required to reveal trends.

The predicted, 2015/2016, and current flue gas pressure drops by boiler component are shown in Figures 6 and 7. Pressure drops were prorated from actual operating conditions to 174.2 MW for illustration. The 174.2 MW output is not currently achievable on either unit with the current furnace pressure constraint. For Unit #1, the air heater is the largest contributor to current total pressure drop. For Unit #2, the economizer is currently the largest pressure drop contributor.

The superheater and reheater pressure drops are also significantly higher than design, indicating fouling in these components and / or tube misalignment. The magnitude of this contribution to furnace pressure is small relative to the air heater and economizer. Hot reheat temperatures are currently much lower than design, which combined with the high pressure drop suggests that the cleanliness of these surfaces is also poor.

Figure 7 Unit #1 Pressure Drops Prorated to 174.2 MW

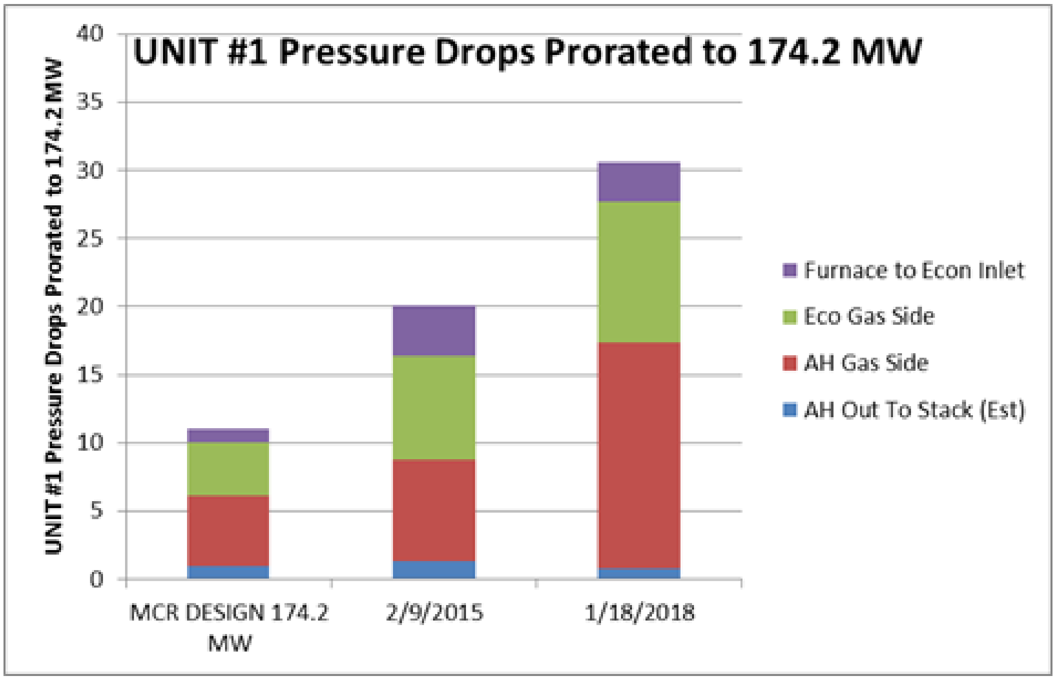
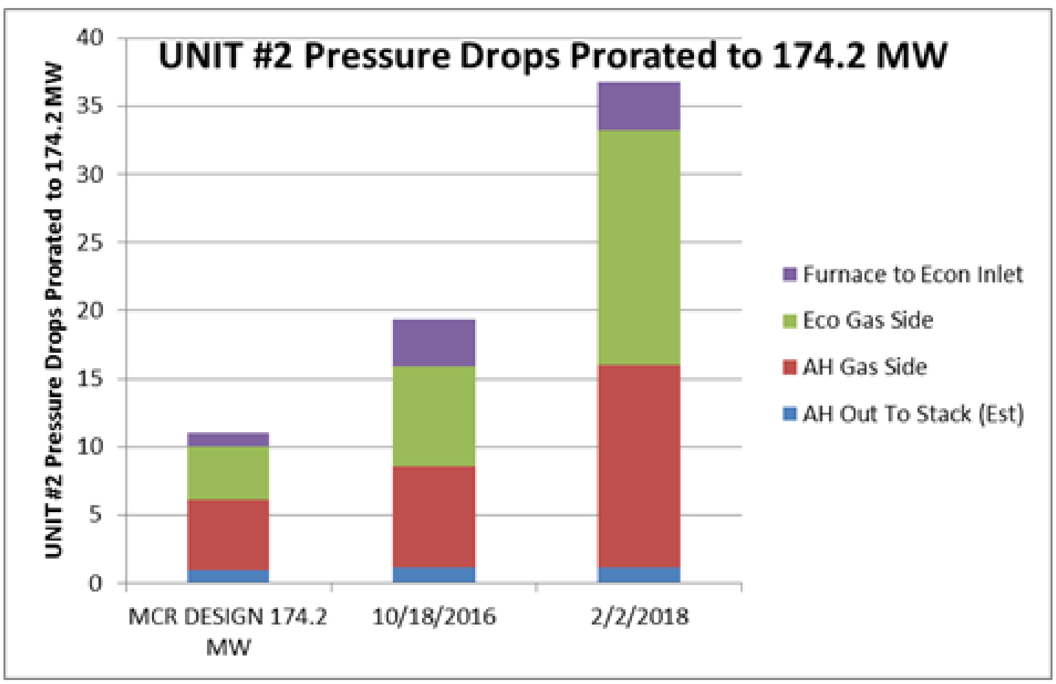


Figure 8 Unit #2 Pressure Drops Prorated to 174.2 MW



Figures 8 and 9 illustrate the current load limitations of Units 1 and 2 and the expected increases in load capability if:

- The air heater pressure drops can be restored to original design
- The economizer pressure drops can be restored to original design
- Both air heater and economizer pressure drops are restored to original design
- Both air heater and economizer pressure drops and boiler efficiency restored to original design. (Reduced stack temperature will be associated with cleaner surfaces)

Figure 9 Unit #1 Furnace Pressure - Restore Surface Cleanliness/Efficiency

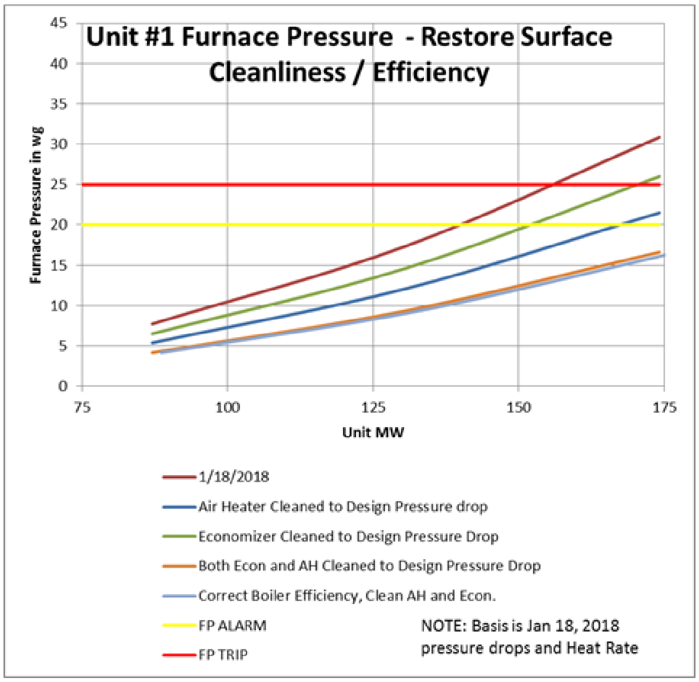
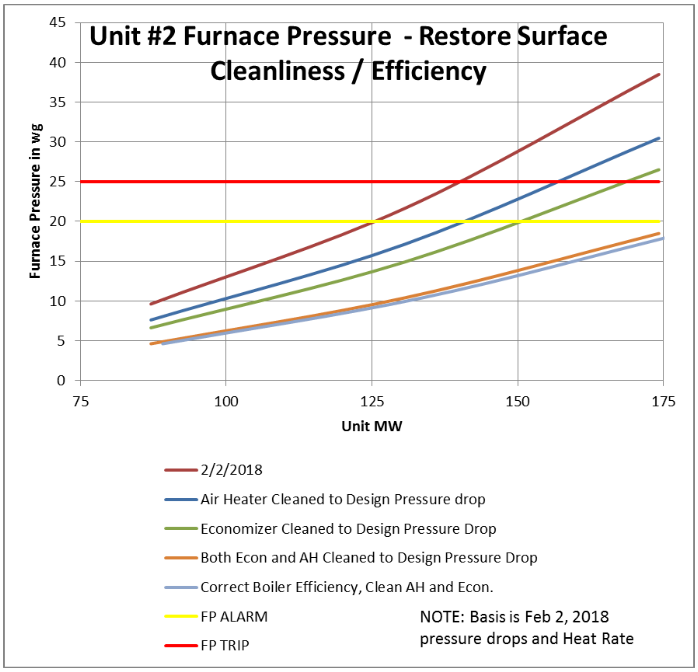


Figure 10 Unit #2 Furnace Pressure - Restore Surface Cleanliness / Efficiency



The potential increases in maximum load as limited by the furnace alarm pressure are shown in the Table 4:

Table 4 Maximum Load As Limited by Furnace Pressure- Restore A/H and/or Econ Pressure Drop

MAXIMUM LOAD AS LIMITED BY FURNACE PRESSURE – RESTORE A/H AND/OF ECON PRESS. DROP					
		Unit #1 (Per Jan 18, 2018 Data @ 133 MW)		Unit #2 (Per Feb 2, 2018 Data @ 125 MW)	
Action	Units	Maximum Load	Load Increase	Max Load	Load Increase
Increase Maximum Furnace Pressure to 19.9” (UNIT #1)	MW	140	+7	-	-
Restore Design Air Heater Pressure Drop	MW	161	+28	141	+16
Restore Design Economizer Pressure Drop	MW	146	+20	151	+26
Restore Both Economizer and Air Heater (Max 175 MW)	MW	175	+53 (Inc. FP Increase)	175	56

Table 4 shows that the largest gain in MW output for Unit #1 is restoring the air heater pressure drop. For unit #2, the biggest gain is in restoring the economizer pressure drop. If both economizer and air heater pressure drops are restored on both units, they will not be load limited below 174.2 MW by furnace pressure. From the charts above, it can be seen that the gains in unit MW output from largest to smallest are:

1) The Unit #1 Air Heater
2) The Unit #2 Economizer
3) The Unit #2 Air Heater
4) The Unit #1 Economizer

If cleaning air heater surfaces is not possible, replacement of heating surfaces which are fouled would restore air heater pressure drop.

Replacement of economizer surface is likely not economically viable if conventional cleaning methods are ineffective. Other methods of cleaning such as the use of explosives or acoustic shock methods (Shock pulse) should be considered.

Improvements in surface cleanliness will increase boiler efficiency, slightly increasing maximum load (if limited by furnace pressure) and reducing fuel oil consumption. There will also be a reduction in FD fan power consumption. These effects were not calculated as part of the current study.

4.3.5 Other Considerations to Restore Unit Load

Improvements in TG heat rate through modifications / repairs to the turbine-generator-condenser would increase unit output when unit input is limited by furnace pressure. The effect of this type of modifications has not been considered in this study.

It may be possible to increase the furnace pressure alarm and trip settings. This would increase the maximum achievable load. The original boiler structural design calculations would need to be reviewed. This review would need to be done by the original boiler designer.

Once cleaned (or heating surfaces replaced), methods of preventing future fouling of air heater and economizer surfaces should be employed. For the air heater, a sufficiently high Average Cold End Temperature (ACET) must be maintained at all loads and during startups. Air heater pressure drop trends from Unit #3 (See Unit #3 section of this report) suggest a minimum ACET of 212 F should be maintained. For the economizer, temperatures are high enough during operation to prevent fouling. Fouling may occur during start ups when feedwater temperatures and/or flows are low.

Sootblowing steam must be dry to prevent the formation of sticky oil-ash deposits. This is particularly important during low loads and startups when combustion efficiency is at its lowest.

Unit #1 could deliver an additional 7 MW of output if furnace pressure is increased to the 19.9” wg level per the Unit #2 Feb, 2018 data. While it is unlikely that the furnace trip point of 25” wg may be increased, it may be possible to increase the alarm point from the current 20” wg dependant on the stability of furnace pressure during high load operation.

4.3.6 Heating Surface Effectiveness (Kf Study)

B&W performance program P140 was used to calculate the convective surface Kf values of the boiler components for the operating periods which were considered. FEGT is also calculated by P140 based on heat balance around the boiler components. The air heater Kf values were determined with reference (Kf = 1.0) to the Alstom predicted performance data (2000). The expected and actual Kf’s are shown in Table 5. The expected Kf for bare tube surfaces is 1.0. The expected Kf for finned tube economizer surface is 1.2.

Table 5 Kf and FEGT Summary, Units #1 and #2

Kf and FEGT Summary, Units #1 and #2					
Unit #	1 & 2	1		2	
Date	Expected	Feb, 2015	Jan, 2018	Oct, 2016	Feb, 2018
Unit Load	174.2	170	133	170	125
Air Heater Kf	1.0	0.66	0.7	0.53	0.67
Economizer Kf	1.2	0.67	0.74	0.65	0.64
Superheater Kf (Avg Prim+Sec)	1.0	0.92	0.88	0.93	0.78
Reheater Kf	1.0	0.88	0.67	0.72	0.72
FEGT(^{o}F) (Expected/Actual)	2589	2577 2590	2438 2439	2577 2669	2408 2396
Burner Tilt (Deg) (Expected/Actual)	+10	+10.5 -1.4	+14.8 4.7	+10.5 -6.3	+15.7 +0.2
Hot Reheat Temp (Deg F)	1000	966	901	947	898

Table 5 illustrates that Kf factors in all cases are less than expected, thus all surfaces downstream of the furnace are underperforming from a heat transfer standpoint. The Unit #1 reheater and the Unit #2 superheater Kf's have dropped significantly between 2015/16 and current operation. As expected from the observed greater than expected draft losses, the economizer and air heater surfaces have the lowest Kf's. On the other hand, there is not a significant difference between the 2015/16 Kf's and current Kf's of the economizer and air heater when draft losses were seen to increase. The cause of this apparent anomaly is not clear. It is possible that some sections of these components are currently cleaner than they were, but blockages in other sections (i.e. center of bank where washing has not penetrated) have increased.

The reheater performance is significantly lower than expected. This has the effect of both increasing stack gas temperature (reducing boiler efficiency) and increasing heat rate.

Burner tilts are not being utilized to maintain design hot reheat temperatures. Positive burner tilts between 10 and 15 degrees would be expected; actual burner tilts are in the +/- 5 degree range. The calculated FEGTs are generally higher than expected, even with the lower than expected burner tilts, suggesting that the furnace surfaces are also underperforming.

In general, the most effective means of reducing stack temperature to improve boiler efficiency is by improving the performance of boiler components in the low gas temperature regions i.e. the air heater and then the economizer. Table 6 illustrates the effect of a 10 F reduction in gas temperature on stack temperature and boiler efficiency.

Table 6 Stack Temperature Change for Change in Upstream Gas Temperature

Stack Gas Temperature Change for Change in Upstream Gas Temperature			
Component / Location	Change in Component Gas Outlet Temperature (F)	Change in Stack Temperature (F)	Change in Boiler Efficiency (%)
Air Heater Gas Outlet	10	10	+0.2
Economizer Gas Outlet	10	4	+0.08
Primary SH Gas Outlet	10	0.8	+0.016

Improvements of boiler heat transfer performance to improve unit efficiency should be prioritized as follows:

1) Air heater (Increased Boiler Efficiency)
2) Economizer (Increased Boiler Efficiency)

3) Reheater (Increased Boiler Efficiency and Reduced T-G Heat Rate)
4) Superheater

Note that improvements in reheater heat transfer performance will have three positive effects on unit efficiency:

- A small improvement in on boiler efficiency due to lower stack temperature
- A reduction in T-G heat rate by means of higher hot reheat temperature
- Lowering of burner tilts leading to lower superheat spray quantity

If burner tilts are modulating to control hot reheat steam temperature, improvements in furnace surface performance will have little to no effect on unit efficiency. Burner tilts respond to the required hot reheat steam temperature and adjust for reduced furnace cleanliness until the maximum negative tilt (Normally -30 Degrees) is approached. Excessively dirty furnace surface can lead to slag falls and this must be monitored visually and controlled accordingly.

5 UNIT #3

5.1 Unit Description and History

Holyrood Unit #3 is a B&W 'El Paso' type boiler. The unit is coupled to a 150 MW Hitachi steam turbine. The boiler delivers a nominal 1000/1000 F steam to the HP/IP turbine. Steam temperature is controlled by biasing the firing rate between the three levels of burners. Air is supplied by two "Sheldons" FD fans. Air flows from the two fans to steam coil air heaters for ACET control and then into two "Howdens" Ljungström type regenerative air heaters. Oil is burned in nine circular oil burners arranged in three levels. Flue gas exits the furnace to the reheater and secondary superheater, then down through the primary superheater and bare tube economizer before passing through the regenerative air heater to the stack.

Reheater surface was removed by Alstom in 2001. The intent of this modification is unknown. The most likely reason to would have been to reduce high load reheater sprayflow.

The FD fan VIV's have been limited to approximately 54% and 70% open on the East and West fans respectively due to vibration of the fan inlet ducting.

B&W are not aware of any other modifications to the unit which would affect the results of this study

5.2 Basis of Study

5.2.1 Fuel

The fuel oil analysis as used in the original Unit #3 design was used (Ref discussion in following sections of this report).

5.2.2 Base Heat Balance Information

Baseline predicted unit performance was taken from the original boiler design B&W boiler Performance Data (PD) sheet dated 9/5/78 and the original heat balance diagram sheet NLH Drawing AO-1403-200-M001 Rev 2. These documents are included in Appendices 8.3 and 8.4 for reference.

5.2.3 Unit Operating Data

B&W requested unit operating data representative of operation for a time period when the unit was capable of full load and another when unit was not capable of full load. NLH subsequently provided plant 'PI' data from Oct 22, 2017 with the unit at 150 MW and January 4, 2018 when the maximum attainable load was 128 MW.

Due to the relatively short time period over which maximum attainable unit load was reduced, B&W requested hourly operating data for the time period between Oct 22, 2017 and January 4, 2018 in order to understand the conditions that were leading to maximum load reductions.

5.2.4 Unit Physical Arrangement

The original B&W boiler arrangement drawings of the boiler physical arrangement were used as basis of the calculated performance. The performance model (P140) was adjusted to reflect the 2001 reheater surface removal.

5.2.5 Heat and Mass Balance Calculations

B&W Single Heat and Material Balance Program – P08475 was used to calculate flue gas flow, flue gas analysis, combustion air flow, furnace heat absorption, **F**urnace **E**xit **G**as **T**emperature (**FEGT**), and boiler efficiency based on the fuel analysis, steam/water, air/gas boundary conditions, and furnace heating surface arrangement.

5.2.6 Boiler Surface Heat Transfer Calculations

Convective surface heat transfer was calculated using B&W program “P140”. The methodology is described in the above discussion for Units #1 and #2.

5.2.7 Furnace Heat Transfer Effectiveness Calculations

As described above, for Units #1 and #2, furnace performance is quantified by the difference between the actual and predicted FEGT. For the B&W unit, the predicted FEGT is calculated by P8475 per B&W standard methods. FEGT higher than predicted indicates underperforming furnace surfaces and/or large amounts of burner fuel input biasing.

5.2.8 Air Heater Heat Transfer Effectiveness Calculations

Air heater heat transfer effectiveness is calculated as per the above discussion for Units #1 and #2.

5.3 DISCUSSION OF RESULTS

5.3.1 Review of Operating Data

A discussion of the limitations of PI operating data vs test data and the effect on calculations is included in the above Units #1 and #2 analysis.

Notable omissions and anomalies in the data received were:

- The FD fan inlet/outlet pressures are not available
    - Assumptions were required to estimate FD fan pressure rise
- The #6 Feedwater heater water inlet temperature reading is not valid.
    - Assumptions were required to calculate reheater steam flow
- The PI reported superheater spraywater flow was implausible
    - Assumptions were required in Stage 2 (Kf) analysis

5.3.2 Turbine Generator Heat Rate

The original design and the current turbine heat rates for the Oct 22 and Jan 4 data are shown in Table 7.

Table 7 Turbine - Generator Heat Rate Design vs Current Unit #3

TURBINE – GENERATOR HEAT RATE DESIGN vs CURRENT				
		Design (Ref AG 1403-200-M001 Rev2	Oct 22, 2017	Jan 4, 2018
Gross Output	MW	150	149.2	128.2
Turbine Heat Rate Expected	Btu/kwhr	7621	7623	7665
Turbine Heat Rate (Adjusted for off design boiler boundary Conditions i.e. hot RH Temp)	Btu/kwhr	-	7597	7720
Turbine Heat Rate Actual	Btu/kwhr	-	8188	8260
Required Boiler Output To Turbine	Btu/hr/10^6	1143	1222	1059
Increase in Turbine Heat Rate	%	-	7.4	7.8

The current heat rates are significantly higher than design, increasing the required boiler output per MW generated.

Note that the boiler heat output also includes other loads such as Aux steam to other units/building heat, etc. and output to blowdown. These outputs were not included in the turbine heat rate calculations. The boiler output calculations assumed that:

- No aux steam flowed into or out of the Unit 3 boiler envelope
- The aux steam extracted from the boiler was used within the boiler envelope (Predominantly steam coil air heaters, fuel atomization and fuel oil heating)
- No sootblowing steam consumption
- Boiler blowdown flow 1% of main steam flow

Steam flows for calculation of turbine heat input were determined as follows:

- HP Steam flow to turbine = (Feed Water Flow) – (Blowdown Flow) – (Aux Steamflow)

- Reheater Flow = (HP Steam flow) – (Design HP Turbine Leakages) - (#6 Heater Steam Extraction Steam Flow calc. by heat balance)

HP steam flow calculated from feed water flow is generally more accurate than the commonly used steam flow inferred from HP turbine inlet pressure, particularly for older turbines.

5.3.3 Deviations from Design Turbine – Boiler Boundary Conditions

The operating turbine heat rates illustrated above are affected by deviations in boiler operating conditions from design. These conditions are:

- Main steam temperature / pressure
- Hot reheat temperature
- Superheater and reheater sprayflows
- Boiler blowdown and aux steam flows
- Reheater pressure drop

The magnitude of these corrections is relatively small. The corrections are indicated in Table 8 were made using heat rate correction curves for a Hitachi turbine of similar vintage, size, and design conditions.

Table 8 Heat Rate Corrections Unit #3

HEAT RATE CORRECTIONS DEVIATIONS FROM DESIGN TURBINE – BOILER BOUNDARY CONDITIONS EFFECT						
			Oct 22, 2017		Jan 4, 2018	
Unit Output	MW	150	149.2		128.2	
		Design	Measured	Heat Rate Correction	Measured	Heat Rate Correction
Main Steam Temperature	F	1000	1000	1.0000	1000.4	0.9999
Main Steam Pressure	Psig	1800	1799	1.0000	1798	1.0000
Hot Reheat Temperature	F	1000	1005.5	0.9992	941	1.0089
Superheat Spray flow	Lb/hr	0	48000*	1.0022	48000*	1.0026
Reheat Spray flow	Lb/hr	0	2140	1.0011	2196	1.0013
Boiler Blowdown & Aux Steam Flows	Lb/hr	0	16500	0.9942	12700	0.9945
Overall Turbine Heat Rate Correction Due to Deviations in Boiler Boundary Conditions (Positive=Increased HR)	- (%)	1.0000 (0)	-	0.99660 -0.3%	-	1.0071 0.7%

*Estimated (Plant superheater spray flow measurement is implausible)

A further correction for reheater pressure drop is available but was not applied since total reheat pressure drop (including piping) was not measured. This correction is normally very small. The correction for Condenser vacuum was also not applied. This correction can be substantial, but was not considered as it is outside of the scope of this study.

The small boundary condition corrections here indicate that the majority of the increased T-G heat rate is due to T-G inefficiencies. In general aging steam turbines experience heat rate increases due to high condenser pressure, higher than design turbine valve and gland leakages, depositions on and wear of turbine blades. B&W has seen heat rate increases similar to those indicated in the above table on T-G units of similar vintage and size.

5.3.4 Fuel Oil Flow

Although inaccuracies exist in measurements of the fuel oil flow to the unit and there are variations in fuel heating value, fuel oil flow relative to unit MW load is an indicator of unit heat rate. Table 9 shows the expected, calculated, and measured fuel oil flows for the Oct 22 and Jan 4 data. Calculated oil flows are based on 18450 Btu/Lb. The calculated oil flows are within 3% of the measured oil flows.

Table 9 Fuel Oil Flow Calculated/Expected Unit #3

Unit Output	MW	149.2 (Oct 22, 2017)	128.2 (Jan4, 2018)
Expected Oil Flow (Design HHV, HR and Blr Efficiency)	Lbs/hr	69,579,	60,114
Calculated Oil Flow	Lbs/hr	77,367	67,138
Calculated/Expected Oil Flow	-	1.11	1.12
Plant Measured Oil Flow	Lbs/hr	79,276	67,199
Oil HHV	Btu/lb	18,278-18,472 (2017 Deliveries)	

5.3.5 Boiler Efficiency and Air Heater Leakage

Boiler Efficiency is predominantly driven by excess air and the difference between inlet air temperature and outlet flue gas temperature (Corrected for no a/h leakage i.e. undiluted). Other factors such as atomizing steam flow, radiation loss, and unburned carbon loss are small for oil fired units. The key parameters are illustrated in Table 10 with reference to the original design conditions.

Table 10 Boiler Efficiency and Air Heater Leakage (Unit #3)

BOILER EFFICIENCY AND AIR HEATER LEAKAGE				
		Design (B&W PD Sheet C/7391, MCR Load)	Oct 22, 2017	Jan 4, 2018
Excess air To Burners	%	3	5	7
Air Inlet Temperature	F	80	45	61
Gas Temperature Entering A/H	F	662	747	737
Stack Gas Temperature (Diluted)	F	280	318	324
Air Heater Leakage (% of Inlet Gas Flow)	%	9.5	22.2	27.5
Stack Gas Temperature (Corrected for No Leakage)	F	297	364	376
Boiler Efficiency	%	88.59	86.45	86.46
Air Heater Leakage Flow	Lb/hr	103,000	267,000	305,000

The boiler efficiency is approximately 2% lower than design, mainly due to the higher than design corrected air heater outlet temperature and the lower than design air inlet temperature. The reduction in efficiency combined with the higher than design excess air and much higher than design air heater leakage increases the required FD fan air flows significantly. These increases compound with the additional air flow required by the increased T-G heat rate discussed above.

5.3.6 FD Fan Duty Requirements – Design vs Current

5.3.6.1 Required Air Flows

The required boiler airflows to achieve a 150 MW unit output at current operating conditions are summarized in Table 11. The required air flow leaving the air heater is calculated based on TG heat rate, boiler efficiency, and excess air from the Oct 22 (149.2 MW) site data. The required air flow entering the air heater was calculated based on both the Oct 22 and Jan 4 data to illustrate the effect of the increased air heater leakage resulting from the higher Jan 4 air heater air/gas side differential pressure.

Table 11 FD Fan Airflow Requirements - Design vs Current (Unit #3)

FD FAN AIRFLOW REQUIREMENTS – DESIGN vs CURRENT OPERATION (150 MW)				
		Design (MCR, 150 MW)	Oct 22, 2017	Jan 4, 2018 (Additional AH Leakage)
Original Design Airflow Leaving AH	Lb/hr	1,029,700	-	-
Additional AirFlow due to TG Heat Rate Increases	%	-	7.4	
Additional AirFlow due to Boiler Efficiency Loss	%	-	2.5	
Additional AirFlow due to higher Excess Air	%	-	2.0	
Total Additional AirFlow to Burners (Entering AHs)	%	-	12.3	
Required Air Flow Leaving Air Heaters	Lb/hr	1,029,700	1,156,000	
Additional Flow Air heater leakage (% Air Leaving)	%	10	23.1	26.4
Required AirFLow Entering Air Heaters	Lb/hr	1,132,700	1,423,000	1,461,000
Required Airflow Entering Air Heaters / Fan	Lb/hr/fan		715,500	753,000
% Increase in FD Fan Outlet Airflow vs 150 MW Design	%		25.6	29.0

5.3.6.2 Required FD Fan Pressure Rise

The boiler air and flue gas side pressure drops during Oct 22 and Jan 4 operation vs the design pressure drops are summarized in Table 12. Pressure drops are higher than design due to the combination of increased required air and flue gas flow along with increased resistance. Due to the assumptions made regarding air heater and steam coil air heater air-side pressure drop, the pressure drop summary should be considered ‘approximate only’ until actual FD fan pressure rises can be confirmed.

The pressure drop in the FD fan inlet ducts is not measured thus it has been assumed unchanged from original design. The FD fan outlet pressures are also not available. This was estimated by adding the Ljungström air heater air-side pressure drop (proration of the design pressure drop by the ratio of measured/design gas side pressure drop), and an estimated steam coil air heater pressure drop (estimated at two times a ‘typical’ steam coil since the steam coils are reportedly fouled/damaged).

Table 12 Fd Fan Pressure Rise - Design Vs Operating Unit #3

AIR AND GAS SID PRESSURE DROPS – DESIGN VS OPERATING				
		Design 150 MW	Oct 22, 2017 (149.2 MW)	Jan 4, 2018 (Prorate to 150 MW)
Airflow To burners	Lb/hr	1,029,700	1,156,000	1,156,000
Air Heater Leakage	Lb/hr	103,000	267,000	305,000
Airflow Leaving FD Fans (Inc AH Leakage)	Lb/hr	1,132,700	1,423,000	1,461,000
Draft loss Burners	in Wg	4.9	7.2	9.9
Draft loss Furn and CP	in Wg	6.3	9.2	6.6
Draft Loss AH Gas Side	in Wg	3.1	8.6	11.4
Draft Loss AH Air Side (Prorate from Gas Side)	in Wg	2.4	6.6	8.8
Draft Loss SCAH (Est)	in Wg	1.7	6.4	6.8
Ducts Draft loss (Prorate from Design)	in Wg	5.4	6.8	6.8
Flues Draft Loss (Prorate from Design)	in Wg	2.1	2.6	2.6
Total Draft Losses	in Wg	25.7	47.3	52.8

The largest contributor to additional fan pressure rise duty is the additional flow associated with the combination of increased turbine heat rate, higher than design excess air, and reduced boiler efficiency. The additional pressure drop in the regenerative air heaters and the steam coil air heater are the next most significant contributor to additional fan loading.

5.3.7 FD Fan Capacity Discussion

The combination of turbine heat rate increases, boiler efficiency reduction, air heater leakage, and higher than design combustion system excess air increase the required airflows as discussed above. The increased flows inherently increase the system pressure drop by approximately 26% relative to the original design. Pressure drop increases of a similar magnitude are observed due to changes in flow path resistance, such as dampers throttled, burner air register settings, boiler convection pass and air heater fouling. The expected performance for each fan as operating on Jan 4, 2018 is illustrated in Figures 10 and 11. The curves are based on the original Sheldons Eng. fan curve (Ref Appendix 8.5), corrected for inlet air density and fan RPM.

Figure 11 West FD Fan Jan 4/18 Unit #3

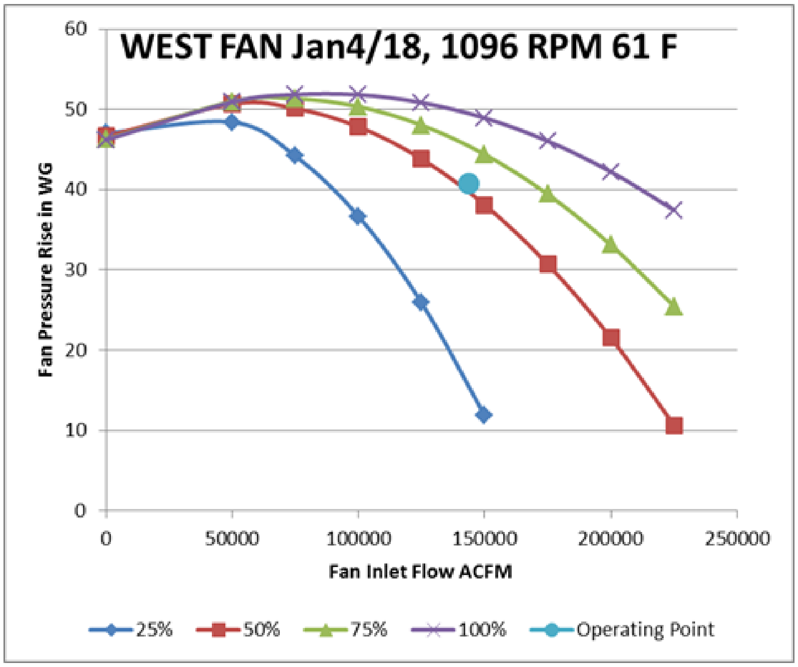
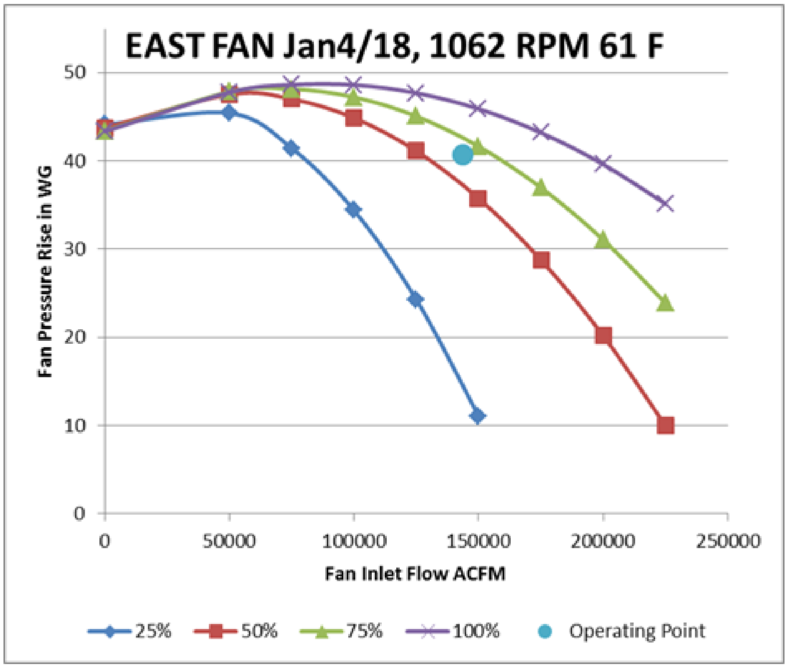
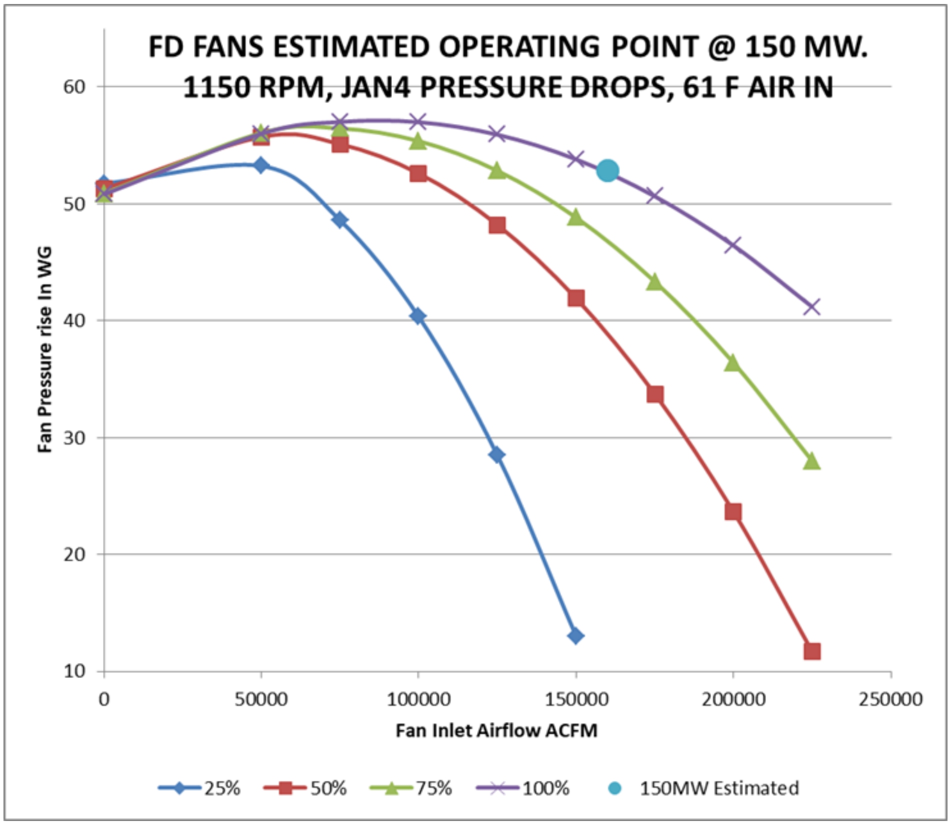


Figure 12 East Fan Jan 4/18 Unit #3



Under the Jan. 4, 2018 operating conditions, the FD fans would have been capable of delivering the required air flow to the unit if operated at the rated 1150 RPM and 100 % VIV opening. On that day, the fan speed was limited to approximately 1080 RPM and the VIV's were in the 54%/70% east/west position with the unit at 128 MW. The required fan duty to make 150 MW per the Jan 4 data is illustrated in Figure 12. This curve is based on the original Sheldons fan curve. A correction was required for lower than design inlet air temperature (Sheldons Fan Curve temperature basis was 105 F).

Figure 13 FD Fans Estimated Operating Point, 150 MW, Jan 4/18 Unit #3



5.3.8 Air Heater 'ARVOS' basket replacement

A proposal from ARVOS for replacement air heater hot end heating elements was reviewed from the standpoint of the restoration of maximum boiler load capability and FD fan capacity. The expected performance as received from ARVOS for the new elements if installed with the existing cold end elements (assumed to be in 'as new' condition from a heat transfer / pressure drop standpoint) is included in Appendix 8.6. Table 13 outlines the required fan performance with the new heating elements installed. The existing FD fans will easily deliver sufficient airflow for 150 MW operation at approximately 960 RPM and 60-65 % average VIV opening.

Table 13 Arvos Replacement Hot End Heating Elements Performance

FAN PERFORMANCE SUMMARY - NEW AIR HEATER HOT END BASKETS / SEALS					
				New Hot End	New Hot End
		10/22/2017	1/4/2018	Baskets and Seals	Baskets, Old Seals
		150 MW	128 MW	150 MW	150 MW
Flows Mlb/hr (Oct Data Unit @ 150 Mw)					
	Air Entering AH	1,390,100	1,312,800	1,279,870	1,354,177
	Leakage Air	267,000	305,000	156,770	231,077
	Air Leaving Air Heater	1,123,100	1,007,800	1,123,100	1,123,100
Temperatures F					
	Air Entering FD Fan	45	61	45	45
	Air Entering Air Heater	99	128	128	128
Pressures In WG					
	FD Fan Pressure Rise	45.2	40.7	34.4	34.9
	AH Outlet Plenum	27.4	26.9	20.9	20.9
	Air Heater Air Side Pressure Drop	6.6	7.1	1.9	1.9
	Air Heater Hot End Differential	22.4	18.0	22.4	22.4
	Air Heater Gas Side Differential	8.6	9.2	3.2	3.2
Fan Performance					
	FD Fan Volume Flow ACFM/Fan	147,816	143,842	135,991	143,886
	FD Fan RPM	1018	1062	960	960
	FD Fan VIV %	54/70	54/70	60.0	65.0
	Horsepower/ Fan (Predicted)	867	940	727	772

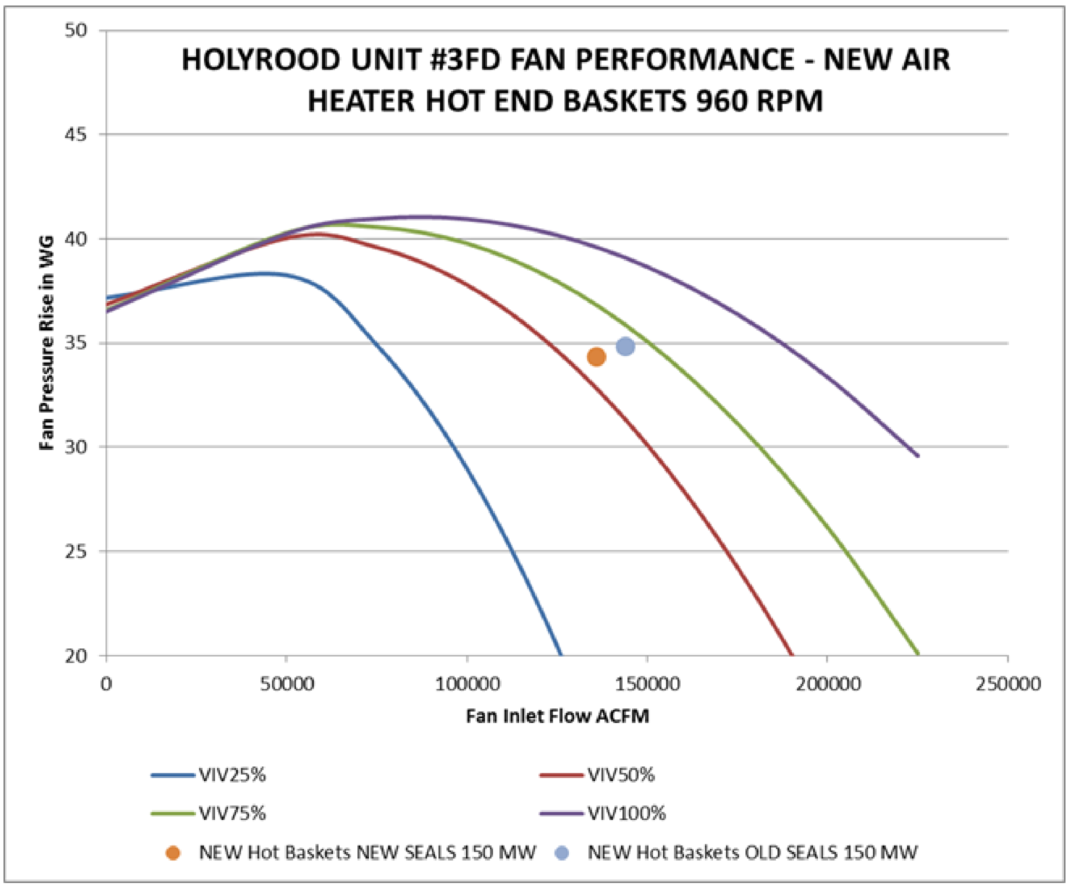
Note the following:

- Combustion air flows requirements are based on boiler operating data Oct 22, 2017 @ 150 MW
    - No adjustments were made for improved efficiency (Which should be achieved with AH basket replacement). This will result in a conservative capacity estimate
- Air heater leakage calculated two ways to assist in evaluation value in new seals
    - Without new seals, air heater leakage adjusted from Oct 22, 2017 calculated leakage for reduced differential pressures with the new baskets
    - With new seals, air leakage adjusted from ARVOS predicted data based on higher hot end air heater differential pressure
- FD Fan performance is calculated based on ‘typical’ current operating VIV openings, Fan RPM selected to match required pressure rise

- Air Heater Outlet plenum pressure setpoint reduced due to reduced air heater pressure drop
- There is a savings in fan power as shown (estimated) in Table 13.

Figure 13 illustrates the estimated fan operating points @ 960 RPM with the proposed air heater upgrades.

Figure 14 FD Fan Performance - New ARVOS Air Heater Hot End Elements Unit #3



5.4 Heating Surface Effectiveness (Kf Study)

Heating surface effectiveness factors (Kf's) were calculated by B&W program P140. Table 14 summarizes the results.

Table 14 Kf and FEGT Summary Unit #3

Kf and FEGT Summary, Unit #3			
Date	Expected	Oct 22, 2017	Jan 4, 2018
Unit Load	150	150	128
Air Heater Kf	1.0	0.91	0.88
Economizer Kf	1.0	0.91	0.98
Superheater Kf (Avg Prim+Sec)*	1.0	0.90	0.75
Reheater Kf	1.0	0.96	0.71
FEGT(^{o}F) (Expected/Actual)	2482	2482 2528	2394 2476
Main Steam Temp (Deg F)	1000	1000	1000
Hot Reheat Temp (Deg F)	1000	1006	941

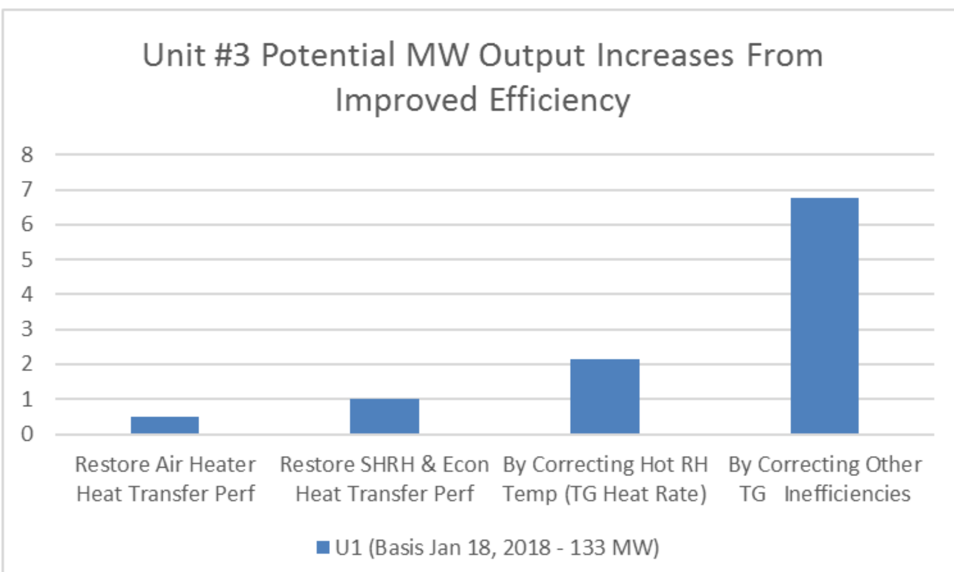
*Superheater Kf Estimated (Spraywater Flow Not Available)

The Kf analysis shows that all surfaces are underperforming from a heat transfer effectiveness standpoint. The effectiveness of the superheater and reheater surfaces dropped significantly during the Oct 2017 – Jan 2018 time period. The air heater and economizer Kf's, while below expected, did not change significantly during that time period; this is somewhat unexpected for the air heater given the large increase in pressure drop seen during this time. One possible explanation may be that localized depositions are blocking flow in a relatively small portion of the depth of the heating surfaces. Flow patterns may also have changed if the two air heaters are not fouling at the same rate, leading to an air and flow 'shift' between them. This could affect the indication of stack temperature from the plant instrumentation.

As discussed above, the major deficiencies in the Unit #2 performance as they affect efficiency as based on the January 2018 data are the higher than expected Turbine-Generator heat rate and reheat cleanliness / hot reheat temperature. The low heat transfer effectiveness of the superheater and reheater surfaces is not a major factor in terms of boiler efficiency due to the relatively good thermal performance of the air heaters and economizers. The significant reduction in superheater and reheater Kf values should be investigated i.e. the surfaces should be inspected for cleanliness. Increases in sootblowing frequency and/or blowing pressures may be necessary to maintain cleanliness of these surfaces.

Figure 14 illustrates the additional unit output that that would be expected if the boiler and T-G inefficiencies are corrected.

Figure 15 Unit #3 Potential MW Output Increase from Improved Efficiency



6 FUEL OIL RELATED ISSUES (COMMON UNITS #1,2,3)

Fuel oil is supplied to the three units from common storage tanks. Oil is pumped and heated to the required pressure and temperature for burner atomization by independent pumping / heating sets for each unit.

The fuel oil analysis data in the NLH supplied spreadsheet database was reviewed. From a combustion and heating value standpoint, the fuel analysis in recent years is very close to the Unit #3 original design fuel. Combustion calculations were therefore based on the Unit #3 design fuel. The Sulphur content has been consistently below 1% since early 2009 per Figure 15. The Vanadium (V2O) content dropped significantly in late 2005 and is currently consistently less than 50 ppm per Figure 16. Overall the fuels currently burned are better than ‘typical’ Bunker fuels with lower than normal levels of both Sulphur and Vanadium.

Figure 16 Fuel Oil Sulphur % by Wt. (1995-2017)

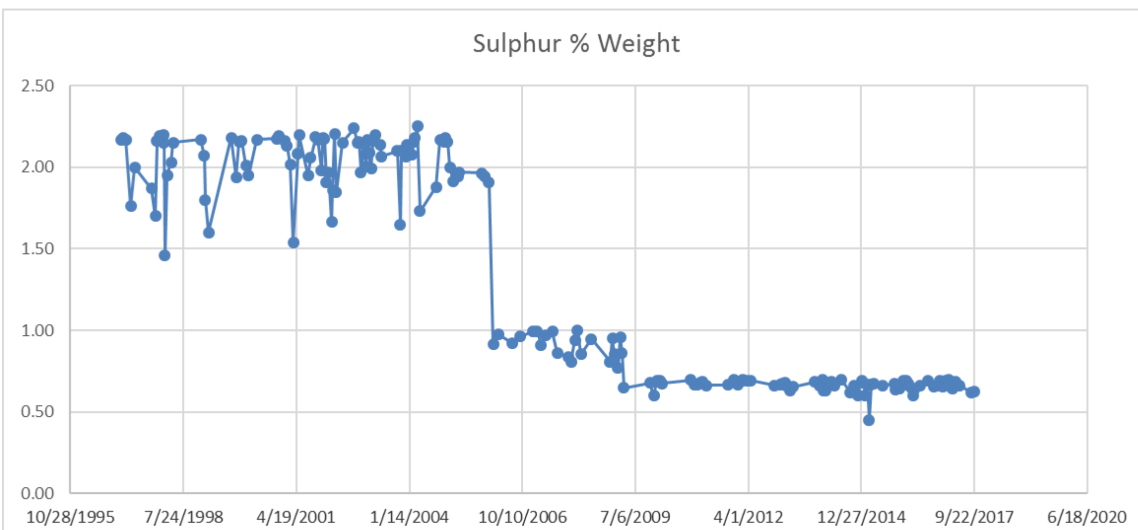
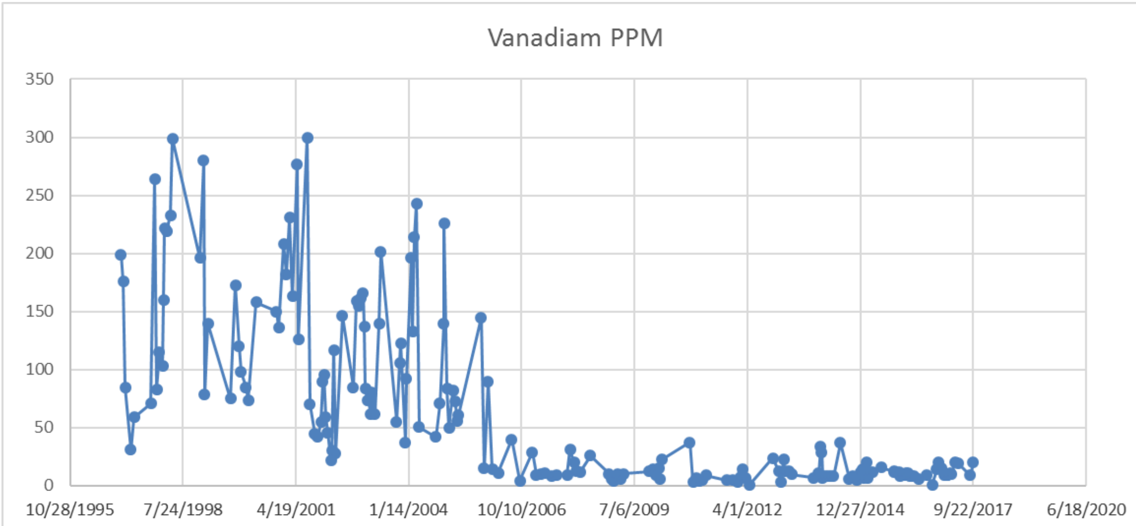


Figure 17 Fuel Oil Vanadium PPM (1995-2017)



Interactions of Vanadium, SO_2 / SO_3, and unburned carbon in the products of combustion lead to air heater fouling. These deposits can block the flue gas passages on air heater heating surfaces, increasing pressure drop and reducing heat transfer effectiveness. Finned tube economizers may also be affected during start-up and very low load operation. Unburned carbon is the largest component of these deposits and it is typically highest during start-up and low load operation.

Low air heater metal temperature as indicated by the Average Cold End Temperatures (ACET) increase the condensation rate of SO3 on the baskets and increase the tendency for deposits to form. Air heater metal temperatures are also lowest at low loads if sufficient inlet air preheating is not supplied. It is thus imperative that air heater ACET is maintained at all loads and operating conditions.

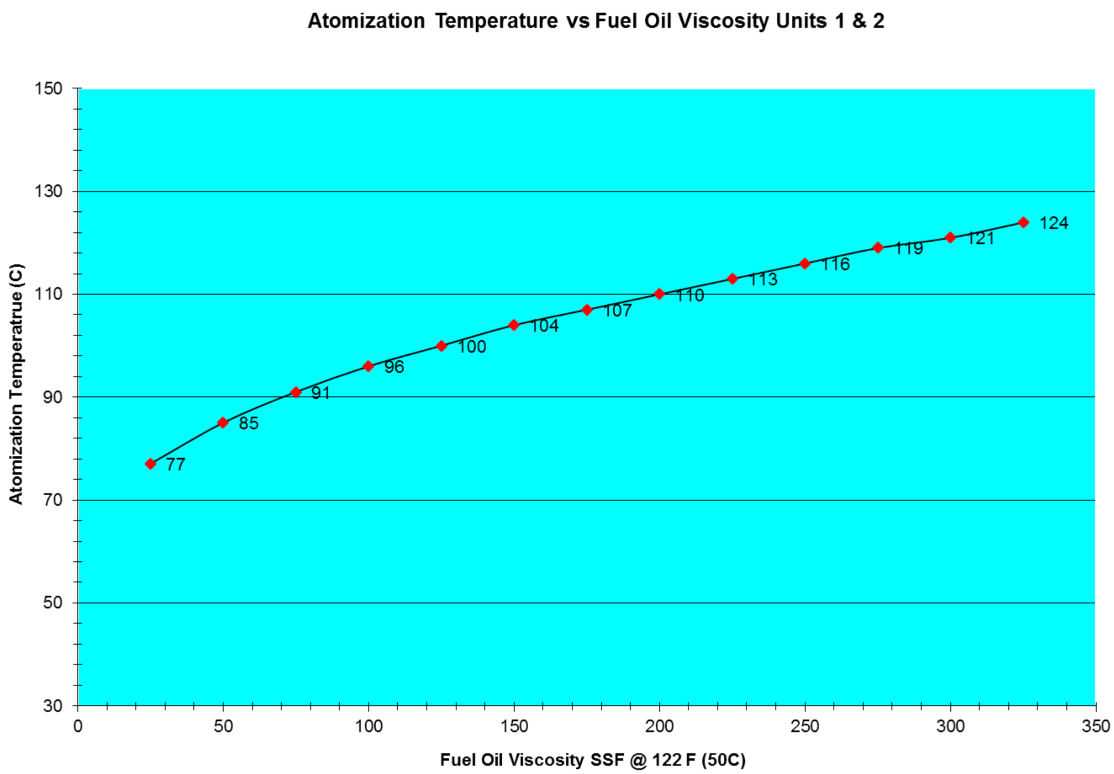
The regenerative air heaters of all three units and the finned tube economizers of units #1 and #2 are experiencing significantly higher than design pressure drops.

6.1 Atomizing Temperature

The viscosity of oils currently utilized at Holyrood range between 50 and 189 SFS (@ 122 F). Sufficient fuel oil heating must be supplied to ensure proper atomization and complete combustion.

The required atomizing temperature for Units #1 and #2 atomizers as a function of SFS viscosity is shown in Figure 17 (Ref. Alstom info supplied to B&W by NLH). According to site reports, atomizing temperatures are currently approximately 187 F (86 C)

Figure 18 Atomization Temperature vs Fuel Oil Viscosity Units 1 & 2



The Unit #3 B&W atomizers are designed for 135 SSU viscosity at the burners. Figures 18 and 19 illustrates the required atomizing temperature as a function of the fuel oil SFS @122 F to achieve the required atomizing viscosity.

Figure 19 Recommended Oil Atomizing Temperature, Holyrood Unit #3 (Celsius)

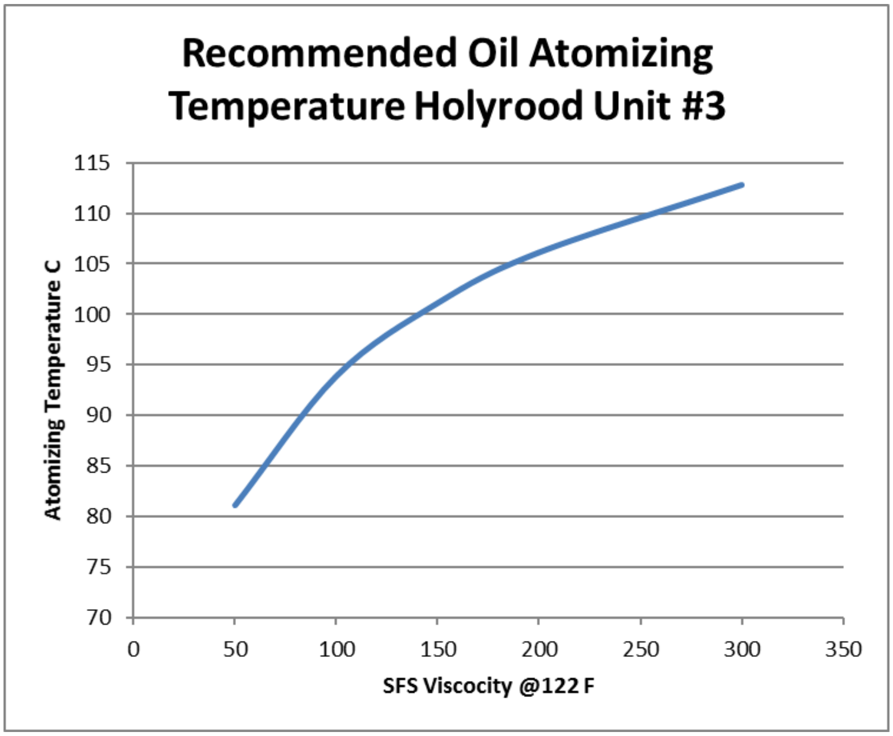
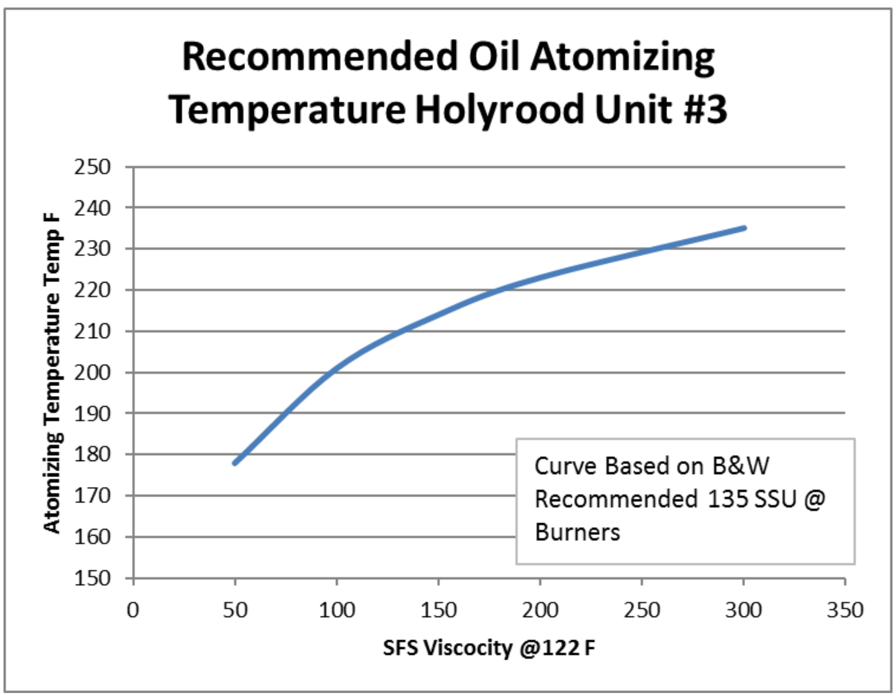


Figure 20 Recommended Oil Atomizing Temperature, Holyrood Unit #3 (Fahrenheit)



To accommodate fuel oil viscosities up to 200 SFS (@122 F):

For Units 1 & 2, an atomizing temperature sufficient to achieve 100 SSU Is recommended or in the absence of viscosity data 110 C (230 F)

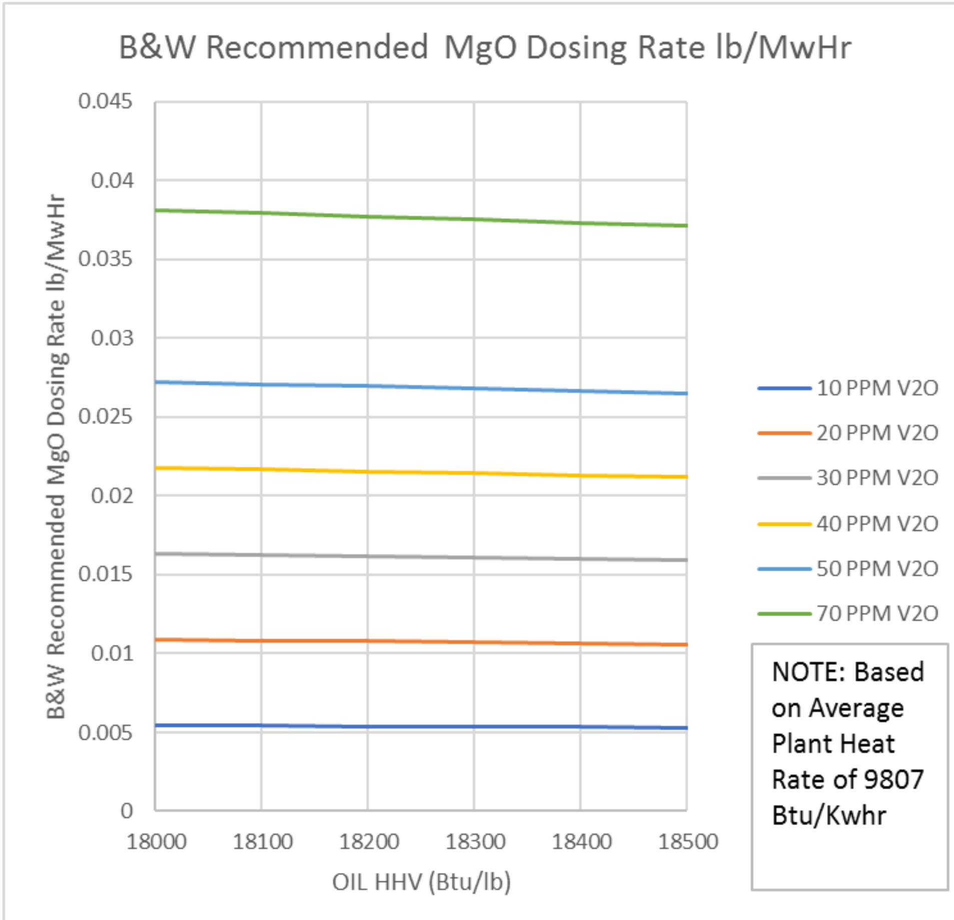
For Unit #3, an atomizing temperature sufficient to achieve 135 SSU Is recommended or in the absence of viscosity data 225 F.

Low atomizing temperature leads to incomplete combustion and increased unburned carbon in fly ash. This ash combined with SO_3 condensate in low temperature regions of the boiler lead to corrosion and fouling.

6.2 Fuel Oil Additives

Fuel oil additives reduce the potential for high temperature corrosion and low temperature fouling due to the fuel oil Vanadium. These issues are linked to the catalysing effect of Vanadium on high temperature tube metal corrosion and on the conversion of SO_2 to SO_3. MgO added to the fuel stream is effective in reducing these effects. B&W recommends a minimum dosing rate of 1 lb MgO per lb V_2O in the fuel oil to reduce the potential for both corrosion and fouling. Figure 20 illustrates this recommended dosing rate per unit MWhr output based on an average unit heat rate of 9807 Btu/Kwhr. If a higher dosage rate is recommended by the supplier of the additive due to the specific composition of his additive package, the higher recommended dosage rate should be implemented.

Figure 21 B&W Recommended MgO Dosing Rate lb/Mwhr



NLH discontinued the use of the plant fuel oil additive system in 2014. The decision to take the system out of service may have been based on the improved fuel quality in 2006 and 2009. Load limitations started to occur in 2015 and 2016 on Unit #1 and #2 respectively and 2017 on Unit #3. No significant changes are seen in the fuel analysis between 2009 and 2015. With no other apparent changes in operating conditions, the MgO system was most likely reducing the tendency towards fouling of the air heater surfaces. It is recommended that the MgO dosing system is returned to service.

Vendors of oil additive packages often supply and recommend fuel oil additives which are designed to improve combustion. B&W has not seen any benefit to using these ‘combustion improvers’ in utility boilers as it relates to fouling or ash ‘stickiness’.

6.3 Air Heater Differential Trend – Oct 22, 2017 to Jan 4, 2018 (Unit #3)

Unit #3 experienced a relatively rapid increase in air heater pressure drop associated with a reduction in load capability between Oct 22, 2017 and Jan 4, 2018. A trend of air heater differential vs. time based on Unit #3 PI operating data on an hourly basis was developed to identify if low load operation and/or low ACET was leading to increased fouling. An ‘index’ of air heater cleanliness was calculated i.e. (Air Heater Differential)/(Total Air Flow). If no further pluggage is occurring this index would be a constant over time. The index is plotted below In Figure 21. A plot of the unit MW output follows in Figure 22, and Figure 23 illustrates the air heater Average Cold End Temperatures (ACET) trend. Although these trends are based on Unit #3 data, they are also relevant to the similar air heaters of Units #1 and #2.

Figure 22 Air Heater Differential Index - Unit 3

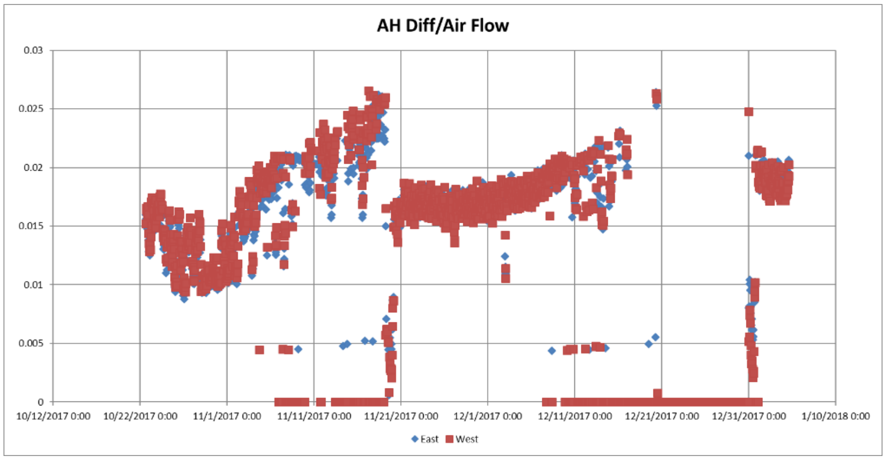


Figure 23 Unit 3 MW Output Oct 2017- Jan 2018

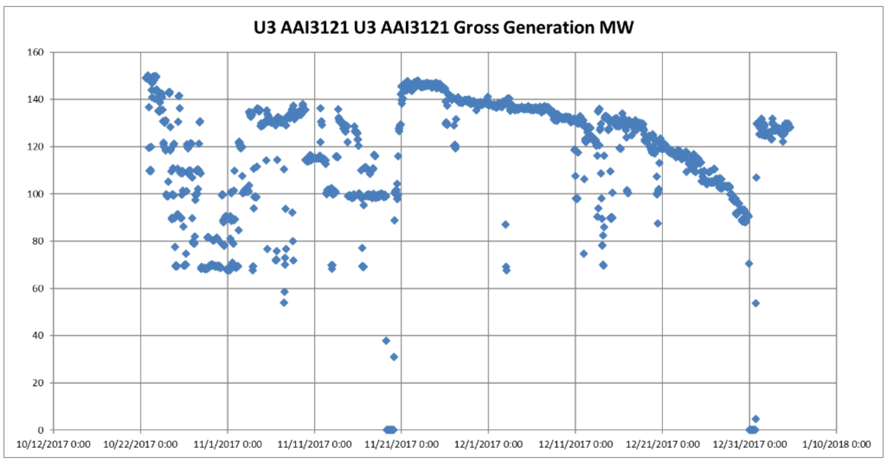
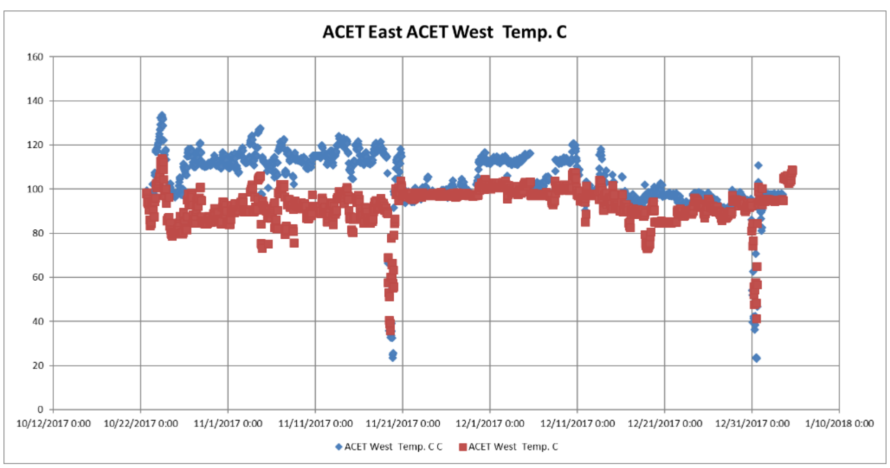


Figure 24 Air Heater ACET Oct 2017- Jan 2018 Unit 3



Gaps in the chart data correspond to times when unit was off line or the differential pressure measurement was not available. The most rapid rise in differential index was during the November operating time frame. During this time period the unit operated for significant periods at loads less than 100 MW. The ACET for the west air heater was significantly lower than the east side, often dropping to as low as 80 C (176 F). The B&W recommended minimum ACET for regenerative air heaters on oil fired units is 190 F (88C). Although the operating ACET was not significantly lower than recommended there is a correlation between low ACET and increased draft loss.

Air heater differential pressure measurements were not available from mid December until a short shutdown on December 31 as the economizer gas outlet gas pressure transmitter appeared to be malfunctioning (Pegged?). During this time period, load dropped rapidly and the ACET's were at even lower levels. This further suggests that low ACET is leading to high rates of air heater pressure drop increase. Note that the air heater differential did not increase during the period from Nov 21 to mid December when the ACET was maintained above 100 C and unit load was above 140 MW. Based on this, a minimum 100 C (212 F) ACET target is recommended.

Figure 21 shows a significant drop in the differential index on or about November 20 and another on December 31, suggesting that the air heaters were washed at that time.

6.4 Heating Surface Removal

Removal of boiler heating surfaces (economizer or heater surfaces) which are leading to increased pressure drop would reduce furnace pressures and reduce FD fan loading. Surface removal can have multiple negative effects on boiler performance and mechanical integrity as follows:

6.4.1 Air Heater Heating Surface Removal

If removing just the 'hot end' elements, the air heater vender predicted performance with only cold end baskets installed would be required to evaluate the effect on boiler performance and efficiency. The air heater vendor would need to advise the effect if the air heaters structural integrity is suitable for the higher outlet gas temperatures under these conditions and any effect on air heater leakage rates.

Other problems that may occur if removing only the hot end baskets are as follows:

- Reduced combustion air temperature leading to unacceptable combustion i.e. high CO, high unburned carbon loss, and a visible plume. (Likely at part loads, possible for high loads)

- High flue gas outlet temperatures leading to possible structural damage to the air heater, downstream expansion joints, flues, and stack. (Likely at high loads, possibly at low loads)

- A significant drop in boiler efficiency (Certainly - all loads)

- Reheat spray flow required at high loads (Likely at high loads)

- Overheating of superheater and reheater tube metals, particularly primary outlets due to increased superheat sprayflow and high fluegas/steam flow ratio (Possibly - all loads)

The removal of hot end air heater baskets for continued operation is therefore not recommended.

Complete removal of air heater surfaces would certainly lead to very poor combustion and very likely structural damage of the flues / expansion joints / stack and thus would not be recommended.

6.4.2 Economizer Heating Surface Removal

Limited removal of economizer surfaces which are blocked by fouling may be a viable option to reduce pressure drop if cleaning these surfaces is not possible. Any removal of economizer heating surfaces must consider the following:

- Increases in flue gas temperature to the air heaters which could lead to structural damage to air heaters and air heater inlet gas flues/expansion joints.

- Increases in air heater outlet gas temperature possibly leading to similar structural problems discussed for air heater surface removal.

- Exceedance of maximum stack temperature limitations structurally or environmentally

- Combustion air temperature increases, possibly beyond the temperature limitations of structural design and expansion joints in the ducts and burners.

- Higher levels of s/h spray and possible overheating of superheater tube metallurgy

- Possible negative effects on boiler natural circulation issues due to low feedwater temperature to drum (Would require review by boiler designer)

A thorough ‘survey’ of where the current areas of blockage are located in both banks would be required to estimate performance and performance predictions of the remaining surface would be ‘estimates’ at best. The path forward would be dependent on the results and accuracy of the survey.

If the blockages are primarily in the bottom bank simplest would be removal of entire bank (After investigating the constraints listed above). If the blockage is in the top bank, and that bank is removed the temperature limitations of the bottom bank supports would also need to be understood.

Considering the above issues, partial removal of economizer surfaces should be considered as a last resort solution. It would also require a considerable inspection, engineering (including pressure part modifications), and construction effort.

Complete removal of economizer surfaces would certainly lead to boiler structural and operational problems and is thus not recommended.

7 WARRANTY / LIMITATION OF LIABILITY

B&W warrants that advice and consultation services and engineering studies will be performed in a manner consistent with generally accepted industry standards and practices. The sole remedy is that any portion of the services furnished to Purchaser which is shown not to have been so performed shall be corrected or re-performed to the standards in effect at the time of original performance at B&W expense; provided all necessary information and access requested by B&W is given to substantiate such claim, and further provided that such non-conformance is detected by Purchaser within ninety (90) days following completion of that portion of the services, and B&W is immediately notified in writing.

The foregoing shall not apply to services performed under the direct supervision of Purchaser. B&W shall not be responsible for suitability or performance of work done by others or for loss or expense arising from same, unless it is specifically ordered by B&W.

There is no warranty or representation, express or implied, with respect to the accuracy, completeness or usefulness of the information contained in any report, or that the use of any report contents may not infringe privately-owned rights. Moreover, B&W will assume no liability for any direct or indirect damages, however caused, including (without limitation) by professional negligence or fundamental breach of contract, resulting from reliance upon or application of the contents of the report by any person.

IN CONSIDERATION OF THE ABOVE EXPRESS WARRANTY EXTENDED BY B&W, ALL OTHER WARRANTIES *OR CONDITIONS, EITHER EXPRESS OR IMPLIED WHETHER ARISING AT LAW, IN EQUITY, BY STATUTE, CUSTOM OF TRADE, OR OTHERWISE, INCLUDING MERCHANTABILITY AND FITNESS FOR A PARTICULAR PURPOSE ARE EXCLUDED.*

End of Report

8 APPENDICES

8.1 Alstom Letter Fritz Vogel – NLH Aug 3, 2000 "Predicted Performance Data For Boiler #1 & 2

RECEIVED
AUG 7 2000

ABB ALSTOM
POWER

2000 Metric

Customer Services Division
Canada
Aug.03.2000

Newfoundland and Labrador Hydro
P.O.Box 29
Holyrood, NF
A0A 2R0

Attention: Herb Dowden-George Moore-Ray Rossiter-John Mallam-Jerry Goulding
Mike Taylor-Alonzo Pollard-Bob Garland
Cc
Terry LeDrew

Dear Gentlemen:

Reference: Predicted Performance Data for Boiler #1 & 2

Our performance design engineer has completed the review of the performance data and attached you will find the following:

Two (2) Tables of Performance Data in metric units
One (1) Graph indicating the recommended Burner Tilt while operating the boiler

The data are based upon the same parameters as applied during the upgrade review and are in no way reflecting Station Data as the basis for recalculation.

Our engineering department emphasizes the fact that the burner tilt must be kept horizontal at all times while operating at less then Control Load i.e. 70 % of MCR.

To assure that these information winds up at the appropriate location I suggest that copies be included in every available instruction/operation instruction manual.

Should you find any discrepancies or have questions with regard to the data feel free to get in touch with John Adams or me.

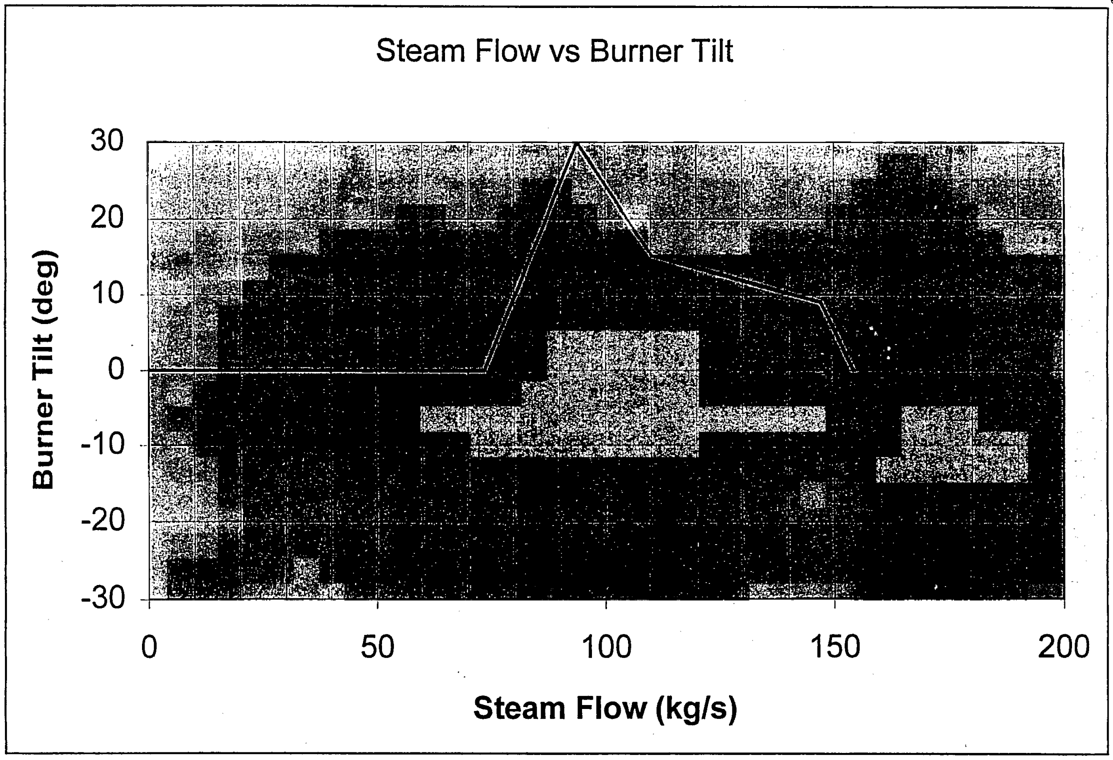
Yours truly,
F. Vogel
Fritz Vogel
ALSTOM Power Canada Inc.

P.O. Box 246
Upper Gullies, Newfoundland, Canada
A0A 4C0
Tel.: (709) 744-4666
Fax: (709) 744-4666

175 179.8 131.25

Load	%	MCR	VWO	75%	50%	25%
STEAM						
Steam Generated	kg/s	147.1	154.4	110.3	73.5	36.8
Reheater Flow	kg/s	131.7	138.1	99.5	67.0	33.9
Operating Pressures	kPa(g)					
Drum		14162	14286	13700	13287	13025
Primary SH Outlet		13976	14080	13590	13238	13011
Final SH Inlet		13907	14004	13550	13219	13006
Final SH Outlet		13480	13542	13300	13094	12969
Reheater Inlet		3399	3572	2558	1696	814
Reheater Outlet		3185	3337	2386	1579	752
Operating Temperatures	°C					
Primary SH Outlet		377	376	377	365	353
Final SH Inlet		370	376	363	365	353
Final SH Outlet		541	541	541	541	531
Reheater Inlet		353	358	329	308	289
Reheater Outlet		541	539	528	495	468
Design Pressures	kPa(g)					
Waterwalls & Headers		15203				
Superheater		15203				
Reheater		4254				
BOILER FEEDWATER						
Economizer flow	kg/s	147.6	157.3	108.9	75.5	38.4
Blowdown	kg/s	1.47	1.54	1.10	0.74	0.37
Operating Pressures	kPa(g)					
At Economizer Inlet (incl. static)		14348	14480	13845	13390	13107
Economizer Outlet		14162	14286	13700	13287	13025
Operating Temperatures	°C					
Economizer Inlet		240	243	225	205	174
Economizer Outlet		302	303	294	270	241
Design Pressure	kPa(g)					
Economizer		15548				
DESUPERHEATING WATER						
Source		Boiler Feed Pump				
Pressure at Pump Discharge	kPa(g)	16286	16286	16286	16286	16286
Temperature	°C	149	151	140	127	107
SH Spray Flow (Operating)	kg/s	2.3	0.0	3.8	0.0	0.0
SH Spray Flow (Design)	kg/s	13.86				
RH Spray Flow (Operating)	kg/s	0.0	0.0	0.0	0.0	0.0
RH Spray Flow (Design)	kg/s	5.27				
FLUE GAS						
Flow	kg/s					
Through Boiler - Economizer		159.8	166.7	134.6	94.9	53.0
Air Heater Inlet		159.8	166.7	134.6	94.9	53.0
Air Heater Outlet (corrected)		173.8	181.2	146.4	104.6	57.6
Operating Drafts	Pa(g)					
Furnace Outlet		2816	3063	1996	997	309
Final SH Outlet		2741	2982	1943	970	301
RH Outlet		2567	2792	1820	909	282
Economizer Outlet		1595	1735	1131	566	175
Air Heater Outlet		324	352	230	117	36

Load	%	MCR	VWO	75%	50%	25%
Operating Temperatures	°C					
Furnace Outlet		1421	1407	1333	1177	1008
Final SH Outlet		1124	1122	1051	919	764
Primary SH Outlet		570	575	536	475	407
RH Outlet		827	831	768	665	543
Economizer Outlet		323	327	299	262	214
Air Heater Outlet (uncorrected)		172	174	164	151	136
Air Heater Outlet (corrected)		163	165	156	144	133
Gas Velocities (Average)	m/sec					
SH Platen 1 - 12" transverse pitch		16.9	17.5	13.5	8.6	4.2
SH Platen 2 - 12" transverse pitch		16.5	17.1	13.1	8.4	4.1
SH Finish - 12" transverse pitch		15.1	15.7	12.0	7.7	3.7
RH Finish - 6" transverse pitch		17.1	17.8	13.6	8.6	4.2
RH Inlet - 6" transverse pitch		16.4	17.2	13.1	8.3	4.0
Primary SH - 4" transverse pitch		17.1	18.0	13.7	8.8	4.4
Economizer		12.5	13.2	10.1	6.6	3.4
AIR						
Flow	kg/s					
Air Heater Inlet		158.4	165.1	134.1	96.1	53.1
Air Heater Outlet (corrected)		144.4	150.6	122.3	86.5	48.5
Air to Burners		144.4	150.6	122.3	86.5	48.5
Operating Pressures	Pa(g)					
Air Heater Inlet		6006	6417	4642	2977	1262
Air Heater Outlet		5158	5495	4041	2678	1169
Windbox		4137	4384	3317	2317	1057
Operating Temperatures	°C					
Air Heater Inlet		54	52	63	76	90
Air Heater Outlet		233	234	222	204	179
Excess Air	%					
Leaving Furnace		5	5	15	20	30
Leaving Economizer		5	5	15	20	30
FUEL BURNT						
No. Burners in Service		12	12	12	8	8
#6 Fuel Oil (Total)	kg/s	10.99	11.46	8.50	5.76	2.98
#6 Fuel Oil (Per Burner)	kg/s	0.92	0.96	0.71	0.72	0.37
Burner Tilts	+ / - Deg	+9	0	+15	0	0
ATOMIZING STEAM						
No. Burners in Service		12	12	12	8	8
Flow (Total)	kg/s	0.961	0.957	0.998	0.680	0.736
Pressure	kPa(g)	724	724	724	724	724
Temperature	°C	193	193	193	193	193
HEAT BALANCE	%					
Dry Gas Loss		3.87	4	3.62	2.8	1.86
Moisture in Fuel		0	0	0	0	0
Moisture from Hydrogen		4.83	4.85	4.73	4.58	4.42
Moisture in Air		0.09	0.1	0.09	0.07	0.05
Carbon Loss		0	0	0	0	0
Radiation Loss		0.2	0.2	0.28	0.4	0.85
Unaccounted Loss		0.5	0.5	0.5	0.5	0.5
Manufacturers Margin		0.5	0.5	0.5	0.5	0.5
Total Losses		9.99	10.15	9.72	8.85	8.18
Overall Efficiency		90.01	89.85	90.28	91.15	91.82


Steam Flow vs Burner Tilt
30
20
10
0
-10
-20
-30
Burner Tilt (deg)
0
50
100
150
200
Steam Flow (kg/s)

8.2 Turbine Heat Balance Conditions Units #1 and #2 Uprated 1988

VWO at 1875 psig

NEWFOUNDLAND TB. NO. 940310+940311 8/5/88
TIR# 10236-893A, UPRATE
1875G-1000/1000F-1.5 IN. HGA

GROSS HEAT RATE = 7991 BTU/KWHR
GENERATOR OUTPUT = 181198 KW -- RATED 194445 KVA, .90 P.F., CONV COOLED
GENERATOR LOSS = 1864 KW AT .93 P.F., 45 PSIG H2, MECH LOSS = 609 KW
STEAM CONDITIONS 1875 PSIG, 1000/1000 F, 1.5 IN HGA 3600 RPM

	F LB/HR	P PSIA	T F	H BTU/LB
HEAT SOURCE				
STEAM FROM BOILER	1225560	1890.	1000.0	1477.70
BLOWDOWN	0			
WATER TO ATTEMPERATOR	0			286.54
FEEDWATER TO BOILER	1225560		470.2	453.82
STEAM FROM REHEATER	1095677	488.6		1520.66
STEAM TO REHEATER	1095677	542.9	681.3	1344.42
TURBINE				
STEAM TO THROTTLE	1225560	1890.	1000.0	1477.70
VALVE STEM LEAKAGE				
TO H.P. TURB. EXHAUST	1431	542.9		1477.70
TO STEAM SEAL REG.	999	16.70		1477.70
ENTERING 1-R CONTROL STAGE NO. 1	1223130	1856.		1477.70
ENTERING DIAPHRAGM STAGE NO. 2	1204254	1507.		1456.74
3-R PACKING				
LEAK-OFF TO HEATER NO. 4 EXTR.	6794	155.9		1344.26
SEAL FLOW TO STEAM SEAL REG.	2952	16.70		1344.26
VENT FLOW TO GLAND SEAL COND.	182			1344.26
BEFORE PRESSURE DROP	1194326	548.4		1344.26
BEFORE FLOW ENTRY	1194326	542.9	681.0	1344.26
BEFORE PRESSURE DROP	1095677	488.6		1520.66
BEFORE ENTRY OF LEAKAGE	1095677	478.9		1520.66
1-R PACKING				
FLOW FROM STAGE 1 SHELL	18876	1507.		1456.74
ENTERING DIAPHRAGM STAGE NO. 11	1114553	478.9		1519.57
ENTERING DIAPHRAGM STAGE NO. 14	1064554	259.5		1440.09
ENTERING DIAPHRAGM STAGE NO. 16	1034026	155.9		1380.28
2-R PACKING				
SEAL FLOW TO STEAM SEAL REG.	1800	16.70		1310.25
VENT FLOW TO GLAND SEAL COND.	287			1310.25
BEFORE PRESSURE DROP	995045	79.82		1310.25
MAIN FLOW DIVIDED BY 2 AT THIS POINT				
ENTERING DIAPHRAGM STAGE NO. 18	497523	78.23		1310.25
ENTERING DIAPHRAGM STAGE NO. 19	465096	46.04		1260.23
ENTERING DIAPHRAGM STAGE NO. 21	423646	12.68		1160.41
ENTERING COND. LAST STAGE NO. 22	423646	5.607		1109.39
BEFORE ENTRY OF LEAKAGE	423646	1.006		1044.37
2-R PACKING				
SEAL FLOW FROM STEAM SEAL REG.	1401	16.70		1356.80
VENT FLOW TO GLAND SEAL COND.	499			1356.80
BEFORE PRESSURE DROP	424096	1.006		1044.70
EXHAUST FLOW	424096	0.7367	91.7	1044.70

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HEATER NO. 6 (CLOSED WITH D.C.)				
CONDITIONS AT H.P. TURB. EXHAUST		542.9	681.3	1344.42
STEAM TO HEATER (5.0 PC DELTA P)	100080	515.8		1344.42
FEEDWATER LEAVING (0 DEG TTD)	1225560		470.2	453.82
FEEDWATER ENTERING DRAIN COOLER	1225560		397.9	375.38
DRAINS LEAVING D.C. (10 DEG TD)	100080	515.8	407.9	383.89
HEATER NO. 5 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		259.5	832.9	1440.09
STEAM TO HEATER (7.0 PC DELTA P)	49999	241.3		1440.09
FEEDWATER LEAVING (0 DEG TTD)	1225560		397.9	375.38
FEEDWATER ENTERING DRAIN COOLER	1225560		350.8	326.09
DRAINS ENTERING	100080			383.89
DRAINS LEAVING D.C. (10 DEG TD)	150079	241.3	360.8	333.24
HEATER NO. 4 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		155.9	707.3	1380.28
STEAM EXTRACTED FROM TURBINE	30528	155.9		1380.28
STEAM FROM 3-R PACKING LEAK	6794			1344.26
EXTRACTION STEAM (7.0 PC DELTA P)	37322	145.0		1373.72
FEEDWATER LEAVING (5 DEG TTD)	1225560		350.8	326.09
FEEDWATER ENTERING DRAIN COOLER	1225560		314.3	288.54
DRAINS ENTERING	150079			333.24
DRAINS LEAVING D.C. (10 DEG TD)	187401	145.0	324.3	294.91
FLOW FROM F.W. TO BOILER	0	2362.	314.3	288.54
FEEDWATER PUMP (12. BTU HEAT RISE)				
FEEDWATER LEAVING	1225560	2362.	314.3	288.54
FEEDWATER ENTERING	1225560		306.9	276.84
HEATER NO. 3 (OPEN)				
TURBINE SHELL CONDITIONS		79.82	558.2	1310.25
EXTRACTION STEAM (7 PC DELTA P)	36894	74.24		1310.25
FEEDWATER LEAVING	1225560	74.24	306.9	276.84
FEEDWATER ENTERING	1001265		266.4	235.38
DRAINS ENTERING	187401			294.91
HEATER NO. 2 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		46.04	450.3	1260.23
STEAM TO HEATER (7.0 PC DELTA P)	64854	42.82		1260.23
FEEDWATER LEAVING (5 DEG TTD)	1001265		266.4	235.38
FEEDWATER ENTERING DRAIN COOLER	1001265		196.8	165.09
DRAINS LEAVING D.C. (10 DEG TD)	64854	42.82	206.8	175.02

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HEATER NO. 1 (PUMPED DRAINS)				
TURBINE SHELL CONDITIONS		12.68	231.2	1160.41
STEAM EXTRACTED FROM TURBINE	82900			1160.41
STEAM FROM STEAM SEAL DUMP	4350			1356.80
STEAM TO HEATER (7 PC DELTA P)	87250	11.79		1170.20
FLOW FROM MAKEUP SOURCE	0			453.82
FLOW FROM FW. BELOW HEATER 1	0			453.82
DRAINS ENTERING	64854			175.02
DRAINS PUMPED TO FEEDWATER	152104	11.79	201.1	169.18
FEEDWATER AFTER DRAIN ENTRY	1001265		196.8	165.09
FEEDWATER LEAVING (5 DEG TTD)	849161		196.1	164.36
FEEDWATER ENTERING	849161		92.8	61.06
STEAM SEAL REGULATOR				
FLOW FROM VALVE STEM PACKING	999			1477.70
FLOW FROM 3-R PACKING SEAL	2952			1344.26
FLOW FROM 2-R PACKING SEAL	1800			1310.25
FLOW TO 2-R PACKING SEAL	1401			1356.80
MAKE-UP FROM TURBINE INLET	0			1477.70
DUMP TO HEATER NO. 1 EXTR	4350	12.68		1356.80
GLAND SEAL CONDENSER				
STEAM FROM 3-R PACKING VENT	182			1344.26
STEAM FROM 2-R PACKING VENT	287			1310.25
STEAM FROM 2-R PACKING VENT	499			1356.80
FEEDWATER LEAVING	849161		92.8	61.06
FEEDWATER ENTERING	849161		91.5	59.74
DRAINS TO CONDENSER	968			179.48
FLOW FROM F.W. TO HEATER NO. 1	0	11.79	91.5	453.82
FEEDWATER PUMP (0. BTU HEAT RISE)				
FEEDWATER LEAVING	849161	100.0	91.5	59.74
FEEDWATER ENTERING	849161		91.7	59.74
CONDENSER				
STEAM TO CONDENSER	424096	0.7367		1044.70
DRAINS ENTERING	968			
FEEDWATER LEAVING	849161	0.7367	91.7	59.74

RATING FLOW (GUARANTEED) IS 1167200 LB/HR AT INITIAL STEAM CONDITIONS OF 1875 PSIG, 1000 F. TO ASSURE THAT THE TURBINE WILL PASS THIS FLOW, CONSIDERING VARIATIONS IN FLOW COEFFICIENTS FROM EXPECTED VALUES, MANUFACTURING TOLERANCES ON DRAWING AREAS, ETC., WHICH MAY AFFECT THE FLOW, THE TURBINE IS BEING DESIGNED FOR AN EXPECTED FLOW OF 1225560 LB/HR.

CALCULATED DATA NOT GUARANTEED.

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MCR

NEWFOUNDLAND TB. NO. 940310+940311 6/5/88
TIR# 10236-893A, UPRATE
1875G-1000/1000F-1.5 IN. HGA

GROSS HEAT RATE = 7982 BTU/KWHR
GENERATOR OUTPUT = 174160 KW -- RATED 194445 KVA, .90 P.F., CONV COOLED
GENERATOR LOSS = 1864 KW AT .90 P.F., 45 PSIG H2, MECH LOSS = 609 KW
STEAM CONDITIONS 1875 PSIG, 1000/1000 F, 1.5 IN HGA 3600 RPM

	F LB/HR	P PSIA	T F	H BTU/LB
HEAT SOURCE				
STEAM FROM BOILER	1167200	1890.	1000.0	1477.70
BLOWDOWN	0			
WATER TO ATTEMPERATOR	0			285.40
FEEDWATER TO BOILER	1167200		465.4	448.44
STEAM FROM REHEATER	1044878	466.3		1521.31
STEAM TO REHEATER	1044878	518.1	672.0	1340.63
TURBINE				
STEAM TO THROTTLE	1167200	1890.	1000.0	1477.70
VALVE STEM LEAKAGE				
TO H.P. TURB. EXHAUST	1477	518.1		1477.70
TO STEAM SEAL REG.	953	16.70		1477.70
ENTERING 1-R CONTROL STAGE NO. 1	1164770	1860.		1477.70
ENTERING DIAPHRAGM STAGE NO. 2	1146779	1432.		1451.49
3-R PACKING				
LEAK-OFF TO HEATER NO. 4 EXTR.	6503	149.0		1340.46
SEAL FLOW TO STEAM SEAL REG.	2824	16.70		1340.46
VENT FLOW TO GLAND SEAL COND.	183			1340.46
BEFORE PRESSURE DROP	1137269	523.3		1340.46
BEFORE FLOW ENTRY	1137269	518.1	671.7	1340.46
BEFORE PRESSURE DROP	1044878	466.3		1521.31
BEFORE ENTRY OF LEAKAGE	1044878	457.0		1521.31
1-R PACKING				
FLOW FROM STAGE 1 SHELL	17991	1432.		1451.49
ENTERING DIAPHRAGM STAGE NO. 11	1062869	457.0		1520.13
ENTERING DIAPHRAGM STAGE NO. 14	1015914	247.8		1440.67
ENTERING DIAPHRAGM STAGE NO. 16	987378	149.0		1380.86
2-R PACKING				
SEAL FLOW TO STEAM SEAL REG.	1705	16.70		1310.80
VENT FLOW TO GLAND SEAL COND.	287			1310.80
BEFORE PRESSURE DROP	950578	76.31		1310.80
MAIN FLOW DIVIDED BY 2 AT THIS POINT				
ENTERING DIAPHRAGM STAGE NO. 18	475289	74.79		1310.80
ENTERING DIAPHRAGM STAGE NO. 19	444689	44.05		1260.78
ENTERING DIAPHRAGM STAGE NO. 21	405879	12.16		1160.98
ENTERING COND. LAST STAGE NO. 22	405879	5.369		1109.82
BEFORE ENTRY OF LEAKAGE	405879	0.9833		1045.02
2-R PACKING				
SEAL FLOW FROM STEAM SEAL REG.	1402	16.70		1355.10
VENT FLOW TO GLAND SEAL COND.	500			1355.10
BEFORE PRESSURE DROP	406330	0.9833		1045.36
EXHAUST FLOW	406330	0.7367	91.7	1045.36

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HEATER NO. 6 (CLOSED WITH D.C.)				
CONDITIONS AT H.P. TURB. EXHAUST		518.1	672.0	1340.63
STEAM TO HEATER (5.0 PC DELTA P)	93868	492.2		1340.63
FEEDWATER LEAVING (0 DEG TTD)	1167200		465.4	448.44
FEEDWATER ENTERING DRAIN COOLER	1167200		393.9	371.15
DRAINS LEAVING D.C. (10 DEG TD)	93868	492.2	403.9	379.54
HEATER NO. 5 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		247.8	833.2	1440.67
STEAM TO HEATER (7.0 PC DELTA P)	46955	230.4		1440.67
FEEDWATER LEAVING (0 DEG TTD)	1167200		393.9	371.15
FEEDWATER ENTERING DRAIN COOLER	1167200		347.2	322.43
DRAINS ENTERING	93868			379.54
DRAINS LEAVING D.C. (10 DEG TD)	140823	230.4	357.2	329.49
No. 5	46955			
HEATER NO. 4 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		149.0	707.8	1380.86
STEAM EXTRACTED FROM TURBINE	28536	149.0		1380.86
STEAM FROM 3-R PACKING LEAK	6503			1340.46
EXTRACTION STEAM (7.0 PC DELTA P)	35039	138.5		1373.36
FEEDWATER LEAVING (5 DEG TTD)	1167200		347.2	322.43
FEEDWATER ENTERING DRAIN COOLER	1167200		311.2	285.40
DRAINS ENTERING	140823			329.49
DRAINS LEAVING D.C. (10 DEG TD)	175862	138.5	321.2	291.70
No. 4	35039			
FLOW FROM F.W. TO BOILER	0	2362.	311.2	285.40
FEEDWATER PUMP (12. BTU HEAT RISE)				
FEEDWATER LEAVING	1167200	2362.	311.2	285.40
FEEDWATER ENTERING	1167200		303.9	273.70
HEATER NO. 3 (OPEN)				
TURBINE SHELL CONDITIONS		76.31	558.8	1310.80
EXTRACTION STEAM (7 PC DELTA P)	34809	70.97		1310.80
FEEDWATER LEAVING	1167200	70.97	303.9	273.70
FEEDWATER ENTERING	956529		263.7	232.64
DRAINS ENTERING	175862			291.70
HEATER NO. 2 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		44.05	451.0	1260.78
STEAM TO HEATER (7.0 PC DELTA P)	61200	40.97		1260.78
FEEDWATER LEAVING (5 DEG TTD)	956529		263.7	232.64
FEEDWATER ENTERING DRAIN COOLER	956529		194.8	163.04
DRAINS LEAVING D.C. (10 DEG TD)	61200	40.97	204.8	172.96

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HEATER NO. 1 (PUMPED DRAINS)				
TURBINE SHELL CONDITIONS		12.16	232.0	1160.98
STEAM EXTRACTED FROM TURBINE	77619			1160.98
STEAM FROM STEAM SEAL DUMP	4079			1355.10
STEAM TO HEATER (7 PC DELTA P)	81699	11.30		1170.67
FLOW FROM MAKEUP SOURCE	0			448.44
FLOW FROM FW. BELOW HEATER 1	0			448.44
DRAINS ENTERING	61200			172.96
DRAINS PUMPED TO FEEDWATER	142899	11.30	199.1	167.14
FEEDWATER AFTER DRAIN ENTRY	956529		194.8	163.04
FEEDWATER LEAVING (5 DEG TTD)	813630		194.1	162.32
FEEDWATER ENTERING	813630		92.8	61.12
STEAM SEAL REGULATOR				
FLOW FROM VALVE STEM PACKING	953			1477.70
FLOW FROM 3-R PACKING SEAL	2824			1340.46
FLOW FROM 2-R PACKING SEAL	1705			1310.80
FLOW TO 2-R PACKING SEAL	1402			1355.10
MAKE-UP FROM TURBINE INLET	0			1477.70
DUMP TO HEATER NO. 1 EXTR	4079	12.16		1355.10
GLAND SEAL CONDENSER				
STEAM FROM 3-R PACKING VENT	183			1340.46
STEAM FROM 2-R PACKING VENT	287			1310.80
STEAM FROM 2-R PACKING VENT	500			1355.10
FEEDWATER LEAVING	813630		92.8	61.12
FEEDWATER ENTERING	813630		91.5	59.74
DRAINS TO CONDENSER	970			179.48
FLOW FROM F.W. TO HEATER NO. 1	0	11.30	91.5	448.44
FEEDWATER PUMP (0. BTU HEAT RISE)				
FEEDWATER LEAVING	813630	100.0	91.5	59.74
FEEDWATER ENTERING	813630		91.7	59.74
CONDENSER				
STEAM TO CONDENSER	406330	0.7367		1045.36
DRAINS ENTERING	970			
FEEDWATER LEAVING	813630	0.7367	91.7	59.74

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75% MCR

NEWFOUNDLAND TB. NO. 940310+940311 8/5/88
TIR# 10236-893A, UPRATE
1875G-1000/1000F-1.5 IN. HGA

GROSS HEAT RATE = 7987 BTU/KWHR
GENERATOR OUTPUT = 135841 KW -- RATED 194445 KVA, .90 P.F., CONV COOLED
GENERATOR LOSS = 1566 KW AT .90 P.F., 20 PSIG H2, MECH LOSS = 609 KW
STEAM CONDITIONS 1875 PSIG, 1000/1000 F, 1.5 IN HGA 3600 RPM

	F LB/HR	P PSIA	T F	H BTU/LB
HEAT SOURCE				
STEAM FROM BOILER	875400	1890.	1000.0	1477.70
BLOWDOWN	0			
WATER TO ATTEMPERATOR	0			267.37
FEEDWATER TO BOILER	875400		437.9	418.26
STEAM FROM REHEATER	789636	353.6		1524.61
STEAM TO REHEATER	789636	392.9	629.9	1325.08
TURBINE				
STEAM TO THROTTLE	875400 693,373	1890.	1000.0	1477.70
VALVE STEM LEAKAGE				
TO H.P. TURB. EXHAUST	1707	392.9		1477.70
TO STEAM SEAL REG.	723	16.70		1477.70
ENTERING 1-R CONTROL STAGE NO. 1	872970	1873.		1477.70
ENTERING DIAPHRAGM STAGE NO. 2	859428	1061.		1429.55
3-R PACKING				
LEAK-OFF TO HEATER NO. 4 EXTR.	4992	113.8		1324.77
SEAL FLOW TO STEAM SEAL REG.	2147	16.70		1324.77
VENT FLOW TO GLAND SEAL COND.	186			1324.77
BEFORE PRESSURE DROP	852103	396.9		1324.77
BEFORE FLOW ENTRY	852103	392.9	629.3	1324.77
BEFORE PRESSURE DROP	789636	353.6		1524.61
BEFORE ENTRY OF LEAKAGE	789636	346.5		1524.61
1-R PACKING				
FLOW FROM STAGE 1 SHELL	13542	1061.		1429.55
ENTERING DIAPHRAGM STAGE NO. 11	803178	346.5		1523.01
ENTERING DIAPHRAGM STAGE NO. 14	770653	188.7		1443.67
ENTERING DIAPHRAGM STAGE NO. 16	751484	113.8		1383.86
2-R PACKING				
SEAL FLOW TO STEAM SEAL REG.	1216	16.70		1313.68
VENT FLOW TO GLAND SEAL COND.	286			1313.68
BEFORE PRESSURE DROP	728248	58.46		1313.68
MAIN FLOW DIVIDED BY 2 AT THIS POINT				
ENTERING DIAPHRAGM STAGE NO. 18	362624	57.29		1313.68
ENTERING DIAPHRAGM STAGE NO. 19	340875	33.89		1263.72
ENTERING DIAPHRAGM STAGE NO. 21	314735	9.473		1164.08
ENTERING COND. LAST STAGE NO. 22	314735	4.148		1112.09
BEFORE ENTRY OF LEAKAGE	314735	0.8826		1049.49
2-R PACKING				
SEAL FLOW FROM STEAM SEAL REG.	1408	16.70		1348.53
VENT FLOW TO GLAND SEAL COND.	503			1348.53
BEFORE PRESSURE DROP	315187	0.8826		1045.92
EXHAUST FLOW	315187	0.7367	91.7	1045.92

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HEATER NO. 6 (CLOSED WITH D.C.)				
CONDITIONS AT H.P. TURB. EXHAUST		392.9	629.9	1325.08
STEAM TO HEATER (5.0 PC DELTA P)	64174	373.3		1325.08
FEEDWATER LEAVING (0 DEG TTD)	875400		437.9	418.26
FEEDWATER ENTERING DRAIN COOLER	875400		371.0	347.14
DRAINS LEAVING D.C. (10 DEG TD)	64174	373.3	381.0	354.87
HEATER NO. 5 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		188.7	834.5	1443.67
STEAM TO HEATER (7.0 PC DELTA P)	32525	175.5		1443.67
FEEDWATER LEAVING (0 DEG TTD)	875400		371.0	347.14
FEEDWATER ENTERING DRAIN COOLER	875400		326.9	301.52
DRAINS ENTERING	64174			354.87
DRAINS LEAVING D.C. (10 DEG TD)	96698	175.5	336.9	308.14
HEATER NO. 4 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		113.8	710.2	1383.86
STEAM EXTRACTED FROM TURBINE	19169	113.8		1383.86
STEAM FROM 3-R PACKING LEAK	4992			1324.77
EXTRACTION STEAM (7.0 PC DELTA P)	24161	105.8		1371.65
FEEDWATER LEAVING (5 DEG TTD)	875400		326.9	301.52
FEEDWATER ENTERING DRAIN COOLER	875400	136	293.5	267.37
DRAINS ENTERING	96698			308.14
DRAINS LEAVING D.C. (10 DEG TD)	120860	105.8	303.5	273.35
FLOW FROM F.W. TO BOILER	0	2362.	293.5	267.37
FEEDWATER PUMP (12. BTU HEAT RISE)				
FEEDWATER LEAVING	875400	2362.	293.5	267.37
FEEDWATER ENTERING	875400		286.3	255.67
HEATER NO. 3 (OPEN)				
TURBINE SHELL CONDITIONS		58.46	561.9	1313.68
EXTRACTION STEAM (7 PC DELTA P)	24734	54.37		1313.68
FEEDWATER LEAVING	875400	54.37	286.3	255.67
FEEDWATER ENTERING	729806		248.2	216.88
DRAINS ENTERING	120860			273.35
HEATER NO. 2 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		33.89	454.7	1263.72
STEAM TO HEATER (7.0 PC DELTA P)	43499	31.52		1263.72
FEEDWATER LEAVING (5 DEG TTD)	729806	110	248.2	216.88
FEEDWATER ENTERING DRAIN COOLER	729806	73	182.9	151.16
DRAINS LEAVING D.C. (10 DEG TD)	43499	31.52	192.9	161.03

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112 MW

HEATER NO. 1 (PUMPED DRAINS)				
TURBINE SHELL CONDITIONS		9.473	236.6	1164.08
STEAM EXTRACTED FROM TURBINE	52280			1164.08
STEAM FROM STEAM SEAL DUMP	2678			1348.53
STEAM TO HEATER (7 PC DELTA P)	54958	8.810		1173.07
FLOW FROM MAKEUP SOURCE	0			418.26
FLOW FROM FW. BELOW HEATER 1	0			418.26
DRAINS ENTERING	43499			161.03
DRAINS PUMPED TO FEEDWATER	98457	8.810	187.3	155.31
FEEDWATER AFTER DRAIN ENTRY	729806		182.9	151.16
FEEDWATER LEAVING (5 DEG TTD)	631350	73	182.3	150.51
FEEDWATER ENTERING	631350	26	93.2	61.52
STEAM SEAL REGULATOR				
FLOW FROM VALVE STEM PACKING	723			1477.70
FLOW FROM 3-R PACKING SEAL	2147			1324.77
FLOW FROM 2-R PACKING SEAL	1216			1313.68
FLOW TO 2-R PACKING SEAL	1408			1348.53
MAKE-UP FROM TURBINE INLET	0			1477.70
DUMP TO HEATER NO. 1 EXTR	2678	9.473		1348.53
GLAND SEAL CONDENSER				
STEAM FROM 3-R PACKING VENT	186			1324.77
STEAM FROM 2-R PACKING VENT	286			1313.68
STEAM FROM 2-R PACKING VENT	503			1348.53
FEEDWATER LEAVING	631350		93.2	61.52
FEEDWATER ENTERING	631350		91.5	59.74
DRAINS TO CONDENSER	975			179.48
FLOW FROM F.W. TO HEATER NO. 1	0	8.810	91.5	418.26
FEEDWATER PUMP (0. BTU HEAT RISE)				
FEEDWATER LEAVING	631350	100.0	91.5	59.74
FEEDWATER ENTERING	631350		91.7	59.74
CONDENSER				
STEAM TO CONDENSER	315187	0.7367		1049.92
DRAINS ENTERING	975			
FEEDWATER LEAVING	631350	0.7367	91.7	59.74

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NEWFOUNDLAND TB. NO. 940310+940311 8/5/88
TIR# 10236-893A, UPRATE
1875G-1000/1000F-1.5 IN. HGA

GROSS HEAT RATE = 8109 BTU/KWHR
GENERATOR OUTPUT = 93140 KW --- RATED 194445 KVA, .90 P.F., CONV COOLED
GENERATOR LOSS = 1092 KW AT .90 P.F., .50 PSIG H2, MECH LOSS = 609 KW
STEAM CONDITIONS 1875 PSIG, 1000/1000 F, 1.5 IN HGA 3600 RPM

	F LB/HR	P PSIA	T F	H BTU/LB
HEAT SOURCE				
STEAM FROM BOILER	583600	1890.	1000.0	1477.70
BLOWDOWN	0			
WATER TO ATTEMPERATOR	0			243.38
FEEDWATER TO BOILER	583600		401.7	379.49
STEAM FROM REHEATER	531457	238.9		1527.96
STEAM TO REHEATER	531457	265.5	590.6	1312.75
TURBINE				
STEAM TO THROTTLE	583600	1890.	1000.0	1477.70
VALVE STEM LEAKAGE				
TO H.P. TURB. EXHAUST	1942	265.5		1477.70
TO STEAM SEAL REG.	488	16.70		1477.70
ENTERING 1-R CONTROL STAGE NO. 1	581170	1882.		1477.70
ENTERING DIAPHRAGM STAGE NO. 2	572116	699.9		1411.47
3-R PACKING				
LEAK-OFF TO HEATER NO. 4 EXTR.	3412	77.52		1312.19
SEAL FLOW TO STEAM SEAL REG.	1412	16.70		1312.19
VENT FLOW TO GLAND SEAL COND.	188			1312.19
BEFORE PRESSURE DROP	567104	268.2		1312.19
BEFORE FLOW ENTRY	567104	265.5	589.6	1312.19
BEFORE PRESSURE DROP	531457	238.9		1527.96
BEFORE ENTRY OF LEAKAGE	531457	234.1		1527.96
1-R PACKING				
FLOW FROM STAGE 1 SHELL	9054	699.9		1411.47
ENTERING DIAPHRAGM STAGE NO. 11	540511	234.1		1526.00
ENTERING DIAPHRAGM STAGE NO. 14	520981	128.1		1446.88
ENTERING DIAPHRAGM STAGE NO. 16	510064	77.52		1387.12
2-R PACKING				
SEAL FLOW TO STEAM SEAL REG.	694	16.70		1316.86
VENT FLOW TO GLAND SEAL COND.	285			1316.86
BEFORE PRESSURE DROP	493763	39.99		1316.86
MAIN FLOW DIVIDED BY 2 AT THIS POINT				
ENTERING DIAPHRAGM STAGE NO. 18	246882	39.19		1316.86
ENTERING DIAPHRAGM STAGE NO. 19	233427	23.30		1267.02
ENTERING DIAPHRAGM STAGE NO. 21	218791	6.624		1167.74
ENTERING COND. LAST STAGE NO. 22	218791	2.868		1114.65
BEFORE ENTRY OF LEAKAGE	218791	0.8069		1058.92
2-R PACKING				
SEAL FLOW FROM STEAM SEAL REG.	1411	16.70		1344.61
VENT FLOW TO GLAND SEAL COND.	505			1344.61
BEFORE PRESSURE DROP	219245	0.8069		1059.51
EXHAUST FLOW	219245	0.7367	91.7	1059.51

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SEP-28-1988 12:14 FROM GE CAN RTS BLD.101 PBO TO

HEATER NO. 6 (CLOSED WITH D.C.)				
CONDITIONS AT H.P. TURB. EXHAUST		265.5	590.6	1312.75
STEAM TO HEATER (5.0 PC DELTA P)	37588	252.2		1312.75
FEEDWATER LEAVING (0 DEG TTD)	583600		401.7	379.49
FEEDWATER ENTERING DRAIN COOLER	583600		340.7	315.73
DRAINS LEAVING D.C. (10 DEG TD)	37588	252.2	350.7	322.71
HEATER NO. 5 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		128.1	836.2	1446.88
STEAM TO HEATER (7.0 PC DELTA P)	19530	119.2		1446.88
FEEDWATER LEAVING (0 DEG TTD)	583600		340.7	315.73
FEEDWATER ENTERING DRAIN COOLER	583600		299.9	273.93
DRAINS ENTERING	37588			322.71
DRAINS LEAVING D.C. (10 DEG TD)	57118	119.2	309.9	280.03
HEATER NO. 4 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		77.52	713.1	1387.12
STEAM EXTRACTED FROM TURBINE	10917	77.52		1387.12
STEAM FROM 3-R PACKING LEAK	3412			1312.19
EXTRACTION STEAM (7.0 PC DELTA P)	14329	72.10		1369.28
FEEDWATER LEAVING (5 DEG TTD)	583600		299.9	273.93
FEEDWATER ENTERING DRAIN COOLER	583600		269.8	243.38
DRAINS ENTERING	57118			280.03
DRAINS LEAVING D.C. (10 DEG TD)	71447	72.10	279.8	248.99
FLOW FROM F.W. TO BOILER	0	2362.	269.8	243.38
FEEDWATER PUMP (12. BTU HEAT RISE)				
FEEDWATER LEAVING	583600	2362.	269.8	243.38
FEEDWATER ENTERING	583600		262.9	231.68
HEATER NO. 3 (OPEN)				
TURBINE SHELL CONDITIONS		39.99	565.6	1316.86
EXTRACTION STEAM (7 PC DELTA P)	15322	37.19		1316.86
FEEDWATER LEAVING	583600	37.19	262.9	231.68
FEEDWATER ENTERING	496831		227.3	195.73
DRAINS ENTERING	71447			248.99
HEATER NO. 2 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		23.30	459.1	1267.02
STEAM TO HEATER (7.0 PC DELTA P)	26909	21.67		1267.02
FEEDWATER LEAVING (5 DEG TTD)	496831		227.3	195.73
FEEDWATER ENTERING DRAIN COOLER	496831		166.8	134.95
DRAINS LEAVING D.C. (10 DEG TD)	26909	21.67	176.8	144.78

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HEATER NO. 1 (PUMPED DRAINS)				
TURBINE SHELL CONDITIONS		6.624	242.6	1167.74
STEAM EXTRACTED FROM TURBINE	29272			1167.74
STEAM FROM STEAM SEAL DUMP	1183			1344.61
STEAM TO HEATER (7 PC DELTA P)	30454	6.160		1174.61
FLOW FROM MAKEUP SOURCE	0			379.49
FLOW FROM FW. BELOW HEATER 1	0			379.49
DRAINS ENTERING	26909			144.78
DRAINS PUMPED TO FEEDWATER	57363	6.160	171.2	139.18
FEEDWATER AFTER DRAIN ENTRY	496831		166.8	134.95
FEEDWATER LEAVING (5 DEG TTD)	439468		166.2	134.40
FEEDWATER ENTERING	439468		94.0	62.30
STEAM SEAL REGULATOR				
FLOW FROM VALVE STEM PACKING	488			1477.70
FLOW FROM 3-R PACKING SEAL	1412			1312.19
FLOW FROM 2-R PACKING SEAL	694			1316.86
FLOW TO 2-R PACKING SEAL	1411			1344.61
MAKE-UP FROM TURBINE INLET	0			1477.70
DUMP TO HEATER NO. 1 EXTR	1183	6.624		1344.61
GLAND SEAL CONDENSER				
STEAM FROM 3-R PACKING VENT	188			1312.19
STEAM FROM 2-R PACKING VENT	285			1316.86
STEAM FROM 2-R PACKING VENT	505			1344.61
FEEDWATER LEAVING	439468		94.0	62.30
FEEDWATER ENTERING	439468		91.5	59.74
DRAINS TO CONDENSER	979			179.48
FLOW FROM F.W. TO HEATER NO. 1	0	6.160	91.5	379.49
FEEDWATER PUMP (0. BTU HEAT RISE)				
FEEDWATER LEAVING	439468	100.0	91.5	59.74
FEEDWATER ENTERING	439468		91.7	59.74
CONDENSER				
STEAM TO CONDENSER	219245	0.7367		1059.51
DRAINS ENTERING	979			
FEEDWATER LEAVING	439468	0.7367	91.7	59.74

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25% MCR

NEWFOUNDLAND TB. NO. 940310+940311
TIR# 10236-893A, UPRATE
1875G-1000/1000F-1.5 IN. HGA

8/5/88

GROSS HEAT RATE = 8752 BTU/KWHR
GENERATOR OUTPUT = 45259 KW --- RATED 194445 KVA, .90 P.F., CONV COOLED
GENERATOR LOSS = 817 KW AT .90 P.F., .50 PSIG H2, MECH LOSS = 609 KW
STEAM CONDITIONS 1875 PSIG, 1000/1000 F, 1.5 IN HGA 3600 RPM

	F LB/HR	P PSIA	T F	H BTU/LB
HEAT SOURCE				
STEAM FROM BOILER	291800	1890.	1000.0	1477.70
BLOWDOWN	0			
WATER TO ATTEMPERATOR	0		174 C	205.89
FEEDWATER TO BOILER	291800		346.3	321.46
STEAM FROM REHEATER	268940	121.5		1531.37
STEAM TO REHEATER	268940	135.0	572.3 300	1313.12
TURBINE				
STEAM TO THROTTLE	291344	1890.	1000.0	1477.70
VALVE STEM LEAKAGE				
TO H.P. TURB. EXHAUST	2182	135.0		1477.70
TO STEAM SEAL REG.	248	16.70		1477.70
ENTERING 1-R CONTROL STAGE NO. 1	288914	1888.		1477.70
ENTERING DIAPHRAGM STAGE NO. 2	284403	348.7		1408.92
3-R PACKING				
LEAK-OFF TO HEATER NO. 4 EXTR.	1775	39.88		1311.84
SEAL FLOW TO STEAM SEAL REG.	583	16.70		1311.84
VENT FLOW TO GLAND SEAL COND.	189			1311.84
BEFORE PRESSURE DROP	281857	136.3		1311.84
BEFORE FLOW ENTRY	281857	135.0	569.8	1311.84
BEFORE PRESSURE DROP	268940	121.5		1531.37
BEFORE ENTRY OF LEAKAGE	268940	119.1		1531.37
1-R PACKING				
FLOW FROM STAGE 1 SHELL	4511	348.7		1408.92
ENTERING DIAPHRAGM STAGE NO. 11	273450	119.1		1529.35
ENTERING DIAPHRAGM STAGE NO. 14	265148	65.57		1450.61
ENTERING DIAPHRAGM STAGE NO. 16	261049	39.88		1391.03
2-R PACKING				
SEAL FLOW TO STEAM SEAL REG.	49	16.70		1320.80
VENT FLOW TO GLAND SEAL COND.	284			1320.80
BEFORE PRESSURE DROP	253909	20.69		1320.80
MAIN FLOW DIVIDED BY 2 AT THIS POINT				
ENTERING DIAPHRAGM STAGE NO. 18	126954	20.27		1320.80
ENTERING DIAPHRAGM STAGE NO. 19	121021	12.14		1271.18
ENTERING DIAPHRAGM STAGE NO. 21	116204	3.546		1172.79
ENTERING COND. LAST STAGE NO. 22	116204	1.520		1118.34
BEFORE ENTRY OF LEAKAGE	116204	0.7571		1087.86
2-R PACKING				
SEAL FLOW FROM STEAM SEAL REG.	1336	16.70		1399.65
VENT FLOW TO GLAND SEAL COND.	479			1399.65
BEFORE PRESSURE DROP	116633	0.7571		1089.01
EXHAUST FLOW	116633	0.7367	91.7	1089.01

Nfld. & Labrador Hydro
ENGINEERING & CONST.
AUG 31 1988
ST. JOHN'S, NFLD.

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HEATER NO. 6 (CLOSED WITH D.C.)				
CONDITIONS AT H.P. TURB. EXHAUST		135.0	572.3	1313.12
STEAM TO HEATER (5.0 PC DELTA P)	15099	128.2		1313.12
FEEDWATER LEAVING (0 DEG TTD)	291800		346.3	321.46
FEEDWATER ENTERING DRAIN COOLER	291800		293.8	267.68
DRAINS LEAVING D.C. (10 DEG TD)	15099	128.2	303.8	273.71
HEATER NO. 5 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		65.57	838.8	1450.61
STEAM TO HEATER (7.0 PC DELTA P)	8302	60.98		1450.61
FEEDWATER LEAVING (0 DEG TTD)	291800		293.8	267.68
FEEDWATER ENTERING DRAIN COOLER	291800		257.7	231.22
DRAINS ENTERING	15099			273.71
DRAINS LEAVING D.C. (10 DEG TD)	23401	60.98	267.7	236.66
NO.5	8302			
HEATER NO. 4 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		39.88	717.2	1391.03
STEAM EXTRACTED FROM TURBINE	4100	39.88		1391.03
STEAM FROM 3-R PACKING LEAK	1775			1311.84
EXTRACTION STEAM (7.0 PC DELTA P)	5875	37.09		1367.10
FEEDWATER LEAVING (5 DEG TTD)	291800		257.7	231.22
FEEDWATER ENTERING DRAIN COOLER	291800		232.5	205.89
DRAINS ENTERING	23401			236.66
DRAINS LEAVING D.C. (10 DEG TD)	29276	37.09	242.5	210.99
FLOW FROM F.W. TO BOILER NO.4	5875	2362.	232.5	205.89
FEEDWATER PUMP (12. BTU HEAT RISE)				
FEEDWATER LEAVING	291800	2362.	232.5	205.89
FEEDWATER ENTERING	291800		225.9	194.19
HEATER NO. 3 (OPEN)				
TURBINE SHELL CONDITIONS		20.69	570.9	1320.80
EXTRACTION STEAM (7 PC DELTA P)	6808	19.24		1320.80
FEEDWATER LEAVING	291800	19.24	225.9	194.19
FEEDWATER ENTERING	255717		194.0	162.27
DRAINS ENTERING	29276			210.99
HEATER NO. 2 (CLOSED WITH D.C.)				
TURBINE SHELL CONDITIONS		12.14	465.4	1271.18
STEAM TO HEATER (7.0 PC DELTA P)	11866	11.29		1271.18
FEEDWATER LEAVING (5 DEG TTD)	255717		194.0	162.27
FEEDWATER ENTERING DRAIN COOLER	255717		140.6	108.79
DRAINS LEAVING D.C. (10 DEG TD)	11866	11.29	150.6	118.57

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HEATER NO. 1 (PUMPED DRAINS)				
TURBINE SHELL CONDITIONS		3.546	251.5	1172.79
STEAM EXTRACTED FROM TURBINE	9635			1172.79
STEAM FROM STEAM SEAL DUMP	0			1399.65
STEAM TO HEATER (7 PC DELTA P)	9635	3.298		1172.79
FLOW FROM MAKEUP SOURCE	0			321.46
FLOW FROM FW. BELOW HEATER 1	0			321.46
DRAINS ENTERING	11866			118.57
DRAINS PUMPED TO FEEDWATER	21501	3.298	145.2	113.15
FEEDWATER AFTER DRAIN ENTRY	255717		140.6	108.79
FEEDWATER LEAVING (5 DEG TTD)	234216		140.2	108.39
FEEDWATER ENTERING	234216		96.3	64.53
STEAM SEAL REGULATOR				
FLOW FROM VALVE STEM PACKING	248			1477.70
FLOW FROM 3-R PACKING SEAL	583			1311.84
FLOW FROM 2-R PACKING SEAL	49			1320.80
FLOW TO 2-R PACKING SEAL	1336			1399.65
MAKE-UP FROM TURBINE INLET	456			1477.70
DUMP TO HEATER NO. 1 EXTR	0	3.546		1399.65
GLAND SEAL CONDENSER				
STEAM FROM 3-R PACKING VENT	189			1311.84
STEAM FROM 2-R PACKING VENT	284			1320.80
STEAM FROM 2-R PACKING VENT	479			1399.65
FEEDWATER LEAVING	234216		96.3	64.53
FEEDWATER ENTERING	234216		91.5	59.74
DRAINS TO CONDENSER	951			179.48
FLOW FROM F.W. TO HEATER NO. 1	0	3.298	91.5	321.46
FEEDWATER PUMP (0. BTU HEAT RISE)				
FEEDWATER LEAVING	234216	100.0	91.5	59.74
FEEDWATER ENTERING	234216		91.7	59.74
CONDENSER				
STEAM TO CONDENSER	116633	0.7367		1089.01
DRAINS ENTERING	951			
FEEDWATER LEAVING	234216	0.7367	91.7	59.74

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8.3 B&W Boiler Performance Data Sheet (C/7391)

Form 220

A FUEL AS FIRED

SOURCE	Samples		
	Analyses by Customer		
SOLID	Type		
ASTM	Class		
	Group		
SOURCE	Mine		
	Seam		
	District		
	County		
	Prov. or State		
	Size – in.		
	Grindability – Hgl		
	Max. Total Moisture – %		
Prox. Analysis % by Weight	Moisture		
	Volatile Matter		
	Fixed Carbon		
	Sulphur		
	Ash		
	Total		
A.S.T.M. Fusibility of Ash–°F	Initial Def. Red./Ox.		
	Softening, Spherical Red./Ox.		
	Softening, Hemispherical Red./Ox.		
	Fluid Red./Ox.		
LIQUID	Type Bunker "C"		Oil
	Req'd. Viscosity at Bnr.–SSU	140	
	Req'd. Fuel Pres. at Bnr.–psig	250	
	Req'd. Stm. Pres. at Bnr.–psig	150	
GASEOUS	Type		
	Sp. Gr. (Rel. to Air)		
	Req'd. Fuel Pres. at Bnr.–psig		
ULTIMATE ANALYSIS	Fuel Type Bunker "C"		Oil
	% By	Weight	
	O_2	0.30	
	N_2	0.60	
	S	2.30	
	H_2	10.80	
	C	85.80	
	CO		
	CO_2		
	CH_4		
	C_2H_4		
	C_2H_6		
	C_3H_8		
	C_4H_{10}		
	C_6H_{12}		
	C_6H_{14}		
	SO_2		
	H_2O	0.10	
	ASH	0.10	
	Total	100.00	
	HHV –Btu/Lb.	18450	
	HHV –Btu/SCF		
	(At °F, in Hg.)		

Babcock & Wilcox Canada Ltd.
Cambridge, Ontario.

By: IT | App'd: IT | Date: 9/5/78 | Page: 1
Ref. No.: | Cont. No.: 7391

B EXPECTED PERFORMANCE

		VWO	THO	MCR	75%	50%	25%
	Load Condition	VWO	THO	MCR	75%	50%	25%
	Fuel Type	Bunker C	Bunker C	Bunker C	Bunker C	BunkerC	Bunker C
	Excess Air Leaving %	3	3	3	3	12	20
	No. of Burners In Operation	9	9	9	6	6	3
	Fuel Heat Input – MKB/Hr.	1452.4	1477.8	1329.3	1012.2	683.5	387.5
	Heat Avail. MKB/Hr.(Fuel & Heated Air)	1589.0	1611.5	1454.3	1111.8	758.0	421.1
QUANTITIES MLb./Hr.	Steam Lvg. Sec. Superheater	1072.2	1020.9	960.6	696.6	476.5	264.5
	Steam Lvg. Reheater	963.7	993.0	865.7	631.6	435.2	242.5
	Aux. Steam From Prim. Outlet	30.0	30.0	30.0	30.0	30.0	30.0
	Cont. Blowdown	10.7	10.2	9.6	7.0	4.8	2.6
	Fuel	78.72	80.10	72.05	54.86	37.59	21.0
	Flue Gas Ent. Prim. Airheater(s)	–	–	–	–	–	–
	Flue Gas Ent. Sec. Airheater(s)	1182.4	1203.1	1082.2	823.3	610.6	364.0
	Air Lvg. Prim. Airheater(s)	–	–	–	–	–	–
	Air Lvg. Sec. Airheater(s)	1125.0	1144.7	1029.7	783.3	584.0	349.6
	Prim. A.H. Leakage	–	–	–	–	–	–
	Sec.A.H. Leakage	109.0	107.0	103.0	93.6	69.0	34.0
PRESSURES PSIG	Steam at Sec. S.H. Outlet	1910	1900	1890	1855	1834	1821
	Steam at Reheater Inlet	542	558	487	351	238	127
	Drum Operating	2050	2027	2002	1914	1862	1830
PRESSURE DROP–psi	Drum to Sec. S.H. Outlet	140	127	112	59	28	9
	Reheater	18	19	16	11	8	4
	Economizer & Hyd. Head	20	20	19	18	17	16
TEMPERATURES °F – STEAM	Leaving Sec. S.H.	1005	1005	1005	1005	965	900
	Leaving Reheater	1005	1005	1005	1005	965	900
	Entering Reheater	704	713	683	638	637	655
Water	Entering Economizer	476	412	464	433	400	352
	Leaving Economizer	513	463	502	475	445	404
FLUE GAS	Ent. Prim./Sec. A.H.	680	661	662	619	570	507
	Leaving Prim. A.H. Unc./Cor.	–	–	–	–	–	–
	Leaving Sec. A.H. Unc./Cor.	308/290	301/284	297/280	285/267	286/275	266/257
AIR	Entering Unit	80	80	80	80	80	80
	Entering Prim./Sec. A.H.	–	–	–	–	–	–
	Leaving Prim./Sec. A.H.	501	487	492	482	464	434
RESISTANCES in. H_2O – GAS	Furnace & Convection Banks	7.57	7.83	6.32	4.15	2.02	0.72
	Flues & Dampers Prim./Sec.	2.48	2.54	2.05	1.34	0.66	0.24
	Airheater Prim./Sec.	3.55	3.60	3.05	2.10	1.15	0.50
	Total Draft Loss	13.60	13.97	11.42	7.59	3.83	1.46
AIR	Burners & Windbox	5.80	6.00	4.86	7.04	3.42	1.23
	Ducts, Dampers & AFMD Prim./Sec.	6.38	6.59	5.36	3.48	1.75	0.62
	Airheater Prim./Sec.	2.75	2.80	2.35	1.70	0.90	0.35
	Pulverizers & Coal piping						
	Total Air resistance Prim./Sec.	14.93	15.39	12.57	12.22	6.07	2.20
HEAT LOSS %	Dry Gas	4.19	4.20	4.14	3.86	3.34	2.74
	H_2 & H_2O In Fuel	5.95	5.95	5.94	5.88	5.81	5.71
	Moisture In Air	0.11	0.11	0.11	0.10	0.08	0.07
	Unburned Combustable	–	–	–	–	–	–
	Radiation	0.20	0.21	0.22	0.29	0.42	0.75
	Unacc. Loss & Mfg. Margin	1.00	1.00	1.00	1.00	1.00	1.00
	Total Heat Loss	11.45	11.47	11.41	11.13	10.65	10.27
	Efficiency of Unit – %	88.55	88.53	88.59	88.87	89.35	89.73
	Max. Allow. Boiler Water Conc. – PPM	500					
	Solids in Steam – PPM	0.5					
Pulv., Fan, & Feeder	No. Pulverizers Operating						
	Total Power–KW Hr./Ton Coal Pulv.						
	Fineness, % Thru 200 U.S.S. Sieve						
	Fineness, % Thru 50 U.S.S. Sieve						

EXPECTED PERFORMANCE IS BASED ON COMBUSTION AIR ENTERING UNIT AT °F, % RELATIVE HUMIDITY WITH LB. MOISTURE/LB. DRY AIR, IN. Hg. BAROMETRIC PRESSURE, ON CONDITIONS AND EQUIPMENT GIVEN ON THIS SUMMARY SHEET, AND ARRANGEMENT AS SHOWN ON DRAWING

C EQUIPMENT

UNIT	Type: Radiant; 150 MW; St. Drum I.D. 66 In.	1
	No. of Units: One (1)	2
	Design Pressure: 2200 psi	3
	Furnace Width x Depth: 30.5 Ft. x 28.0 Ft.	4
	Furnace Volume: 56160 $Ft.^3$	5
Super-Heater	Type: 2 Stage Drainable; Make: B & W Can. Ltd.	6
	Gas Pass 2 Steam Pass	7
	Steam Temp. Control: Interstage Attemperation	8
REHEATER	Type: Drainable; Make: B & W Can. Ltd.	9
	1 Gas Pass 1 Steam Pass	10
	Design Pressure: 650 psi	11
	Steam Temp. Control: Attemperation	12
ECONOMIZER	Type: Drainable; Make: B&W Canada Ltd.	13
	1 Gas Pass 1 Steam Pass	14
	Design Pressure, Headers/Tubes: 2250 / 2250 psi	15
		16
AIR HEATER	Type: Regenerative; Make: Ljungstrom	17
	1 Gas Pass 1 Air Pass; No. Per Unit: 2	18
	22½ VIRX – 44 Case 1	19
	Type: ; Make:	20
	Gas Pass Air Pass; No. Per Unit:	21
		22
Burner	Type: Parallel; Make: B & W Can. Ltd.	23
	No. Per Unit: 9; Size:	24
	Steam Atomizing – Low Excess Air	25
Pulverizer	Type: ; Size: ; No. Per Unit:	26
	No. Outlets Per Pulv.: ; Diameter: In.	27
	Feeder Type: ; Feeder Inlet/Outlet Dia.: In.	28

PRESSURE PART HEATING SURFACE Ft^2 – FURNACE & RADIANT (FLAT PROJECTED)

Area \ Tube Type	2½ : 3 Membrane				Total	
Front Wall	2470				2470	31
Rear Wall	2722				2722	32
Side Walls	3682				3682	33
Roof						34
Division Wall						35
Platens						36
						37
Total:	8874				8874	38

CONVECTION

Superheater (Full Circ.):	51743	39
Reheater (Full Circ.):	15440	40
Economizer (Full Circ.):	12821	41
Sat. Enclosure (½ Full Circ.)	–	42
Screen (Full Circ.):	–	43
Total:	89040	44
Total furnace, Radiant, and Convection Heating Surface:	17910	45

F.D. FAN	Net: 625,145 Lb./Hr.	31.40 in. H_2O	80 °F		46
	T.B.: 801,200 Lb./Hr.	44.64 in. H_2O	105 °F		47
	Type: Centrifugal			No./Unit: 2	48
I.D. FAN	Net: Lb./Hr.	in. H_2O	°F		49
	T.B.: Lb./Hr.	in. H_2O	°F		50
	Type:			No./Unit:	51
P.A. FAN	Net: Lb./Hr.	in. H_2O	°F		52
	T.B.: Lb./Hr.	in. H_2O	°F		53
	Type:			No./Unit:	54
FAN	Net: Lb./Hr.	in. H_2O	°F		55
	T.B.: Lb./Hr.	in. H_2O	°F		56
	Type:			No./Unit:	57
Stack	Height Above Grade: Ft.	Dia. at Top: Ft.			58
	Construction:				59

DRIVES

Equipment	Power Basis	No./Unit	H.P.	R.P.M.	Type	Coupling	Volts	Hz.	Ph.	
F.D. Fan	Motor	2	1600	1200						61
I.D. Fan										62
										63
										64
Pulverizers										65
										66
										67

REVISIONS

No.	Description	By	Date	
				69
				70
				71
				72

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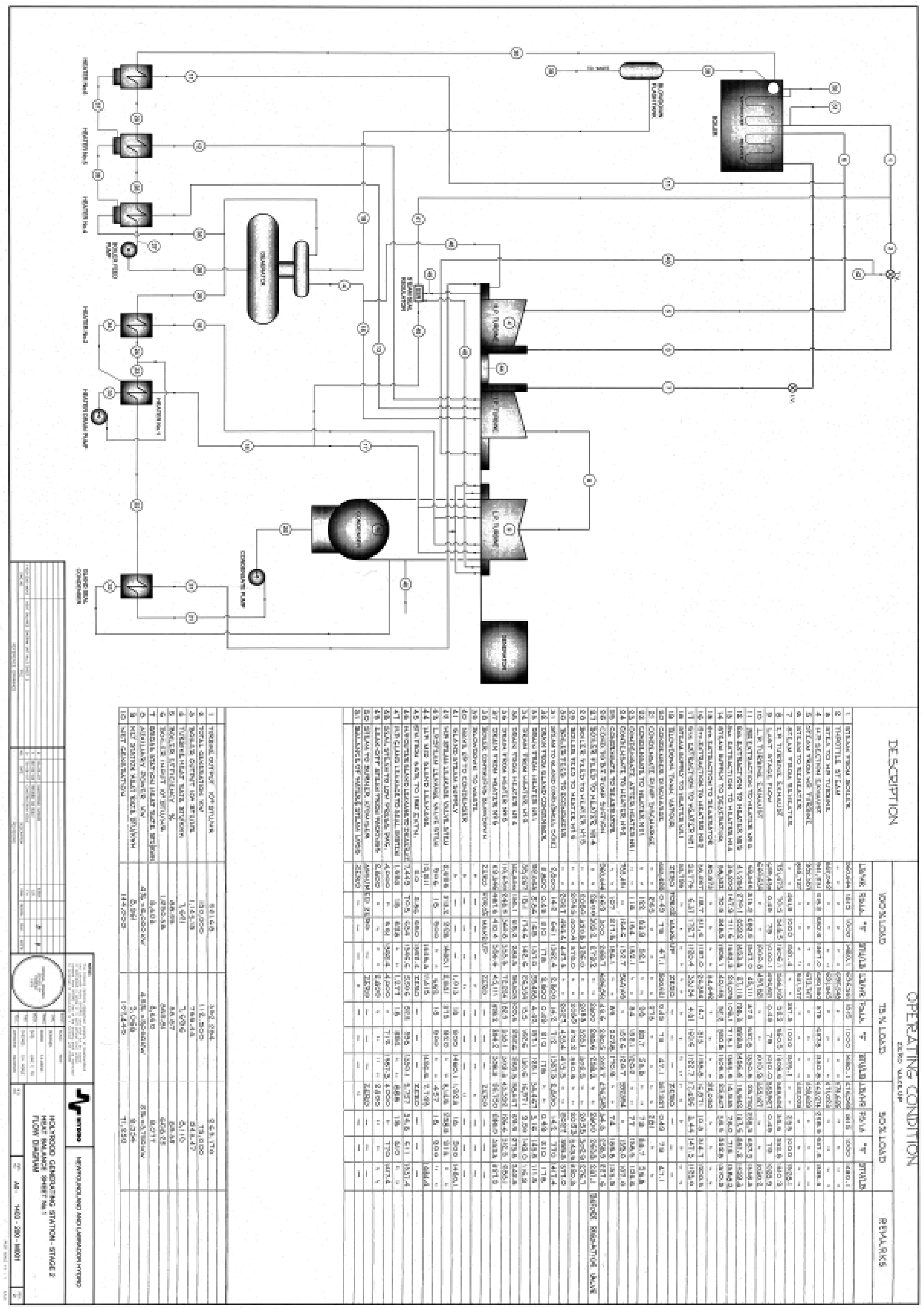
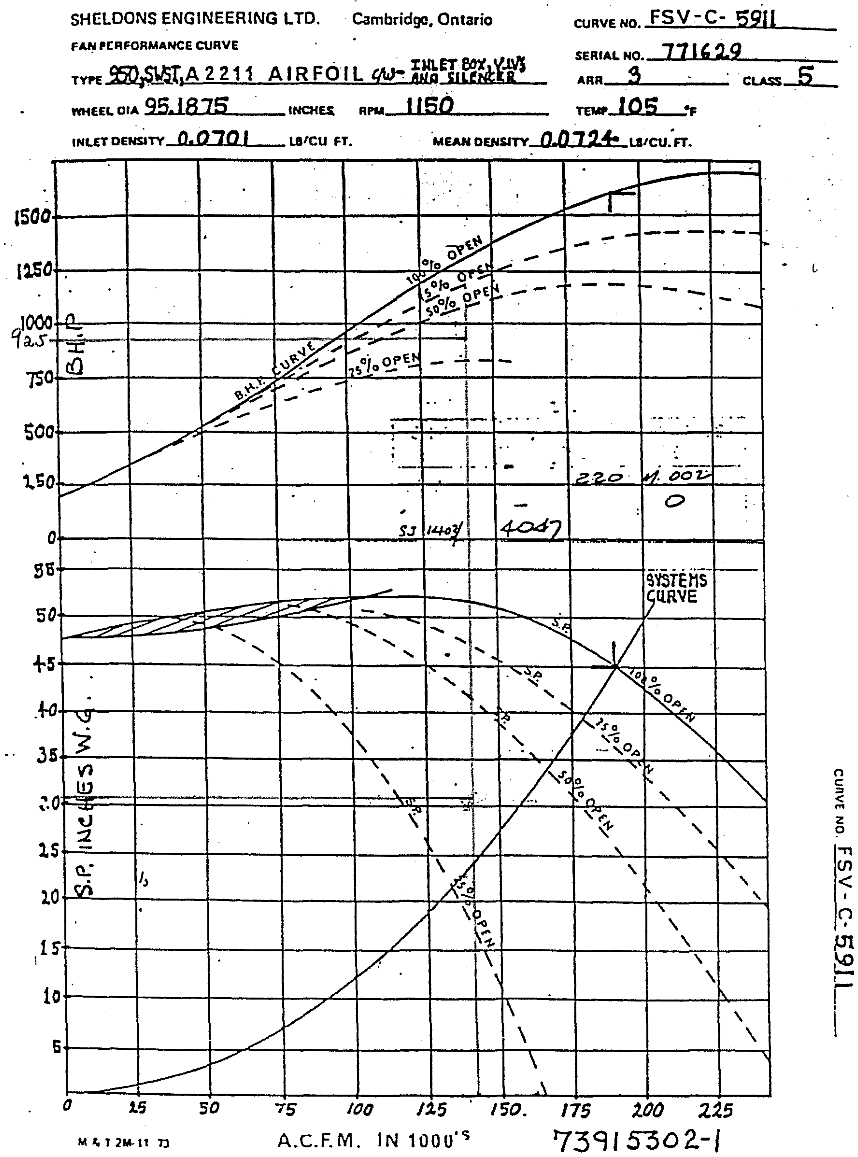
8.4 Unit #3 Heat Balance Diagram (NLH 1403-200-M001 Rev 2)



8.5 Unit #3 FD Fan Performance Curve (Sheldons Engineering)



UNIT 3 F.D. FAN WEST

LOAD 120MW
CFM = 141,623 , 16-4-2002

8.6 ARVOS Replacement Hot End Heating Surfaces Performance (Unit #3)

Performance Tabulation	LAP-HOW1019	01/17/18
Selection Designation:	HOW-1019 Present	HOW-1019 Proposed
Model Number:	2-22.5-VI	2-22.5-VI
Element Configuration:	HE: 32.0" 22LA DU ND CE: 12.0" 22/20E NF6 FW	HE: 30.0" 22LA DN7™ ND CE: 12.0" 22/20E NF6 FW
Elevation:	100	100
Flows, LBS./HR.	Design	Design
AIR ENTERING	1,111,000	1,110,500
AIR LEAVING	1,000,000	1,000,000
GAS ENTERING	1,071,000	1,071,000
GAS LEAVING	1,182,000	1,181,500
Temperatures, DEG. F.		
AIR ENTERING	128.3	128.3
AIR LEAVING	560.	560.
GAS ENTERING	734.	734.
GAS LEAVING UNCORR.	362.	362.
GAS LEAVING CORR.	342.	342.
AVE COLD END TEMP	245.	245.
Pressures, IN.WC..		
PRESSURE DROP AIR	2.1	1.85
PRESSURE DROP GAS	2.85	2.5
HOT END DIFFERENTIAL	11.0	11.0
COLD END DIFFERENTIAL	15.95	15.35
RATIO OF SPECIFIC HEATS	0.923	0.923

 Performance tabulation is for reference only.