

# Fairhaven Power Plant

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## 1 Introduction

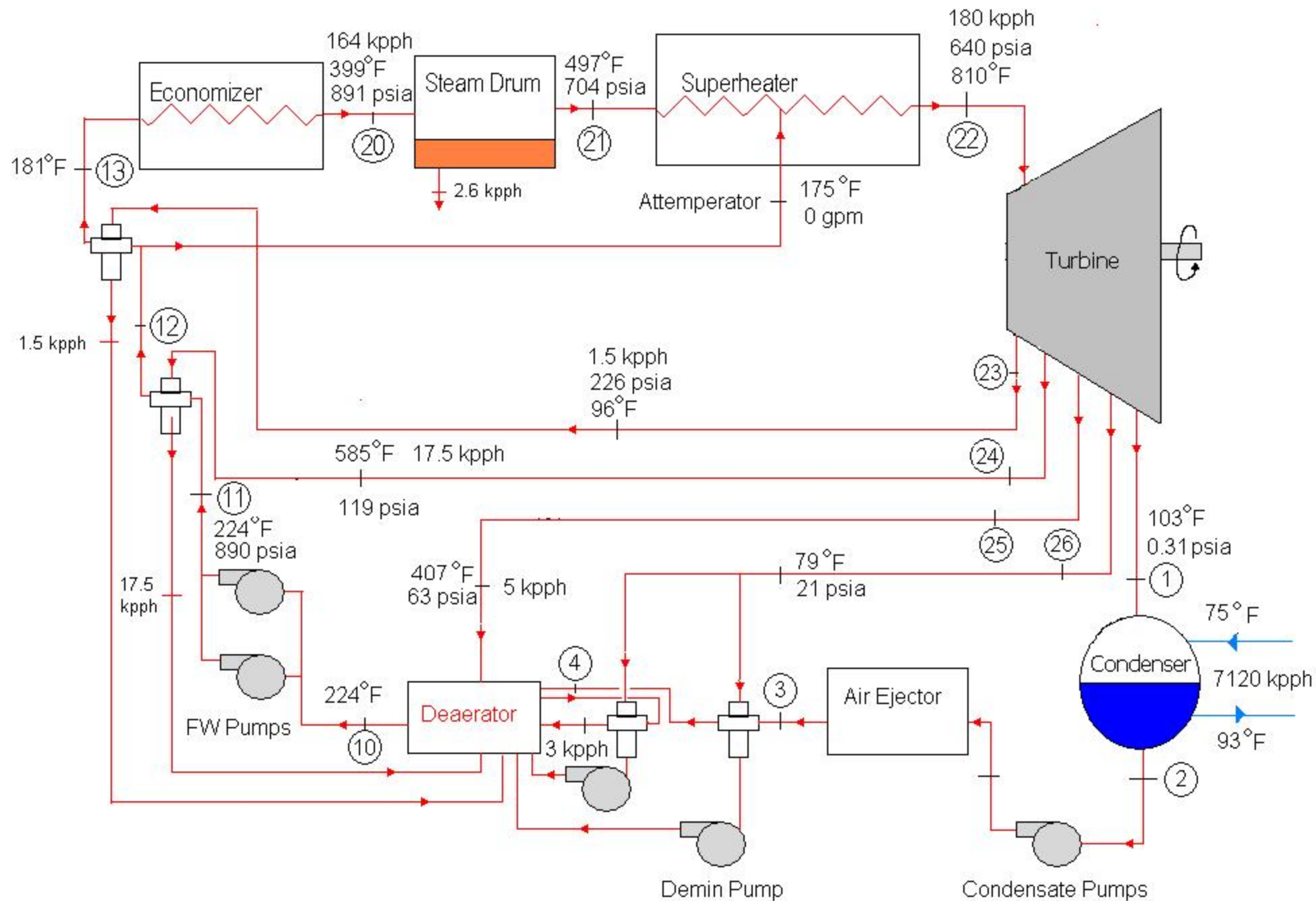
The Fairhaven power plant, owned by DG Energy Solutions, LLC, is a biomass fired steam turbine electrical generating plant located on the Samoa Peninsula, five miles from Eureka, California. Information about the plant was provided by Bob Marino, plant manager, to an engineering class from Humboldt State University (HSU) on February 17, 2006, and also to a class from HSU some years prior. Documents provided by Mr. Marino are included in the appendix.

The plant has been operating since 1986. Nameplate generating capacity is  $18MW_e$ , with  $1.8MW_e$  parasitic load on site. Fuel for the plant is biomass hog fuel in a variety of forms - sawdust, wood chips, wood bark, wood shavings, and rejected pulp. The energy content of the fuel varies significantly with the source material, and especially with the moisture content of the fuel. In the winter, with fuel moisture content over 60%, output power may be 16MW or less.

In this paper I describe the basic plant components and present a thermodynamic analysis of the Fairhaven plant. I also discuss the plant emissions of greenhouse gases and other pollutants, and suggest theoretical ways to improve efficiency and reduce emissions.

## 2 Design Details

A schematic representation of the plant suitable for thermodynamic analysis is shown in Figure 1. Values shown are from a snapshot when the plant was producing  $18MW_e$  (see



**FIGURE 1**

Schematic for Fairhaven Power Plant

appendix). Figure 1 shows state properties as presented by the plant control software. Because of unresolved questions regarding some of those values, for the thermodynamic analysis, some changes were made as explained below. The general discussion presented here uses the values reported by the plant software.

Steam pressure at the turbine inlet is 625 psig. Temperature at this point is 810°F. Steam enters the condenser as a vapor at 0.31 psia and 103°F. Mass flow rate entering the turbine is 180 thousand pounds per hour (kpph).

Water is condensed by cooling water entering the condenser at 75°F and exiting at 93°F. The flow rate of the coolant is 7120 kpph. The coolant rejects heat in the cooling towers.

Steam is extracted from the turbine at four points to provide feedwater heating. At the lowest pressure extraction point, 79°F water is provided to two closed low-pressure heaters. The deaerating open feedwater heater is provided with steam at 407°F. Two more, high-pressure, closed feedwater heaters are provided with steam from the high pressure extraction points (211 psig and 104 psig). The feedwater entering the high pressure heaters is raised in pressure by the feedwater pumps to 875 psig, and is 224°F.

The plant incorporates an economizer, in which hot flue gases heat the feedwater after it has passed through the high pressure feedwater heaters, and before it enters the steam drum. Water enters the drum at 875 psig and 399°F and exits at 689 psig and 497°F. In the superheater, the temperature is raised to 810°F and the pressure drops to 625 psig. Temperature in the superheater can be controlled with an attemperator fed by 175°F water from a high pressure heater.

Flue gases are also used to preheat incoming air. Exhaust gases pass through a dust collector and an electrostatic precipitator, reducing particulate emissions. Sand settles out, and ash is collected and used as fertilizer. Flue gases are sent back to the boiler for recombustion. Incoming air is also preheated with exhaust gases.

### 3 Unresolved Issues

At this point, several values in the data provided by Fairhaven need clarification:

- As indicated on sheets 8A and 8D in the appendix, the mass flow rate entering the

drum is 164 kpph, and leaving the superheater is 180 kpph. The only additional source of mass shown in the schematics is the attemporator, but the flow rate indicated is 0 gpm. The unresolved issue is the source of the extra 16 kpph. Mr. Marino indicated that 180 kpph is the correct value.

- The temperature at the first extraction point is 96°F, according to sheets 8E and 8G. This appears to have the effect of cooling the feedwater from 224°F to 181°F. I don't understand why this is done.
- The temperature and pressure reported for the water entering the condenser are 103°F and 0.31psia, respectively. At these values, the water would be superheated. Furthermore, the turbine work derived from these values would not be great enough to produce 18MW<sub>e</sub>. I found that a temperature of 103°F and a quality of 0.9 led to turbine work of 19.5MW, and used these values in the analysis. This would indicate a condenser pressure of 1.039psia.

## 4 Plant Efficiency

The following calculations were performed using Engineering Equation Solver (EES)®, a product of F-Chart Software, Inc, as seen in the appendix.

There are several ways to estimate plant thermodynamic efficiency. According to data provided, at 180 kpph, 194,747 kBtu/hr of heat are added to the steam. With 18MW<sub>e</sub> output, a 95%efficient generator, and 1.8MW<sub>e</sub> back work, thermal efficiency can be approximated as

$$\eta_{th,0} = \frac{(18 - 1.8)/.95MW}{194,747kBtu/hr} \quad (1)$$

After unit conversion, this indicates an efficiency of 29.9%, though the back work in this case includes all on-site electrical loads, which are not usually included in a thermodynamic efficiency analysis of a Rankine cycle power plant.

In this case, the figure of 194,747 kBtu/hr was based on fuel supply figures of 79,950 lb/hr wood waste at 3560 Btu/lb plus 954 lb/hr natural gas at 22,680 Btu/lb. These are the values provided for wood which is 60% moisture. The energy content in the fuel is therefore 306,258

kBtu/hr, which indicates an assumption of 64% boiler efficiency for the 60% moisture fuel. According to these figures, gross efficiency, as measured by the ratio of net electrical output to fuel input, is 19.1%.

Another way to estimate thermal efficiency is to estimate net work as the difference between turbine work and pump work, and to calculate

$$\eta_{th,1} = \frac{\dot{W}_{net}}{\dot{Q}_H} \quad (2)$$

Turbine work is the product of mass flow rate and the enthalpy difference across each section of the turbine. Pump work is estimated as the product of mass flow rate, the specific volume of the water, and the pressure difference across the condensate and feedwater pumps.  $\dot{Q}_H$  is the product of the mass flow rate and the enthalpy difference across the boiler,  $\dot{m}(h_{22} - h_{13})$  according to Figure 1. These calculations are shown in the appendix, and lead to an efficiency of 29.1%.

Efficiency can also be estimated by calculating the heat transfer rate in the condenser,  $\dot{Q}_L$ , which can be calculated from the temperature difference and mass flow rate of the cooling water:  $\dot{m}_c C_p (T_{out} - T_{in})$ . According to this calculation, efficiency is:

$$\eta_{th,2} = \frac{\dot{Q}_H - \dot{Q}_L}{\dot{Q}_H} = 43.5\% \quad (3)$$

If the heat transfer rate in the condenser,  $\dot{Q}_L$ , is estimated by the enthalpy change in the working fluid through the condenser rather than the change in coolant enthalpy, then  $\dot{Q}_L = \dot{m} \times (h_2 - h_1)$ . This leads to the following estimate:

$$\eta_{th,3} = \frac{\dot{Q}_H - \dot{Q}_L}{\dot{Q}_H} = 35.9\% \quad (4)$$

Some possible explanations for the discrepancies in the estimates are as follows.  $\dot{Q}_H$  as calculated, 226,900 kBtu/hr, is 14% greater than the value provided of 194,747 kBtu/hr. If the lower mass flow rate of 164 kpph (see above discussion of unresolved issues) is used,  $\dot{Q}_H$  would be 206,640 kBtu/hr, which is much closer to the number provided by the power plant. However, the plant manager has reiterated that 180,000 kpph is the correct value.

Another possible source of error is in the value of  $\dot{Q}_L$  as determined by heat transferred through the cooling fluid. This value of  $\dot{Q}_L$  may not represent all heat rejection. The value

of  $\dot{Q}_L$  as determined by enthalpy change in the working fluid is also in question as it depends on assumptions about the state of the working fluid at the condenser inlet that contradict data provided.

These explanations are not really adequate for the magnitude of the discrepancies. I believe the data are incomplete or inaccurate, but I don't know in what regard. The estimate of 29% efficiency is probably the safest estimate until more information is obtained.

#### 4.1 Second Law Efficiency

The second law efficiency of the plant is the ratio of thermal efficiency to the ideal, Carnot, efficiency. The Carnot efficiency is

$$\eta_c = 1 - \frac{T_L}{T_H} \quad (5)$$

which in this case is

$$\eta_c = 1 - \frac{103 + 460}{810 + 460} = 0.557 \quad (6)$$

Using the 29.9% figure for thermal efficiency, second law efficiency is:

$$\eta_{II} = \frac{\eta_{th}}{\eta_c} = \frac{0.299}{0.557} = 53.7\% \quad (7)$$

#### 4.2 Turbine Efficiency

Turbine efficiency can be estimated by the ratio of actual enthalpy drop across the turbine with the isentropic enthalpy drop. This calculation is shown in the appendix and indicates a turbine efficiency of 80%.

### 5 Possible Enhancements

DG Energy is investigating the potential of the Samoa Peninsula for wind and wave energy production. These sources of energy fluctuate, and it is conceivable that some storage of energy would be desirable in order to achieve a more constant output of electrical energy. This is not a realistic scenario at this time as DG Energy would no doubt prefer to sell the electricity directly rather than store some for later use. Mr. Marino reports that the

transmission lines have plenty of excess capacity for the wind and wave power. However, I chose to investigate this scenario for reasons of theoretical interest.

Storage of electricity involves substantial losses. One alternative for storage of electrical energy is the use of electrolysis of water to produce hydrogen. The hydrogen can later be used in fuel cells to produce electricity. With a fuel cell efficiency of 36%, as with phosphoric acid fuel cells (PAFC), and an electrolysis efficiency of 71% (Kato et al., 2005), this process has a theoretical efficiency of 26%.

I investigated two ways to effectively enhance the efficiency of the hydrogen system by using by-products of the storage and conversion of hydrogen to improve the efficiency of the biomass plant. The first potentially useful by-product is the waste heat generated by the fuel cells. This heat could be used to dry the wood chips, providing improved fuel efficiency. The second by-product is the oxygen produced in electrolysis. Oxygen-enriched combustion can also improve fuel efficiency in the power plant.

Another way to improve plant performance is to recover energy from the exhaust gas. This heat could be used to dry fuel or to pre-heat air or water. Most of this heat is the latent heat in the water. I calculated the amount of energy that could be recovered by condensing the water in the plant's exhaust.

## 5.1 Fuel Drying

The Fairhaven biomass plant was designed to burn wood chips that have a moisture content of around 50%. In the rainy season, which can last five months or more, the moisture content can be as high as 65%. Under these conditions the plant may only be able to produce  $16MW_e$ . Reducing the moisture content from 65% to 50% would allow a 12.5% increase in plant output if it enabled the full  $18MW_e$ . The same quantity of fuel would enter the dryer, but the fuel entering the boiler would be higher quality. Large scale conveyor and rotary dryers suitable for drying of wood chips are commercially available (Cummer and Brown, 2002). The heat required to lower the moisture content can be estimated as follows.

According to documents provided by Riley, Inc. (see appendix), at 65% moisture the plant burns 88,000 pounds of wood per hour. To eliminate 15% of that mass, all water, means eliminating 13,200 pounds of water per hour. To raise this quantity of water from

50°F to 212°F (an overestimate of the required heat) would require:

$$13,200 \frac{lbs}{hr} \times 1 \frac{Btu}{lb R} \times (212 - 50)^\circ F = 2138 \frac{kBtu}{hr} \quad (8)$$

To vaporize this water would require:

$$13,200 \frac{lbs}{hr} \times 970.35 \frac{Btu}{lb} = 12810 \frac{kBtu}{hr} \quad (9)$$

The total then is  $15,000 kBtu/hr$ , or  $4.4 MW$ . With a 65% efficient dryer,  $6.7 MW_{th}$  would be required to dry the wood.

The plant exhausts 365,380 pounds gas per hour at 350°F. The water content of the exhaust gas is 15% by mass (see appendix). The dew point is 145°F. If the exhaust gas were cooled to 150°F while drying wood, the energy available would be (treating the non-water portion as air):

$$\left(310,573 \frac{lbs \text{ air}}{hr} \times 0.240 \frac{Btu}{lb R} + 54,807 \frac{lbs \text{ steam}}{hr} \times 0.447 \frac{Btu}{lb R}\right) \times 200R = 19,800 \frac{kBtu}{hr} = 5.53 MW \quad (10)$$

This corresponds to about 82% of the necessary heat to dry the wood. Most of the energy in the flue gases is contained in the latent heat in the water. See below for a discussion of flue gas condensation.

## 5.2 Heat from Fuel Cells

Phosphoric acid fuel cells are a mature technology with significant operating experience. Systems are available commercially at the 250KW scale and demonstration systems exist at the multi-megawatt scale (Onovwiona and Ugursal, 2006). Heat as a by-product is produced at a temperature of 400°F (Onovwiona and Ugursal, 2006) at the rate of  $1.085 KW_{th}$  per  $KW_e$ . Assuming that the quoted value for heat production is with reference to 70°F, and that we only want to use the portion above 150°F, the required heat production to provide a useful  $6.7 MW_{th}$  is

$$\dot{m}C_v(400^\circ F - 70^\circ F) = X \quad (11)$$

where

$$\dot{m}C_v(400^\circ F - 150^\circ F) = 6.7 MW \quad (12)$$



This yields  $X = 8.84MW_{th}$ , which corresponds to a PAFC system size of  $8.15MW_e$ , about half the size of the biomass plant.

### 5.3 Oxygen from Electrolysis

Kato et al. (2005) investigated the use of the oxygen by-product of electrolysis as a way to improve the efficiency of the hydrogen production process. They found that while oxygen-enriched combustion in power plants could be an effective use of electrolysis-produced oxygen, the oxygen could more economically be used for medical or other purposes. Pulp mills are consumers of oxygen, and Fairhaven could probably do better economically to sell oxygen to the nearby pulp mill than to burn it. Nevertheless, here I investigate the potential for oxygen-enriched combustion to improve power plant fuel efficiency.

A PAFC system can produce electricity at 36% efficiency, based on the HHV of hydrogen (Onovwiona and Ugursal, 2006). A  $5MW_e$  PAFC then would need a quantity of hydrogen corresponding to 13.9 MW ( $HHV_{H_2}$ ). 3550KW ( $HHV_{H_2}$ ) corresponds to  $1000Nm^3/hr$  or 90 kg/hr hydrogen. For every  $1000m^3/hr$  hydrogen produced, 500  $m^3/hr$  oxygen are produced. For 13.9MW ( $HHV_{H_2}$ ), the oxygen produced is

$$500 \frac{Nm^3}{hr} \times \frac{13,900 KW}{3550 KW} \times 3.151 \frac{lb}{Nm^3} = 6170 \frac{lb O_2}{hr} \quad (13)$$

A  $15MW_e$  electrolysis/PAFC system therefore would produce 18,510  $lbs O_2/hr$ . This compares with 295,000  $lbs air/hr$  used for combustion in the power plant. 18,510 pounds of oxygen would raise the fraction of oxygen in the fuel as follows:

$$\frac{(295,000 - 18,510) \times 0.21 + 18,510 \times 1.00}{295,000} = 0.260 \quad (14)$$

According to the Department of Energy (DOE), an oxygen concentration of 25%, in combustion of hydrocarbons at 1500°F, can be expected to lead to a 7% reduction in fuel consumption (DOE, 2005). A 7% reduction in fuel consumption rate corresponds to an increase of plant efficiency from 19.1% to  $19.1/0.93$  or 20.5%. I could not find a source that specified the reduction in biomass fuel consumption that could be expected with oxygen-enriched combustion. Oxygen enrichment can raise the temperature of combustion, potentially increasing  $NO_x$  emissions. Carbon monoxide emissions can be expected to decrease with improved combustion.

It is difficult to imagine a scenario in which DG Energy would install a  $15MW_e$  fuel cell system on the Samoa Peninsula. The oxygen benefit of such a system would be an improved fuel consumption of under 10%. The heat energy in a fuel cell system of  $8MW_e$  would be sufficient to dry wood chips from 65% to 50% moisture.

## 5.4 Flue gas condensation

A great deal of energy at the Fairhaven plant goes up in steam at  $350^\circ\text{F}$ . Designers have long sought to avoid condensation of the water, as the nitrogen and sulphur compounds dissolve in the liquid water and form acids which corrode smokestack materials (Huijbregts and Leferink, 2004). In addition, metals in the effluent precipitate out and cause fouling. However, at present many plants in Denmark, Sweden, Finland, Austria, Italy, Germany, and Switzerland are equipped with flue gas condensation units (Oberberger, 1996). Che et al. (2004) describes retrofitting a natural gas fired plant with flue gas condensation in China. I could not find any descriptions of the material used, nor how they manage corrosion issues.

Flue gas condensation has the potential to reduce harmful emissions. Che et al. (2004) report:

Previous research has shown that  $\text{SO}_x, \text{NO}_x$ , dust and soot, etc., which are the constituents of the flue gas, can be partially, even totally, dissolved in the condensed water, and the pollutants emitted to the environment can be noticeably reduced.

However, I could not find enough of the previous research to substantiate this claim. In addition, significant amounts of the heavy metals reportedly can be captured in the condensate sludge (Oberberger et al., 1997). The lower temperatures lead to compounds precipitating. Once again, however, I could not find quantitative data.

The energy that could theoretically be recovered from the latent heat in the water in the flue gases is  $15.5\text{MW}$  (see appendix). In addition to the potential sensible heat recovery from the gases (see above), the potential thermal energy that could be recovered from the flue gases is  $21\text{MW}$ .

At present,  $85MW_{th}$  of fuel energy (LHV) are used to provide  $16.2 MW_e$  net, an efficiency

of 19%. If half of the 21MW from the flue gas water were recovered and used to replace fuel,  $74MW_{th}$  from the wood would be necessary, and gross plant efficiency would improve to 22%.

## 6 Plant Emissions

A primary benefit of biomass combustion is that it is, to a first approximation, CO<sub>2</sub> neutral. If the biomass burned is replaced with new growth, equal amounts of carbon are emitted and removed from the atmosphere. This assumption of greenhouse gas (GHG) neutrality bears closer examination. In addition to GHGs, the Fairhaven plant has other emissions of concern.

### 6.1 Greenhouse gases

Forest management issues are a key concern in relation to GHGs. Forests are an important sink for carbon, and it is not trivial to evaluate whether we should use forests as a source of renewable fuel or try to increase their size (Schlamadinger et al., 1997). The Fairhaven plant uses waste products from the timber and lumber industries, but even in this case there is the consideration that, from a GHG point of view, the residue may be better used to build forest soil carbon (Wihersaari, 2005b). I will not attempt to deal with these issues here. At present, California forests are adding biomass (Rogers, 2004), so I assume the fuel is sustainably produced.

Energy (assumed to be fossil-fuel derived) can be expended and GHGs emitted in each of the following phases: forest fertilization, timber transportation, chipping, chip transportation, storage, power production, and waste disposal. Reported values of energy consumed and GHGs produced in the major phases are shown in Table 1. The values are typical, as reported in the literature, normalized to the Fairhaven case of 19% efficiency in converting fuel to electricity. The kg CO<sub>2</sub> eq/MWh<sub>e</sub> for the Fairhaven plant is the stoichiometric CO<sub>2</sub> based on numbers provided by the plant for fuel consumption, air consumption, and exhaust gases produced per hour, plus a typical value for other GHGs produced in combustion as explained below. The energy produced in the combustion of the biomass is 96.3% of the

total.

Table 1: Energy consumed and GHGs produced in the wood chip combustion cycle (Wihersaari, 2005b; Raymer, 2006; Malkki and Virtanen, 2003; Wihersaari, 2005a)

	Energy (kWh/MWh <sub>e</sub> )	kg CO <sub>2</sub> eq/MWh <sub>e</sub>
Fertilizer	36.8	9.78
Harvest	47.8	12.7
Timber transport	19.1	4.3
Chipping	76.6	16.7
Chip transport	19.1	4.3
Storage	-	76-190
Combustion at plant, gross	5250	1530
Combustion (net, from CH <sub>4</sub> , N <sub>2</sub> O)		15-100
Natural gas combustion	5250	139

An additional source of GHGs at the Fairhaven plant is the natural gas that is sometimes burned in addition to the wood chip fuel. With 65% moisture fuel, 2008 lbs/hr natural gas are burned, leading to 5522 lbs CO<sub>2</sub>/hr or 139 kg CO<sub>2</sub>/MWh<sub>e</sub>. Drying the fuel to 50% moisture would eliminate the need to burn any natural gas and thus eliminate these GHG emissions.

A further analysis of the chip transportation follows. According to the USEPA, 22.2 lbs CO<sub>2</sub> are produced per gallon of diesel fuel burned. In the Fairhaven case, with the wood chips mostly coming from local sawmills, I assumed a 60 mile average trip from sawmills to the plant. With 24 truckloads a day, and 6 miles/gallon, this leads to 2420 kg CO<sub>2</sub>/day. At 18MWh<sub>e</sub>, this is 5.6 kg CO<sub>2</sub>/MWh<sub>e</sub>, which is in line with reported values.

The combustion process is not completely CO<sub>2</sub> neutral, in that the more potent greenhouse gases CH<sub>4</sub> and N<sub>2</sub>O are released. According to typical values, these account for 15-100kg CO<sub>2</sub> eq/MWh<sub>e</sub> of the GHG emissions from combustion.

CH<sub>4</sub> and N<sub>2</sub>O are also released in anaerobic decomposition of the piles of woodchips. Wihersaari (2005a) found emissions of 58-144kg CO<sub>2</sub> eq/MWh<sub>fuel</sub>, or 305-758kg CO<sub>2</sub> eq/MWh<sub>e</sub>

from a 6 month storage of wood chips. Fairhaven plant manager Bob Marino stated that they like to have 40,000 tons storage on site, which is a 44 day supply at 900 tons/day. CH<sub>4</sub> and N<sub>2</sub>O emissions from composting at Fairhaven may therefore be in the range 76-190 kg CO<sub>2</sub> eq/MWh<sub>e</sub>, which is potentially its largest source of net GHG emissions.

Reducing the size of this storage pile is easy in theory, but probably difficult in practice. A reduction in size to a one week supply would reduce CH<sub>4</sub> and N<sub>2</sub>O emissions from composting by 83% – to 13-32 kg CO<sub>2</sub> eq/MWh<sub>e</sub>. However, it is not clear that the plant could economically function on a year round basis with just a one week storage supply. In addition, the chips may simply be composting elsewhere.

Transportation emissions could be reduced by fueling the trucks with biodiesel. The largest reported theoretical yields (15,000 gals/acres/year) are from algae (Briggs, 2004). Assuming 5000 gals/acre/year, with CO<sub>2</sub> input from flue gases to assist production, 17.5 acres of algae ponds would be required to provide the 87,600 gallons per year of diesel that keep the Fairhaven plant supplied with fuel.

## 6.2 Other Emissions

Emissions from the Fairhaven plant, as reported to the California Air Resources Board (ARB) in 2004 are shown in Table 2 (ARB, 2004). For comparison, emissions from the Humboldt Bay natural gas fired turbine plant are also shown, as well as normalized values for the Fairhaven plant and typical values for wood chip plants.

Values for the Fairhaven plant per MWh<sub>e</sub> are based on the number reported by DG Energy of 120,000 MWh<sub>e</sub>/yr. The actual number in 2004 may have been substantially different. I did not find a value for the number of MWh<sub>e</sub> generated by the Humboldt Bay plant. However, this plant is the main supplier of electricity to Humboldt County, and produces between 50-100 MW<sub>e</sub>. At 65MW<sub>e</sub> year round the plant would produce 570,000 MWh/hr.

The biomass plant produces fewer SO<sub>2</sub> emissions, but substantially more VOC, NO<sub>x</sub>/MWh<sub>e</sub>, PM, and CO emissions than the clean burning natural gas. Values are also generally higher than the typical values reported from a variety of Finnish wood chip plants per KWh<sub>e</sub>.

CO emissions increase with water content of the fuel, as do other products of incomplete

Table 2: Emissions from the Fairhaven and Humboldt Bay power plants, and typical values for wood chip plant emissions are shown.

	VOC	NO <sub>x</sub>	SO <sub>2</sub>	PM <sub>2.5</sub>	PM <sub>10</sub>	CO
Fairhaven (annual tons) <sup>a</sup>	221	280.2	0.1	101.7	109.4	2425
Humboldt Bay (annual tons) <sup>b</sup>	17.7	435	8.5	18.4	18.4	64.5
Fairhaven (kg/MWh <sub>e</sub> )	1.67	2.12	0.0008	0.770	0.829	18.4
Reported biomass (kg/MWh <sub>e</sub> ) <sup>c</sup>	-	0.671	0.017	0.021	-	-

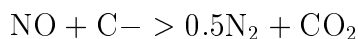
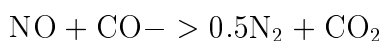
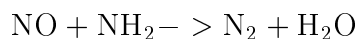
<sup>a</sup>California Air Resources Board, 2004

<sup>b</sup>California Air Resources Board, 2004

<sup>c</sup>(Malkki and Virtanen, 2003)

combustion (Staiger et al., 2005). NO<sub>x</sub> emissions, on the other hand, increase as fuel moisture decreases, with the temperature increase in the boiler. Staiger et al. (2005) found that reductions in both NO<sub>x</sub> and CO emissions were possible with changes in boiler geometry, and air-staging.

The goal is to provide a region for NO<sub>x</sub> reduction via the following reactions (Oberberger, 1996).



Excess air must be limited in well-separated primary and secondary chambers to facilitate the reactions. Both primary and secondary air must be controlled to ensure complete combustion but limit excess air. Fuels with high moisture content require high amounts of primary air and make it impossible to realize the observed 20-50% reduction in NO<sub>x</sub> emissions.

Additional emissions of concern are shown in Table 3 (ARB, 2004).

Table 3: Fairhaven power plant emissions, 2004

Pollutant	Lbs/Yr
Acetaldehyde	270
Benzene	3050.9
Cadmium	12.2
Copper	124.2
DiBenFurans(Cl)	0
Dioxins-w/	0
Formaldehyde	1810.5
Lead	62.8
Manganese	3801
Naphthalene	187
Nickel	107.4
PAHs-w/o	0.1
Zinc	1829.8

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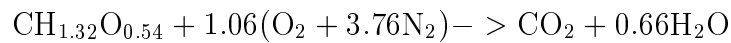
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## **7 Appendix**

1. Thermodynamic EES equations
2. EES solutions
3. Fairhaven plant screenshots
4. Data provided by boiler manufacturer Riler, Inc.
5. Flue gas condensation calculations

## 8 Flue gas condensation

The content of the 55% moisture fuel is reportedly, by mass: Water: 55%; Ash: 0.90%; C: 24.03%; O: 17.37%; N: 0.04%; H: 2.66%. This leads to an average molecular formula of  $\text{CH}_{1.32}\text{O}_{0.54}$ . Stoichiometric combustion leads to:

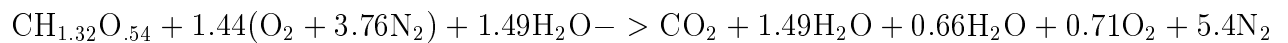


Using the reported values for fuel and air consumption leads to the following.

Table 4: Input components

	Air	$\text{N}_2$	$\text{O}_2$	$\text{CH}_{1.32}\text{O}_{0.54}$	$\text{H}_2\text{O}$
lbs/hr	295,120	226,325	68,792	32,863	40,166
kg/hr	134,145	102,875	31,269	14,938	18,257
M	28.84	28	32	22	18
kmol/hr	4651	3674	977	679	1014
kmol	6.84	5.4	1.44	1	1.49

The actual reaction therefore is:



Excess air, determined by  $1.44/1.06$  is 36%. The 2.15 moles water per mole of effluent is 23.2% as a molecular fraction, and 15% by mass. A partial pressure of 23kPa leads to a dew point of 63°C or 145°F.

{ \$DS. } "Fairhaven power plant"  
"Values are taken from a snapshot, with the plant producing 18MW"  
"Numeric subscripts refer to points in the thermodynamic cycle as shown in schematic"

"Specify temperature and pressure at points in the cycle"  
P\_22 = (625 + 14.7)[psia] "psia at turbine inlet"  
T\_22 = 810[F] "Temp (F) at turbine inlet"  
"Pressure at turbine exit/condenser inlet:"  
P\_1 = 1.039[psia] "Value provided was actually 0.31; this pressure is at saturation point"  
T\_1 = 103[F] "Temp at turbine exit"  
P\_2 = 1.039[psia] "Condenser exit - don't know what this really is"  
T\_2 = 103[F] "Temp at condenser exit"  
x\_1 = .9 "Don't know what this really is"  
x\_2 = 0 "Assume quality 0"  
"Feedwater low-pressure points:"  
T\_4 = 70[F] "Deaerator inlet"  
T\_10 = 223[F] "Deaerator exit"  
T\_11 = 224[F] "Feedwater pump exit temp"  
P\_11 = (875 + 14.7)[psia] "Feedwater pump exit pressure"  
"Feedwater high-pressure:"  
T\_13 = 181[F] "After high-pressure pumps/inlet to economizer"  
P\_13 = P\_11  
P\_20 = P\_13 "After economizer/inlet to drum"  
T\_20 = 399[F] "Inlet to steam drum"  
"Steam header:"  
T\_21 = 497[F] "After drum/inlet to boiler"  
P\_21 = (689 + 14.7)[psia] "Inlet to boiler"  
"Which brings us to the turbine inlet, which is where we started"  
"Extraction points from turbine:"  
T\_23 = 96[F] "Extraction point 1 - Hard to understand this"  
P\_23 = (211 + 14.7)[psia]  
T\_24 = 585[F] "Extraction point 2"  
P\_24 = (104 + 14.7)[psia]  
T\_25 = 407[F] "Extraction point 3"  
P\_25 = 63[psia]  
T\_26 = 79[F] "Extraction point 4"  
P\_26 = 21[psia]

"Mass flow rates:"  
m\_dot\_22 = 180000[lb\_m/hr] "Base flow rate in lb/hr"  
m\_dot\_23 = 1500[lb\_m/hr] "Extraction point 1 - to HP htr 1"  
m\_dot\_24 = 17500[lb\_m/hr] "Extraction point 2 - to HP htr 2"  
m\_dot\_25 = 5000[lb\_m/hr] "Extraction point 3 - to deaerator"  
m\_dot\_26 = 0[lb\_m/hr] "Extraction point 4 - to LP heaters - not sure what this is"  
m\_dot\_20 = 164000[lb\_m/hr] "Hard to understand"  
m\_dot\_1 = m\_dot\_22 - m\_dot\_23 - m\_dot\_24 - m\_dot\_25 - m\_dot\_26 "Mass flow into condenser"  
m\_dot\_10 = 164000[lb\_m/hr]

"Combine for convenience:"  
m\_dot\_x1 = m\_dot\_22 - m\_dot\_23 "Flow rate after first extraction"  
m\_dot\_x2 = m\_dot\_x1 - m\_dot\_24 "Flow rate after 2nd extraction"  
m\_dot\_x3 = m\_dot\_x2 - m\_dot\_25 "Flow rate after 3rd extraction"  
m\_dot\_x4 = m\_dot\_x3 - m\_dot\_26 "After 4rth extraction"

"Cooling water specs"

$m_{\text{dot\_cooling}} = 7120000[\text{lb\_m/hr}]$

$T_{\text{in\_cool}} = 75[\text{F}]$

$T_{\text{out\_cool}} = 93[\text{F}]$

$C_{\text{p\_water}} = 1.00[\text{Btu/lb\_m-R}]$

"Find some enthalpies, first the turbine:"

$h_{22} = \text{enthalpy}(\text{water}, P=P_{22}, T=T_{22})$

$h_{23} = \text{enthalpy}(\text{water}, P=P_{23}, T=T_{23})$

$h_{24} = \text{enthalpy}(\text{water}, P=P_{24}, T=T_{24})$

$h_{25} = \text{enthalpy}(\text{water}, P=P_{25}, T=T_{25})$

$h_{26} = \text{enthalpy}(\text{water}, P=P_{26}, T=T_{26})$

$h_{11} = \text{enthalpy}(\text{water}, P=P_{11}, T=T_{11})$

$h_1 = \text{enthalpy}(\text{water}, x=x_1, T=T_1)$  "Don't really know what this is - assume quality that makes the numbers work"

$h_2 = \text{enthalpy}(\text{water}, x=0, T=T_1)$  "condenser outlet"

"Find turbine work:  $m_{\text{dot}} * \Delta h$  for each section of the turbine"

"For now, don't use the first section - 96 degrees is hard to understand"

$W_{\text{dot\_turb}} = m_{\text{dot}_{22}} * (h_{22} - h_{23}) + (m_{\text{dot}_{x1}})*(h_{23} - h_{24}) + (m_{\text{dot}_{x2}})*(h_{24} - h_{25}) + (m_{\text{dot}_{x3}})*(h_{25} - h_{26}) + m_{\text{dot}_{x4}}*(h_{26} - h_1)$   
"Btu/hr"

$W_{\text{dot\_turb}} = m_{\text{dot}_{22}} * (h_{22} - h_{24}) + (m_{\text{dot}_{x2}})*(h_{24} - h_{25}) + (m_{\text{dot}_{x3}})*(h_{25} - h_{26}) + m_{\text{dot}_{x4}}*(h_{26} - h_1)$

"Convert to metric:"

$W_{\text{dot\_turb\_m}} = W_{\text{dot\_turb}} * (1/3600)[\text{hr/s}] * 1.055[\text{kJ/Btu}]$

"Find turbine efficiency by comparing actual enthalpy change with isentropic enthalpy change"

$s_{22} = \text{entropy}(\text{water}, P=P_{22}, T=T_{22})$

$h_{23\_s} = \text{enthalpy}(\text{water}, P=P_{23}, s=s_{22})$

$h_{24\_s} = \text{enthalpy}(\text{water}, P=P_{24}, s=s_{22})$

$h_{25\_s} = \text{enthalpy}(\text{water}, P=P_{25}, s=s_{22})$

$h_{26\_s} = \text{enthalpy}(\text{water}, P=P_{26}, s=s_{22})$

$h_{1\_s} = \text{enthalpy}(\text{water}, P=P_1, s=s_{22})$  "enthalpy at turbine exit for isentropic turbine"

$W_{\text{dot\_turb\_s}} = m_{\text{dot}_{22}} * (h_{22} - h_{23\_s}) + (m_{\text{dot}_{x1}})*(h_{23\_s} - h_{24\_s}) + (m_{\text{dot}_{x2}})*(h_{24\_s} - h_{25\_s}) + (m_{\text{dot}_{x3}})*(h_{25\_s} - h_{26\_s}) + m_{\text{dot}_{x4}}*(h_{26\_s} - h_{1\_s})$   
"Btu/hr"

$W_{\text{dot\_turb\_s}} = m_{\text{dot}_{22}} * (h_{22} - h_{24\_s}) + (m_{\text{dot}_{x2}})*(h_{24\_s} - h_{25\_s}) + (m_{\text{dot}_{x3}})*(h_{25\_s} - h_{26\_s}) + m_{\text{dot}_{x4}}*(h_{26\_s} - h_{1\_s})$

$\eta_{\text{turb}} = W_{\text{dot\_turb}}/W_{\text{dot\_turb\_s}}$  "turbine efficiency"

"Find heat input as enthalpy change in economizer, stream drum, superheater"

$h_{13} = \text{enthalpy}(\text{water}, P=P_{13}, T=T_{13})$

$q_H = h_{22} - h_{13}$

$Q_{\text{dot\_H}} = m_{\text{dot}_{22}} * q_H$

"Check  $h_{13}$ :"

$h_{12} = (m_{\text{dot}_{24}}*h_{24} + m_{\text{dot}_{10}}*h_{11})/m_{\text{dot}_{10}}$

$h_{13\_2} = (m_{\text{dot}_{10}}*h_{12} + m_{\text{dot}_{23}}*h_{23})/m_{\text{dot}_{10}}$

"Find pump work - lump condensate and feedwater pumps together, since we don't have intermediate pressures"

```

v_2 = volume(water, T=T_2, x=x_2)          "Assume constant specific volume
throughout"
W_dot_pump_1 = m_dot_10 * v_2 * (P_11 - P_2)
conversionFactor = 144 * (1/778.2)[Btu/ft^3-psia]    "Convert to Btu/hr"
W_dot_pump = W_dot_pump_1 * conversionFactor    "Condenser pump"

"Thermal efficiency, first way:"
W_dot_net_1 = W_dot_turb - W_dot_pump
eta_th_1 = W_dot_net_1/Q_dot_H

"Second way - find heat rejected from cooling water flow"
Q_dot_L = m_dot_cooling * C_p_water * (T_out_cool - T_in_cool)
W_dot_net_2 = Q_dot_H - Q_dot_L
eta_th_2 = W_dot_net_2/Q_dot_H

"Third way - find Q_dot_L as enthalpy change in working fluid in condenser"
Q_dot_L_2 = m_dot_1 * (h_1 - h_2)
W_dot_net_3 = Q_dot_H - Q_dot_L_2
eta_th_3 = W_dot_net_3/Q_dot_H

```