

BEFORE THE ENVIRONMENTAL QUALITY COUNCIL  
STATE OF WYOMING

IN THE MATTER OF: )  
BASIN ELECTRICAL POWER COOPERATIVE )  
DRY FORK STATION, ) Docket No. 07-2801  
AIR PERMIT CT-4631 )

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**RESPONDENT DEPARTMENT OF ENVIRONMENTAL QUALITY'S  
MEMORANDUM IN SUPPORT OF MOTION FOR PARTIAL SUMMARY  
JUDGMENT**

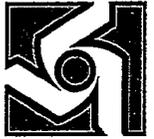
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**Schlichtemeir Affidavit**

**EXHIBIT S**

**BASIN ELECTRIC  
POWER COOPERATIVE**

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June 15, 2007

**FOR DELIVERY MONDAY, JUNE 18, 2007**

Mr. Chad Schlictemeier  
NSR Program Manager  
Department of Environmental Quality  
Air Quality Division  
122 W. 25th Street  
Cheyenne, WY 82002



**Re: Follow Up Response to DEQ Question**

Dear Mr. Schlictemeier:

In my letter of June 11, 2007, I responded to all but two of the items that you, Stewart Griner and I discussed in April and your further inquiry regarding circulating dry scrubber technology. In the June 11th letter I indicated that Basin Electric would respond to the other items as soon as possible. This letter is our response to one of the two items, as discussed below.

**1. Supercritical and Ultra-Supercritical Boilers.** Please see the attached Sargent & Lundy analysis comparing subcritical and supercritical boilers, and explaining why a supercritical boiler would not provide significant advantages at Dry Fork. Attachment 1. This analysis demonstrates that : (1) a 500 MW boiler is considered the minimum size for a supercritical boiler that would achieve a significant increase in efficiency compared to a subcritical boiler, (2) that below 500 MW the efficiency gain for a supercritical boiler falls off sharply and for Dry Fork the increased efficiency would be less than one-half of one percent, (3) that design and manufacture of a supercritical boiler less than 500 MW would be a one-of-a-kind application and therefore it would be impracticable to obtain bids for a smaller boiler, (4) that there would be a significant cost penalty for a supercritical boiler at Dry Fork, and (5) that the subcritical boiler at Dry Fork will be an advanced design that operates at higher temperatures and pressures and therefore has a lower heat rate.

We will provide a response to the one remaining item in the near future. If you have any additional questions or would like further information, please do not hesitate to contact me.

Sincerely,

A handwritten signature in cursive that reads "Jerry Menge".

Jerry Menge  
Air Quality Program Coordinator

jm/mw  
Enclosure

c: Clyde Bush  
Jim K. Miller

Date: June 11, 2007

Project: Dry Fork Unit 1 – Construction Air Permit Application

Subject: Subcritical – Supercritical Boiler Comparison

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The purpose of this memorandum is to provide additional information comparing the technical and economic feasibility of designing the proposed Dry Fork boiler as either an advanced subcritical boiler or a supercritical boiler.

### **Background**

The Dry Fork permit application, dated November 2005, described the proposed Dry Fork boiler as an indoor-type pulverized coal (PC) fired boiler designed for baseload operation. The unit will have a maximum heat input of approximately 3,801 MMBtu/hr, a maximum gross generation output of approximately 422 MW, and a net generation output of approximately 385 MW at annual average conditions. Average net generation will be slightly lower during summer maximum ambient temperature conditions due to the use of an air cooled condenser. The proposed boiler is being designed to be capable of developing main steam turbine throttle pressures and temperatures in the range of 2,520 psig and 1,050 °F, respectively, and a reheat steam temperature at the inlet of the intermediate pressure turbine of approximately 1,050 °F. The proposed main steam turbine throttle pressure is below the critical point of water, therefore, the boiler will be classified as a subcritical PC boiler.

The decision to propose a PC boiler for Dry Fork Unit 1 was based on an engineering evaluation of the available coal-based electricity generating technologies conducted by CH2MHill prior to submittal of the air construction permit application (“Coal Power Plant Technology Evaluation for Dry Fork Station,” CH2MHill, November 1, 2005). That report provided a conceptual level technology evaluation to address the advantages and limitations of PC boilers, circulating fluidized bed (CFB) boilers, and integrated gasification combined-cycle (IGCC) power generating technologies. The various generating technologies were evaluated with respect to Basin Electric’s defined needs for baseload capacity, environmental compliance, reliability and availability, commercial availability, and economic criteria. The evaluation concluded that “PC technology is capable of fulfilling Basin Electric’s need for new generation, and is recommended for the NE Wyoming Power Project [Dry Fork].”

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ATTACHMENT 1

The technology evaluation included a review of the advantages and disadvantages associated with subcritical and supercritical PC steam cycles and the associated equipment, and concluded that:

“[a] Basin Electric 250 MW PC unit would use a subcritical steam cycle design. The additional capital cost for a supercritical cycle is typically only justified by the efficiency improvement for PC units of 350 MW and larger. There is also a minimum 350 MW size limitation due to the first stage design of the steam turbine.” (Technology Evaluation, page 18).

Subsequently, Basin’s projected baseload power requirements increased from 250 MW to 385 MW (net), and the gross electrical output of the proposed boiler increased to 422 MW (gross). This report updates the comparison of subcritical and supercritical PC steam cycles at the proposed 422 MW (gross) level.

### **Subcritical and Supercritical PC Units**

Coal-fired units can be classified by their main steam turbine operating pressure and temperature. Units operating at a main steam pressures and temperatures above the critical point of water (approximately 3,208 psia and 705°F) are termed “supercritical” units. Units operating below the critical point of water are termed “subcritical” units.

In a subcritical boiler, water circulating through tubes that form the furnace wall lining absorbs heat generated in the combustion process which, in turn, generates steam by the evaporation of part of the circulated water. Saturated steam produced in the boiler must be separated from the water before it enters the superheater. Subcritical units utilize a steam drum and internal separators to separate the steam from the water circulating in the boiler tubes. The temperature of the boiler steam is increased in the superheater above the saturated temperature level. As steam enters the superheater in an essentially dry condition, further absorption of heat sensibly increases the steam temperature. The reheater receives superheated steam which has partially expanded through the turbine. The role of the reheater in the boiler is to re-superheat the steam to a desired temperature.

Modern subcritical units have a maximum turbine throttle pressure of approximately 2,520 psig. Turbines for 2,400 psig operation are usually designed for steam pressures of 2,520 psig at the turbine throttle – a condition of 5% overpressure. A boiler-drum operating pressure of between 2,750 and 2,850 psig is required to allow for pressure drop through the superheater and the main steam line. Main steam pressure and temperature, and reheat temperatures of new subcritical units (2,520 psig / 1050°F / 1,050°F) are significantly higher than pressures and temperatures achievable with older units (typically in the range of 2,400 psig, 1,000°F / 1,000°F). Increased pressures and temperatures have improved the plant heat rate of subcritical units by approximately 2%.

Supercritical boilers operate at a main steam pressure above the critical point of water (3,208 psia). When water is heated at a pressure above 3,208 psia it does not boil; therefore, it does not have a saturation temperature nor does it produce a two-phase mixture of water and steam. Instead, the water undergoes a transition in the enthalpy range between 850 and 1,050 Btu/lb. In this range its physical properties (including density, compressibility and viscosity) change continuously from those of a liquid

(water) to that of a vapor (steam), and the temperature rises steadily. Supercritical steam boilers are "once-through" boilers and do not require the use of a boiler drum to separate steam from water. In a supercritical boiler all of the boiler feedwater is turned into steam. Supercritical PC units are typically designed to develop a main steam turbine throttle pressure and temperature in the range of 3,500 to 3,600 psig and 1,050°F, and a reheat steam temperature of 1,050°F.

### **Unit Efficiency**

The efficiency of the thermodynamic process of a coal-fired unit depends upon how much of the heat energy that is fed into the cycle is converted into electrical energy. The throttle pressure and temperature of a subcritical cycle is limited by the properties of water, which limits the amount of heat energy that can be converted into working steam. The throttle pressure and temperature of a supercritical cycle is not limited by the properties of water, but by the capabilities of the materials used in the boiler, piping, and turbine. Therefore, more heat energy can be utilized in a supercritical cycle. If the energy input to the cycle remains constant, output can be increased with elevated pressures and temperatures for the water-steam cycle. Output is increased with increase steam flow (at high pressures) through the steam turbine.

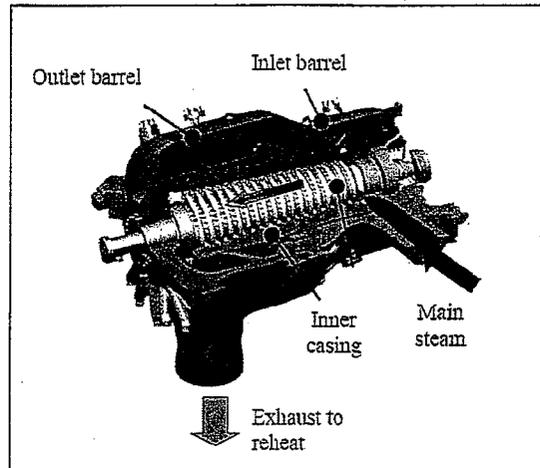
There are several turbine designs available (unique to each supplier) for use in supercritical power plants. Turbines designed for use in supercritical applications are fundamentally similar to turbine designs used in subcritical power plants. For a single reheat supercritical unit with a power output in the range of 600 – 1,000 MW, a typical turboset design would consist of three separate turbine modules operating at different pressure and temperature levels.<sup>1</sup> These three modules are the high pressure (HP) turbine, the intermediate pressure (IP) turbine, and the low pressure (LP) turbine section (which will have one, two or three sections depending on the unit size). The generator is directly coupled to the last LP turbine.

In the HP turbine steam is expanded from the main steam turbine throttle pressure to the pressure of the reheat system. Because of the high pressures associated with supercritical cycles, the inlet volumetric flow to the HP turbine is significantly lower than the inlet volumetric flow to the HP turbine on a subcritical unit. Turbine manufacturers have designed HP turbine blades specifically for use with supercritical cycles to account for this reduced volumetric flow. One HP turbine design capable of handling supercritical main steam conditions is the barrel type outer casing design, shown as a cross-section below. The high temperature components of the supercritical HP turbine, such as the inlet nozzle, rotor, and inner casing must be made with advanced types of steel (e.g., 9-12% CrMoV steel).

The steam flow is further expanded in the IP turbine section. In both subcritical and supercritical cycles there is a trend to increase the temperature of the reheat steam that enters the IP turbine section in order to raise the cycle efficiency. In the LP turbine section the steam is expanded down to the condenser pressure. There are no significant differences between the IP and LP turbine sections of a supercritical and subcritical plant.

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<sup>1</sup> Rosenkranz, J., Wichtmann, A., "Balancing Economics and Environmental Friendliness – The Challenge for Supercritical Coal-Fired Power Plants with Highest Steam Parameters in the Future," Siemens-Westinghouse, Study supported by funds provided by the German Federal State of North Rhine-Westphalia (European Regional Development Fund – ERDF), [registration number 85.65.69-T-138].



Source: Siemens-Westinghouse

### Supercritical Efficiencies and Unit Size

Efficiencies achievable with supercritical cycles are a function of the pressures and temperatures that can be developed in the boiler and the steam flow through the HP turbine. Although a few supercritical units have been built at outputs in the range of 300 – 500 MW, the vast majority of the supercritical units that have been built have been at a 500 MW gross rating or larger. At the larger sizes, volumetric steam flow through the HP turbine is large enough to accommodate larger HP first stage blades. Blade size and design is one of the most important components of overall turbine performance. For unit sizes of 500 MW or more, cycle efficiency improvements will be in the range of 1.5 – 2.0% with supercritical units. Depending on other parameters affecting plant efficiency (e.g., auxiliary power requirements), this difference in cycle efficiency results in a gross plant heat rate (Btu/MW-gross) improvement of approximately 2 to 3%. In other words, less fuel needs to be burned to generate the same electrical output.

Low inlet volumetric flow to the HP turbine (associated with supercritical pressures) is one of the main reasons supercritical units have not been typically considered for sizes less than approximately 500 MW. As size decreases below 500 MW, efficiency improvements associated with the higher inlet pressures are reduced. Some of the decrease in efficiency is due to the necessary application of very short turbine blading in the early HP stages due to the reduced volumetric flow associated with the higher inlet pressure. The shorter blades used with high pressure cycles will still be mounted on relatively high base diameters so that acceptable rotor dynamics can be achieved. This results in a high ratio of seal clearance area to nozzle flow area as compared to a higher MW rated unit with taller blades. The

increased pressure and reduced volumetric flow results in increased nozzle edge friction losses and seal losses, reducing efficiency improvements in the HP turbine.

Furthermore, since there is very little demand for supercritical equipment at sizes below approximately 500 MW, OEMs typically apply available HP turbine elements at the low end of their application range (which would be larger than necessary) to avoid one time engineering costs for new one-of-a-kind smaller units. This approach would result in the HP blades being set on a higher base shaft diameter than would be used if the elements were designed specifically for the high pressure low output condition. The resulting design would not be optimal thermodynamically, further increasing nozzle edge losses and seal losses.

Technical issues associated with high pressure, low volumetric flow, and short turbine blading in the early HP stages will significantly reduce efficiency improvement gains in the HP turbine associated with supercritical cycles. Reduced efficiency gains in the early HP stages will reduce expected cycle efficiency improvements from the 1.5 – 2.0% range on larger units (500 MW and larger) to approximately one-half that benefit as unit output is reduced down toward the 250 MW level. Discussions with the OEMs conducted as part of the technology review process were consistent on two important issues; (1) the commonly accepted break point to justify the increased costs for the efficiency gains associated with a supercritical unit is above 500 MW; and (2) in the smaller MW sizes the cycle efficiency improvements would diminish to less than one-half of the gains achievable with larger units.

### **Auxiliary Power Requirements**

Auxiliary power requirements will also affect the gross plant heat rate of the unit. Everything else being equal, fan requirements for supercritical units are slightly less than the requirements for a similarly sized subcritical unit because of the reduced combustion air and flue gas flows. However, other project unique design requirements will impact the auxiliary power and overall unit efficiency.

As noted earlier, an air cooled condenser (ACC) is being used at Dry Fork, primarily due to a lack of sufficient water to support a water cooled condensing system. Air cooled condensing systems require greater auxiliary power than water cooled condensing systems, and result in greater variations in turbine backpressure compared to water cooled condensing systems. In addition, turbine driven feedpumps, which are often applied to improve overall unit efficiency, are typically not used with an ACC because the additional steam flow from the feedpump turbine would require a larger condenser (and associated auxiliary power consumption) and would not operate as efficiently as motor driven feedpumps because turbines operate efficiently only within a relatively narrow backpressure range. Therefore, motor driven feedpumps have been selected for Dry Fork.

For large supercritical units (e.g., >500 MW) with turbine-driven boiler feed pumps, base auxiliary power requirements will be slightly less than the auxiliary power requirements of a similarly sized subcritical unit because of the efficiency of the turbine-driven feed pumps and the reduced fan requirements. However, for the Dry Fork design, which uses motor driven feedpumps, the auxiliary

power requirements for supercritical units would be in the range of 3.14 % of gross generator output compared to approximately 2.17% for subcritical units. This difference more than offsets the slight reduction in fan requirements.

Although the Dry Fork unique design considerations indicate that higher pressures associated with a supercritical unit will not significantly improve efficiency, higher steam temperatures can still be used. The Dry Fork boiler is being designed with the advanced subcritical steam cycle conditions inlet to the turbine of 2,520 psig, 1050°F / 1050°F. These increased pressures and temperatures will improve the heat rate of the plant by approximately 2% compared to subcritical conditions of 2,400 psig / 1,000°F / 1,000°F.

### ***Commercial Availability at 422 MW***

Supercritical units being planned for the U.S. are in the 500 MW gross rating or larger size. Although a few supercritical units have been built at sizes below 500 MW, turbine suppliers do not offer turbine designs for smaller supercritical steam flows. Based on discussions with OEMs conducted during the technology review process, suppliers advised that supercritical turbine designs below approximately 500 MW would be a one-of-a-kind application, and would require significant up-front design and engineering that OEMs are unable to provide in a competitive environment. Since there is very little demand for supercritical equipment at sizes below approximately 500 MW, turbine vendors would likely apply available HP turbine elements at the low end of their application range to avoid the one-time engineering costs. These HP turbine elements would be larger than necessary, further reducing potential efficiency gains with the supercritical cycle.

Likewise, given current market conditions, the boiler suppliers would not be interested in bidding a one-of-a-kind application, and concern regarding their ability to prepare a competitive offering on a small supercritical unit.

Given this feedback, it was determined to be impractical to obtain competitive bids on the two major pieces of equipment, further increasing the cost penalty for selecting a supercritical cycle.

### **Conclusions**

Although some supercritical units have been built at output levels below 500 MW, a larger majority of the supercritical units that have been built have had a gross output rating of 500 MW or more. At larger output ratings, volumetric steam flow through the HP turbine is large enough to accommodate larger first stage blades in the HP turbine, and achieve cycle improvement efficiencies in the range of 1.5 – 2.0%. The smallest application limit for supercritical boiler/turbine designs would be defined by the HP blading design (i.e., blade height), and would be in the in the range of approximately 200 to 300 MW-gross. However, below approximately 500 MW, efficiency differences between sub- and supercritical cycles become smaller because the low volumetric flows to the HP turbine. Finally, auxiliary power

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requirements for a supercritical unit at Dry Fork are higher than the auxiliary power requirements for subcritical units due to the use of motor-driven boiler feed pumps.

Sargent & Lundy (S&L) prepared heat balances and performance calculations for both subcritical and supercritical units using Dry Fork specific design criteria (e.g., fuel specifications, ambient conditions, air cooled condensing system, feedpump drivers, etc.). Heat balances and performance calculations were prepared taking into consideration expected HP turbine efficiency gains and auxiliary power requirements. The calculations indicate that net plant heat rates for either the sub- or supercritical cycle would be approximately the same. This occurs because of the minimal efficiency gains expected with small supercritical steam flows (in the range of 0.75% because of the small first stage HP turbine blades) and the impact of additional auxiliary power requirements associated with the motor driven boiler feed pumps. The table below summarizes the performance data for this case.

		Subcritical	Supercritical
Gross Turbine heat rate(Annual Average conditions)	Btu/kW -gross	7436	7269
Aux Power	%	8.41	9.30
Boiler efficiency	%	86	86
Net Plant Heat Rate (no margin included)	Btu/kw-hr -net	9440	9319
Plant efficiency	%	36.14	36.61
Difference	%	Base	0.47
Note, this difference is less than the estimated 0.75% due to the application of motor - driven feedpumps, as mentioned earlier. Based on the feedback from the turbine vendors, stated earlier, it's reasonable to estimate that the 0.47% difference shown would be even less, due to the turbine's HP-section inefficiency on smaller size units.			

Therefore, there is no technical basis, nor environmental justification, for designing the proposed Dry Fork boiler as a supercritical unit. Finally, the costs associated with designing the unit for a supercritical cycle would increase overall plant costs by approximately 2 to 4%, and most likely closer to the high value due to the reverse economy of scale effect.