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**DISSERTATION**

**POWER GENERATION USING SUGAR CANE BAGASSE:  
A HEAT RECOVERY ANALYSIS**

**Submitted by**

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**In partial fulfillment of the requirements**

**for the Degree of Doctor of Philosophy**

**Colorado State University**

**Fort Collins, Colorado**

**Summer 2000**

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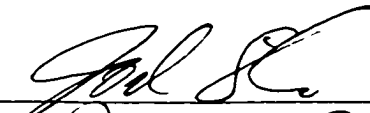
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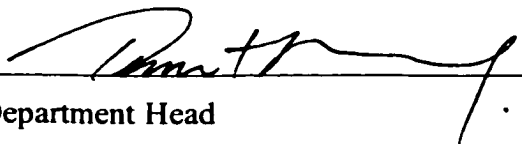
WE HEREBY RECOMMEND THAT THE DISSERTATION PREPARED UNDER OUR SUPERVISION BY JEAN SEGURO, ENTITLED "POWER GENERATION USING SUGAR CANE BAGASSE: A HEAT RECOVERY ANALYSIS" BE ACCEPTED AS FULFILLING IN PART REQUIREMENTS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY.

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## **ABSTRACT OF DISSERTATION**

### **POWER GENERATION USING SUGAR CANE BAGASSE: A HEAT RECOVERY ANALYSIS**

This dissertation describes the development of a software application for simulating co-generation plants that use sugar cane bagasse as fuel. Bagasse is the waste left at the sugar factory after the sugar cane passes through the mill and the juice is extracted from it. Traditionally it has been treated as a waste product and little interest in its efficient use has been shown. More recently, however, the sugar industry has faced intense competition from sugar substitutes and the price of sugar has declined. Consequently, the industry is facing the need to improve its performance by increasing efficiency and developing profitable by-products. An important possibility is the production of electrical power for sale. Co-generation has been practiced in the sugar industry for a long time in a very inefficient way with the main purpose of getting rid of the bagasse. The goal of this research was to develop a software tool that could be used to improve the way that bagasse is used to generate power. Special focus was given to the heat recovery components of the co-generation plant (economizer, air pre-heater and bagasse dryer) to determine if one, or a combination, of them led to a more efficient co-generation cycle.

An extensive review of the state of the art of power generation in the sugar industry was conducted and is summarized in this dissertation. Based on this review and using the fundamental laws of thermodynamics models to simulate all the components of the co-generation plant were developed. After testing the models and comparing the results with the data collected from the literature, a software application that integrated all these models was developed to simulate the complete co-generation plant. Seven different cycles, three different pressures, and sixty-eight distributions of the flue gas through the heat recovery components can be simulated. The software includes an economic analysis tool that can help the designer determine the economic feasibility of different options.

Results from running the simulation are presented that demonstrate its effectiveness in evaluating and comparing the different heat recovery components and power generation cycles. These results indicate that the economizer is the most beneficial option for heat recovery and that the use of waste heat in a bagasse dryer is the least desirable option. Quantitative comparisons of several possible cycle options with the widely-used traditional back-pressure turbine cycle are given. These indicate that a double extraction condensing cycle is best for co-generation purposes. Power generation gains between 40 and 100% are predicted for some cycles with the addition of optimum heat recovery systems.

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***Dedication:***

This dissertation is to my dad, Vittorio, († 2000), my aunt, Lucia († 1999), and my uncle, Aldo, († 1998). Thanks for all your lessons and love, I will always keep you in my heart. To my mother, Adelia, and my sister, Monica, for all your love and support. Love you all.

Jean

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## ***Chapter 1***

# ***Introduction***

### ***Power Generation Using Sugar Cane Bagasse – An Overview***

The world has been experiencing an increase in energy demand, most of it in the Third World, where investment capital and economic resources required to meet this demand are limited and very hard to obtain. Sugar cane is a very efficient crop in terms of biomass production and very common in the Third world. After the cane passes through the mills, where its juice is extracted, the material left is the sugar cane bagasse, a fibrous sub-product of the sugar manufacturing process. Sugar cane is grown all over the Third World and, since it is produced massively, the amount of bagasse generated is considerable. Through history the bagasse has been used as a source of energy, but in a very inefficient way because it has generally been considered a waste. Although it has been used to provide process heat and some power to the mill, this has usually been done very inefficiently because the main objective was to get rid of it. Bagasse has hardly been considered a valuable source of energy. The sugar industry historically has been very profitable and did not need to use bagasse energy effectively.

Today, sugar prices have decreased and in order to survive the sugar industry needs to find ways to be more profitable. One possibility is to generate electricity from bagasse in order to meet the sugar mill needs and for sale to the utility companies, which in many places cannot produce enough power to satisfy the demand. In a well designed plant about two thirds of the steam generated from bagasse is needed for sugar processing and the remaining third can be used for power generation. Currently, most sugar cane mills generate about 10 kWh per ton of cane but the potential is around 100 kWh per ton (Kinoshita (1991)). Clearly, there is a substantial energy resource in sugar cane bagasse which has only begun to be tapped.

In the research described in this dissertation the steam and power generation processes in a bagasse burning plant were modeled using data from literature, manufacturer's specifications and existing plants. To simulate these processes a computer algorithm was developed. The algorithm simulated each of the components of several variations of the Rankine cycle. Tools for an economic analysis are also included in the software for the user, who knowing the equipment costs, could compare the economic with the technical analysis and help him/her to define overall power costs. The combined analysis can be used to guide the design of co-generation plants at sugar mills to the most cost-effective option. For example, one issue of great interest is heat recovery from the flue gases of the plant (waste heat). The objective was to find the best way from a technical point of view to recover this heat and when possible compare it with an economic analysis. Possible uses for the waste heat include, pre-heating the air, pre-heating the water, and drying the bagasse. The results of this dissertation can be used to

determine how to get the lowest electricity cost from a given amount of sugar cane bagasse, given that the user can find equipment prices for the economic analysis. Thus, the simulation can be a valuable tool for industries related to sugar production and energy as their facilities and production methods are modernized.

### ***Goal of the Research***

The main goal of this research was to develop a computer program designed to analyze power generation cycles that use sugar cane bagasse as fuel. The program can be used as a tool to improve existing designs. These newer designs will include more efficient cycles and better use of the bagasse and the energy in it. The analysis was intended to be tied to the economic factors in order to determine how to generate energy at low cost. But due to lack of cooperation from equipment manufacturers sufficient economic data was not found. The economic tools are given in the software and equipment costs need to be ordered for each option to evaluate.

A major objective of the work was to determine where in the cycle it is better, from a technical/economical point of view, to recover the waste heat (heat available in the flue gases). This is a very important issue that has not been addressed yet in power generation from bagasse. The components that were considered in this recuperation analysis are the economizer (water pre-heater), the air pre-heater, and the dryer. All these heat recovery components improve the efficiency of the cycle. Many authors have

recommended their use (Madnaik and Jadhav, 1991; Keenlside, 1983; Bailliet, 1976). However, basic questions remain. These include: Is there an advantage as to where to recover the most heat? Should we use all these components at the same time? Should we recover more in a particular component, and reduce the size or eliminate another? Is there a particular configuration that gives the best performance? Should a customized design be done for each specific scenario? Are there rules of thumb that can be followed?

These critical questions were addressed in this dissertation. It is hoped that the results and the conclusions will help change the current practice of designing by similarity with existing plants and allow us to design more efficient power plants, and more efficient upgrades for existing plants.

### ***Motivation***

The sugar production industry currently faces a loss of profits with the reduction of sugar prices, mainly due to new sugar substitutes. This is a challenging situation that forces the industry to be more efficient and competitive.

One way to do this is to generate electricity. The sugar cane bagasse is a sub-product of the sugar production process. It has been considered a “waste product” and has been used in very inefficient ways. For example, burning it on the field or in very inefficient boilers has been a common practice. However, it is true that bagasse can be

used to generate electricity not only to make the sugar factory self-sufficient in electricity, but also to export energy to the utility companies. This would not only save money from the electrical power no longer needed from the utility companies, but would also represent an income from the energy that can be sold. Taking advantage of this could make the sugar industry more competitive and successful.

The benefits are not only for the sugar industry, the environment is also benefited. This biomass fuel will not generate more CO<sub>2</sub> than it consumes when it is grown. This makes it a clean fuel with clear benefits for the environment. In other words “the use of biomass to produce electricity has zero net emissions of CO<sub>2</sub>” (Thayer and Roach, 1995; Energy, 1995).

### ***State of the Industry***

Most sugar factories operate with very primitive and inefficient power generation cycles. The reason for this is that the bagasse has been considered a waste product and has been treated as such. The factories wanted to get rid of it and burned it in a very inefficient way, generating the steam necessary for the sugar manufacturing process, but in many cases not considering the possibility of power generation. Most of their energy needs were met by buying electrical energy from the local utilities and/or burning fossil fuel to obtain energy (Lobo et al, 1992; Luo and Stanmore, 1994). However, there are some good examples of efficient sugar factories. The sugar industry in Hawaii, and many

sugar factories in Mauritius, Fiji, Brazil and India have moved forward from the rest of the sugar industry (Hurtado, 1989; Kinoshita, 1991; Komen and Koster, 1994; Kinoshita, 1984). Still, they only represent a small fraction of the existing plants (Samuels, 1984). A complete review of the current state of co-generation in the sugar industry is given in Chapter 2.

Most of the world's sugar production comes from the Third World. Coincidentally this is the part of the world with the most rapidly increasing energy demands. For this reason the best technical/economical option is important. We need to find a way to generate energy at the cheapest price.

### ***Methodology***

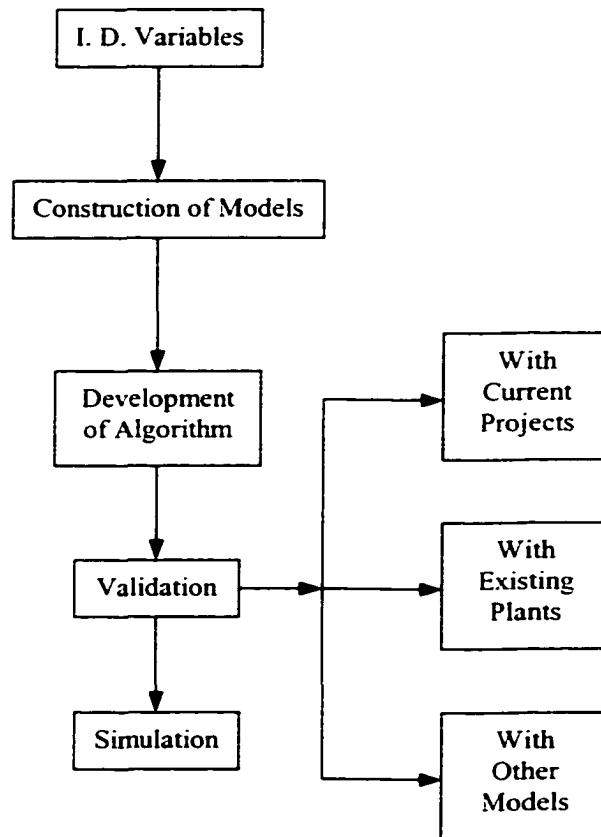
A simulating algorithm was written using the computer language Delphi to estimate the performance of different cycle configurations, and the cost of the energy produced by them.

The research followed these steps (also see Figure 1.1):

- 1- Identification of variables. An extensive literature review was done (Chapter 2 of this dissertation) to identify and define the processes and variables involved in the co-generation cycle. Each process to be evaluated has key variables that

directly impact the performance of the component (ex.: flow rates, temperatures)

- 2- Construction of models. Here the model for each component was defined in terms of the variables to be analyzed.
- 3- Development of the algorithms. From the analytical models of the components the Delphi algorithm simulating the plant was written.
- 4- Validation of the model. The sub-models (components) were validated using the results of previous work. The simulation models for each component were compared with:
  - Other models and results from other authors. These were obtained from current journals and literature.
  - Current projects. The results obtained were compared with those of current design practice. These included projects that are being, or have recently been, designed.
  - Existing plants. Comparisons were with the performance of plants that are currently in operation.
- 5- Simulations. Simulations were carried out to evaluate the different cycles and scenarios and answer fundamental questions about cycle performance and component selection.

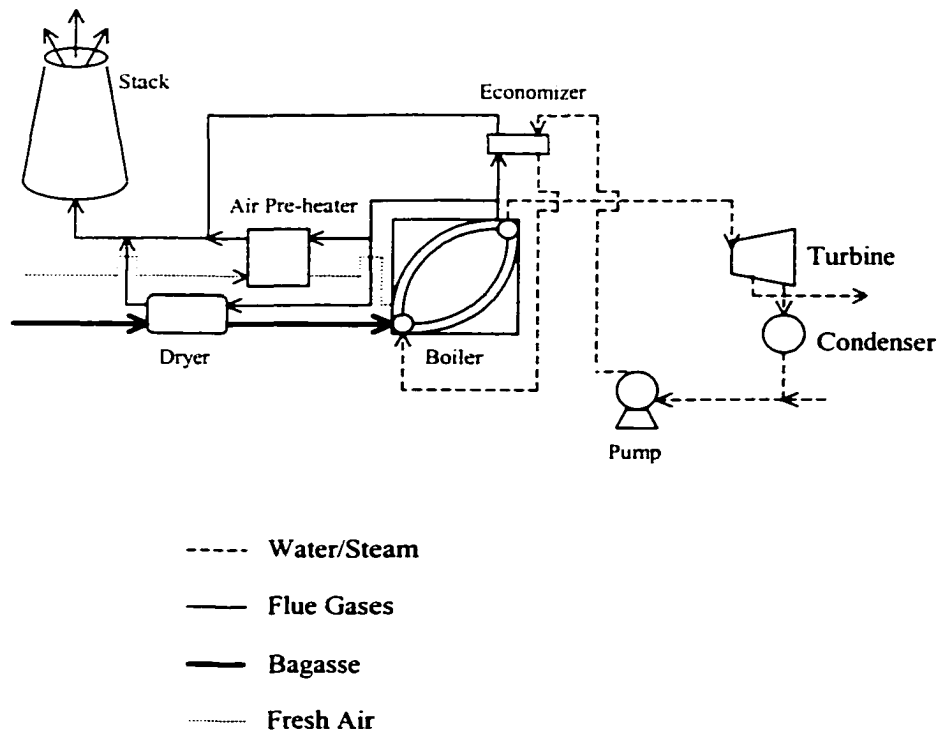


**Figure 1.1: Flow Diagram of the Project**

Each component was modeled independently and then integrated into the whole system. The components considered were the following (see Figure 1.2 for a general sketch of a typical plant):

- **Boiler.** The combustion of the bagasse and heat transfer from the combustion products to the water were modeled.

- Air pre-heater. The heat transfer between the flue gases and the fresh air was modeled.
- Water pre-heater. The heat transfer between the flue gases and the water was modeled.
- Bagasse dryer. The use of the flue gases to reduce the moisture content of the bagasse and its effect on the bagasse's heating value was calculated.
- Turbine. The extraction of power from the steam was estimated based on existing turbine designs.
- Stack. The amount of heat loss in the stack was calculated.



**Figure 1.2: General Sketch of the Co-Generation Plant**

The components were simulated with the aid of literature and manufacturer's data, not by designing each one of them from scratch. The idea was to use off the shelf equipment that is currently being manufactured. Using these components several arrangements were analyzed to evaluate the different possibilities for the Rankine cycle (reheat, regeneration). The analysis covered, first law, and second law parameters. A complete description of component models, and their validation, as well as a description of the overall plant simulation algorithm is presented in Chapter 3.

To illustrate the application of the general methodology to a specific component of the plant, model development for the bagasse combustion process will be described. The four critical steps in the methodology can be set down as follows:

- 1- Identification of variables. The variables with an effect on the combustion process are percent of excess air, chemical composition of the fuel (bagasse), temperature of the incoming air.
- 2- Construction of model. The set of necessary energy balances and chemical reactions were written down.
- 3- Development of the algorithm. The model was converted into a computer algorithm in order to analyze the effect of the variables evaluated on the process.
- 4- Validation of the model. The results were compared with the available data. Results from other authors, or existing plants, or with predictions in current projects being designed were used.

Once all the components of the system were modeled, different arrangements of the components to form overall plant simulations were evaluated. These included multiple alternatives for heat recovery and Rankine cycle configurations. The cycle simulations were compared using kWh generated for a fixed bagasse input. The results from the simulations provided direct information on the effect of the heat recovery components in the overall cycle and the cost of the energy.

The pressures and temperatures evaluated were bounded by materials limitations and widely accepted design practices. Using second law analysis the entropy generation and/or availability loss for each component in a given plant simulation was calculated in order to examine the relationship between the power generated, the entropy generation (or availability loss) and the cost per kWh generated. A complete description of the results of the simulations run, along with an interpretation of these results, is presented in Chapter 4.

In the end the plant simulation model developed in this work is intended to allow designers to evaluate a plant design in a few minutes. Making the best use of the waste heat and defining those components in which it is best recover it.

The conclusions showed that the economizer is the best option for heat recovery from the flue gas when compared against the air pre-heater and the bagasse dryer. Of all the cycles evaluated the double extraction condensing cycle showed a considerable higher performance than the other cycles evaluated and should be the cycle of choice for co-

generation purposes. Year round operation and baling/storing of bagasse also showed to be interesting options to evaluate but further research is still needed.

## ***Chapter 2***

# ***Literature Review***

## ***2.1 – Co-generation in the Sugar Industry***

### ***2.1.1 – Introduction***

A brief introduction to the presence of co-generation in the sugar industry is given in this section. Its evolution, present state and expected developments are mentioned to give the reader an idea of the importance of co-generation in this industry.

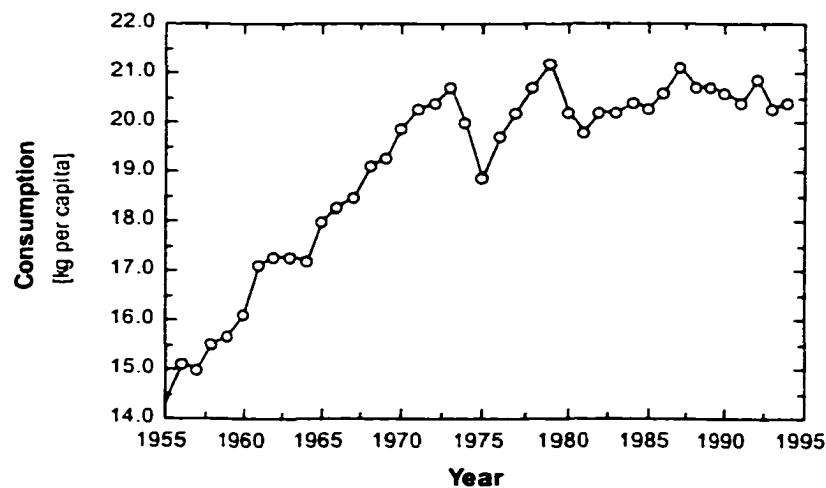
Co-generation has always been present in sugar mills. First it was a way to get rid of a waste while using it as free fuel. Later, with the change in the sugar and oil prices, it became an essential source of energy. Today, progress continues as co-generation is seen as a way to increase the profitability of the industry and a fast and easy way to bring energy to the areas surrounding the mills.

This evolution seems to suggest that this process will continue and sugar mills will become important energy providers. This should include initiating partnerships with local utility companies and government. The sugar industry has always had a big effect on local economies. Since it is labor intensive and employs a lot of the people that live in the areas surrounding the mills, this industry has always had a great interaction with the community. This impact will now become even more important as the sugar industry start playing a role as an energy provider.

### ***2.1.2 – Sugar Industry around the World:***

Through history the annual per capita consumption of sugar has slowly but steadily increased around the world. From 14.4 kg in 1955 to 20.4 kg in 1994 (see Figure 2.1). Traditionally, with some exceptions, countries have produced sugar for their own consumption. Data from 1988 and 1998 show that less than 30% of the world sugar production was destined for export (Paturau, 1989; LAWR, 1998). This number may be reduced in the future as more and more developing countries start up their own national sugar industry, with the expected consequence of the stagnation of the export market for the sugar industry. On the other hand, local consumption will increase since it will no longer be dependent on foreign exchange, which usually increases the cost to the final consumer.

Although a very labor-intensive industry in developing countries, in developed countries, where labor is much more expensive, mechanization of harvest is the tendency. As mechanization is perfected and its price reduced it is expected to expand to developing countries, displacing traditionally used hand labor.



**Figure 2.1: Sugar Consumption per capita - per year (Source: ISO)**

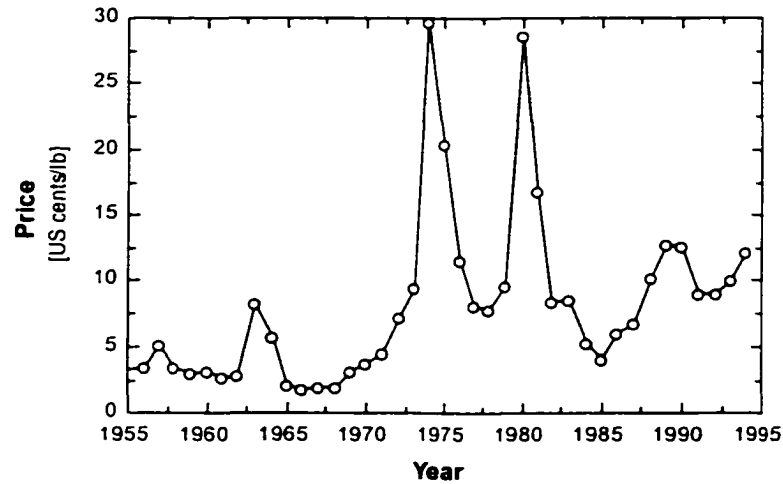
Another important change that is expected in the sugar industry is the energy requirement to transform cane into sugar. Initially the bagasse was considered a waste and was used in very inefficient ways. It was more important to get rid of it than to extract energy from it. As energy has become more expensive and energy demands have increased around the globe, the sugar industry has reduced its net energy consumption. Today's tendency is not only to be totally self sufficient in energy, but also to consider electricity as a by-product of the sugar production process. For example, the energy consumption for processing a tonne of sugar went from 1800 GJ per tonne of sugar in

1960, to less than 1500 GJ in 1985 (Paturau, 1989). In the past, market prices of sugar for human consumption were high enough to ensure that the economic penalties incurred in thermodynamically inefficient process were small (Lobo et al, 1992). Considering this, sugar factories are implementing more energy efficient processes that will allow a more efficient use of the energy, therefore using the bagasse in a more efficient way.

Another very important event that has heavily affected the sugar industry has been the development of sweeteners from sources different from sugar cane. This, together with an over-supply of sugar stock (RONCO, 1987), has heavily affected the price of sugar. Record highs in 1974 and 1980 of 30 and 29 US cents per pound, respectively, declined to 12 US cents per pound in 1994 (see Figure 2.2). By September 1998 sugar prizes had fallen 13% with respect to the beginning of the year (LAWR, 1998). To achieve greater competitiveness to its products the industry is trying to diversify its production by developing markets for sugar cane by-products, giving more value to the bagasse as a by-product and not considering it as a cheap combustible any more. Bagasse utilization may play an important role on many sugar mill survivals, as they diversify and produce bagasse by-products, such as power, ethanol, particle boards, etc.

All these factors have stimulated the appearance of electricity as a by-product of the sugar processing industry and getting the sugar industry to associate with local utilities to sell this needed electricity to be used in the local grids. This situation is more and more common every day in the third world countries where most sugar cane is produced. With low sugar prices showing no signs of improvement, and uncertainty over

access to the USA and European market, sugar growers need to squeeze more money out of their cane fields and co-generation is the way to do it (Hurtado, 1989). Wereko-Brobby and Hagen (1996) even compare the possible revenues from power sales, with the revenues from sugar sales.



**Figure 2.2: Sugar Price per year (Source: ISO)**

### ***2.1.3 – Co-generation in the Sugar Industry***

As it has been mentioned before, it has been common for the sugar industry to use bagasse. In the beginning it was used to generate the mill's energy needs for the steam necessary for mechanical power and heat. Later it was used also to generate electricity. Today sugar factories are attempting to maximize their co-generation capacity in order to sell needed electricity to the utility which serves the area surrounding the mill.

When the cost of fossil fuels was low and the price of the sugar was high, many mills bought oil to burn and generate their power needs, or just bought power from the local utility, without worrying much about the effect of their production cost. Today, with sugar prices very low due to higher competition this practice is highly undesirable, but it is still happening at many sugar mills and it is hurting their profitability. High fuel-oil prices contributed to the development and introduction of technologies and led to greater economy of energy. These included co-generation systems that allowed the combined production of heat and electricity and new manufacturing processes that use less energy.

Sugar mills can be designed to operate at much higher levels of thermal efficiency than the current average mill. This can produce considerable quantities of excess bagasse. This excess bagasse produced is in turn available as fuel for additional steam production, which can be utilized for additional electric power generation. Some good examples exist in the sugar industry that show the possibility and feasibility of transforming a sugar mill into an important electricity generator that supplies power to the local utility. Even though this is not a common practice, it seems that it has started to become important (Bradley, 1994; Kinoshita, 1991).

Cooperative agreements between sugar plants and local utilities can be especially advantageous. For example, the sugar cane crushing season is during the dry season when the hydropower resources run low. Since the crushing season is when most bagasse becomes available, the maximum power from the sugar mill could be available at this time. This might well replace the deficit of the hydroelectric resource. An example of this

is the island of Fiji, where some projects of this type between the local utility and the local sugar industry have been developed successfully (Komen and Koster, 1994).

Some sugar mills are not stopping with co-generation and in order to make their investment on equipment more profitable are considering the use of auxiliary fuels to generate electricity year round (Elliot, 1995; Energy, 1995). A possibility that has been considered is the collection and use of sugar cane tops and leaves (normally left on the field) as additional biomass fuel. However, residual fuel oil or bunker "C" has also been recommended as auxiliary fuel in boilers designed to burn biomass (RONCO, 1986). The use of auxiliary fuels is beyond from the scope of this research but it is an important potential consideration that should be remembered when evaluating a cogeneration project.

Currently the average co-generating sugar mill generates only 10~20 kWh of electricity per tonne of cane processed, generally using low-pressure steam. These mills produce little or no excess bagasse, being unable to generate more energy than that needed by them, and often having to buy part of it (Nogueira and Zarpelon, 1996; USAID, 1989; Walter and Bajay, 1993; World Bank, 1991). In contrast, efficient mills in Hawaii produce from 70 kWh to 100 kWh per tonne of cane processed, using high-pressure steam (Fulmer and Ogden, 1990; Hurtado, 1989; Kinoshita, 1991; Ogden et al, 1990; Oliver, 1986). This has become the trend in the industry, increasing the pressure and efficiency to generate more electricity. As an example, electricity generated by the sugar industry provides 25% of Cuban energy requirements (Silva and Barreda, 1994),

about 10% of the state of Hawaii energy requirements (Kinoshita, 1984; Kinoshita, 1991; Payne, 1991), about 25% of the energy required in Mauritius (Energy, 1989; Hurtado, 1989; Payne, 1991). These are places where traditional fossil fuels are very expensive, and that have increasing energy demands. In these economies the sugar industry plays an enormous role.

New technologies using gasification processes coupled with gas turbines promise to achieve from 285 kWh per tonne of cane processed (Lobo et al, 1992) to 460 kWh per tonne of cane processed (Hurtado, 1989; Vasudevan and Shivanand, 1994). These technologies are in the research and development phase and even though some demonstration projects have been carried out they are not economically feasible yet. For this reason they are not included in the present research.

The improvement of the co-generation capacity on a sugar mill can be achieved not only by installing new technologies, but also through small changes in the operation of the mill (Therdyothim et al, 1992a; Therdyothim et al, 1992b; Kinoshita, 1991). Therdyothin et al (1992b) reported that with minor modifications to a sugar mill the surplus bagasse, which could be used for power generation, would increase from 10% to 25%. Also, McGaw and Pilgrim (1991) reported that 25% of excess bagasse could be achieved with minor work. Wang et al (1991) found that a moisture reduction of 10 to 15% in the bagasse generates a bagasse surplus from 1 to 2%.

Co-generation in the sugar industry is important, not only for the sugar mill but also to the country where the mill is located. The interest in co-generation in developing countries is usually motivated by the interest in saving energy resources. Even more important, if the country has to import all or part of its energy, co-generation will help to maintain or improve the international balance of payments and to reduce the dependence on foreign sources of energy. Moreover, countries that produce oil also benefit if their own sugar factories institute co-generation because this will free more of their own oil for sale on the export market. It is also possible that some countries could satisfy all their energy needs by properly growing and using sugar cane fields (Samuels, 1984).

#### ***2.1.4 – Summary***

The sugar industry is being forced to lower its production costs. In the process of doing this, co-generation has been raised as an option. Low sugar prices, limited energy sources, higher oil prices, and higher energy demands in the host countries are some of the elements that have pushed, and will continue to push, co-generation in the sugar industry. This process has been fostered by the increased energy needs of local utilities, creating new partnerships between these industries that previously had a provider-customer relationship. Several mills have adopted co-generation cycles, but they represent a minor part of the industry. A great deal remains to be done in order to make the designs more feasible and to get co-generation to be a common practice in this industry.

Summarizing, as **Bowell (1996)** does, co-generation at a sugar mill should be pursued in order to:

- Produce additional revenue for the sugar mill
- Bring much needed electricity to local communities surrounding the mill
- Reduce the need for expensive imported fuel oil.
- Use a renewable resource of energy not adding CO<sub>2</sub> to the atmosphere

## ***2.2 – Sugar Cane, Bagasse and Cane Trash***

### ***2.2.1 – Introduction***

A brief presentation of the sugar cane as a source of energy is given. Sugar cane is a plant that can release more energy than it uses during its processing. Thus, it can be called an energy crop. Its use can replace fossil fuels that not only contaminate the environment but are also more expensive to buy. The interest in the use of the cane by-products has motivated research to create new varieties of cane that yield more of the desired by-products.

Two of the sugar cane by-products with energy value are the bagasse and the cane trash. The bagasse, which is the fibrous material left after the cane is milled, has been used for a long time in the sugar industry. Although it has many uses the most popular seems to be as a fuel. Experimental varieties have yielded up to twice more fibrous

material than regular sugar cane and one day may be used as a source of energy mainly, instead of as a source of sugar.

The other by-product treated in this section is the cane trash. Cane trash is the name that popularly has been given to the cane tops and leaves traditionally left in the field to deteriorate and go back to the soil. Recent studies have shown that the potential value of the cane trash as a fuel is as big as that of the bagasse, and some mills are studying its use.

### ***2.2.2 – Sugar Cane***

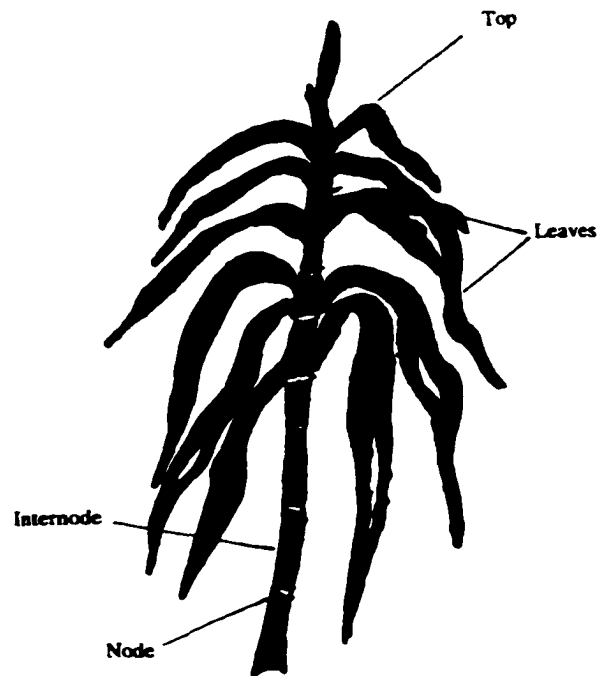
Sugar cane is the most efficient crop in terms of biomass production, that is, in converting solar energy to chemical energy. Sugar cane flourishes around the world in tropical and subtropical areas, in a belt between 30 degrees North latitude and 30 degrees South latitude. With a photosynthetic efficiency of 0.8 % under rain fed conditions or 1.1 % under irrigation, for an average 12-month crop in a tropical environment. Under these condition, an efficiently managed and irrigated crop can yield 100 tons of millable cane per hectare which correspond to a total biomass of 148 tons/ha fresh weight, primarily sugars and cellulose. This consists of around 67.7 % net millable cane, 7.4 % dried leaves, and 25 % of tops (young immature nodes) and green leaves. The total dry matter produced is about 42 tons and the total energy thus harnessed 757 GJ (Kinoshita, 1984; Ricaud and Chang, 1994), this is equivalent to 27 tons of coal or 16 tons of oil.

However, the total (direct and invested) energy required to produce the commercial products, raw sugar and molasses, is also very high, around 320 GJ/ha/yr (Kinoshita, 1984). Despite this, sugar cane has an essential qualification as an energy crop in that it produces a significant gain in total energy; that is, the energy used in growing, harvesting, transporting and processing it, is significantly less than the energy produced.

Under traditional systems of production primarily for sugar, 63 % of the dry matter biomass is exploited to produce sugar, scums, molasses and bagasse. However, there is more and more conviction that the future of agro-industries lies in a greater and optimum utilization of the crop biomass. Already in sugar cane there is a considerable use of by-products and waste products. In particular, there is increased interest in co-generation of energy and recently several countries have been investigating the use of cane trash (cane's tops and leaves) in addition to bagasse for this purpose (Ricaud and Chang, 1994).

The interest for obtaining energy out of sugar cane has motivated the development of "energy canes". These are varieties of cane that have been developed with the main objective of extracting energy out of them. For example in Puerto Rico a 12-month plant crop commercial trial produced 280 tons per hectare of green whole cane (Samuels, 1984), this is about 100 % more than traditional sugar cane. Although several energy crops have been developed, commercial utilization has not been achieved yet.

The sugar cane stalk, as harvested, is cylindrical in shape, up to 4 m long and 50 mm diameter, consisting of a hard outer rind surrounding a fibrous and pithy center containing the sucrose rich juice. The stalk is divided into vertical sections (up to 250 mm long) called joints or internodes, separated by tough fibrous areas called nodes (McGaw and Pilgrim, 1991) (see Figure 2.3).



**Figure 2.3: Sugar Cane Plant**

Traditionally, the stalk is harvested and the tops and leaves are left in the field. With the increasing interest in sources of energy the tops and leaves, popularly known as “cane trash”, have become an interesting possibility and are being used in a few sugar mills.

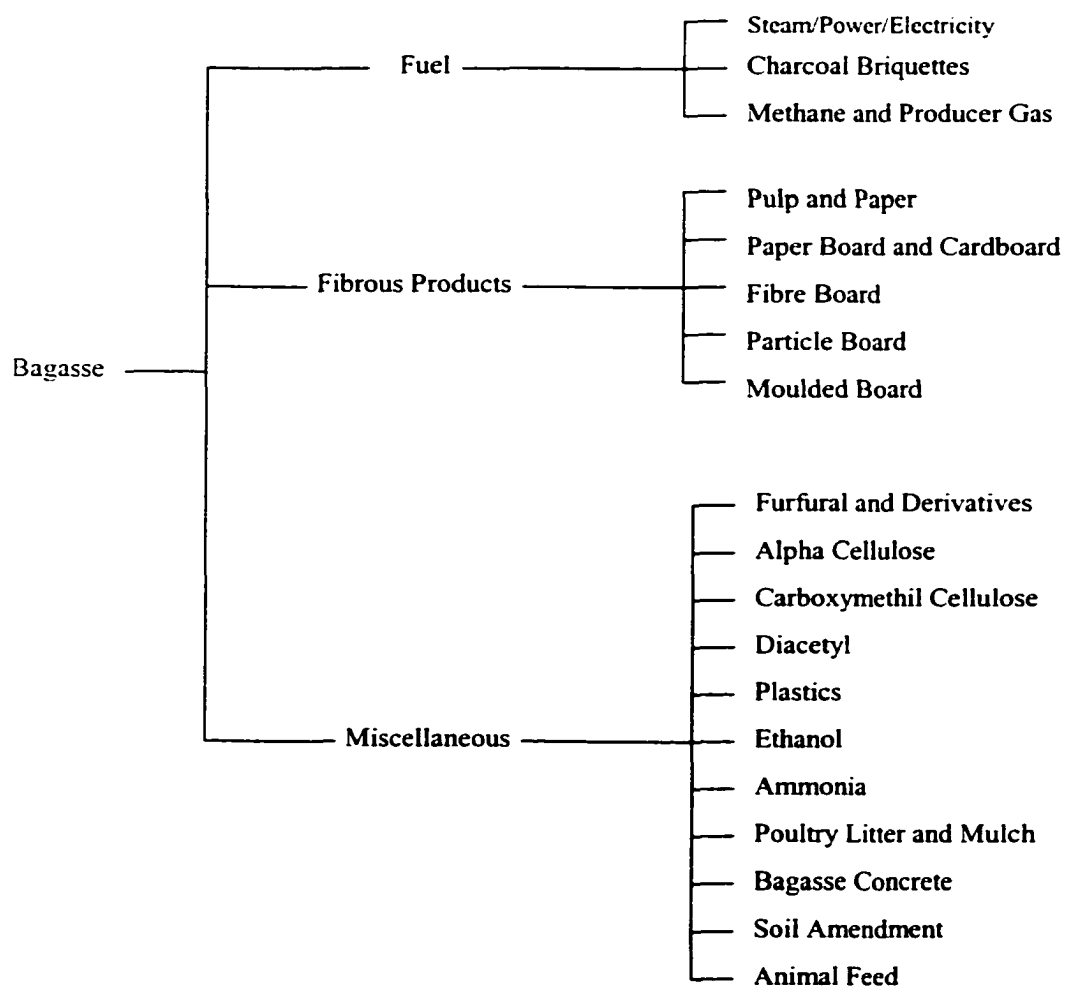
### **2.2.3 – Bagasse**

Bagasse is the fibrous material delivered from the mill after the sugar cane has been crushed and its juices extracted. The bagasse is mainly used for steam generation, but it is also used to produce paper and boards among others (see Figure 2.4).

As preparation for sucrose extraction the cane is first subjected to a knifing or shredding operation which opens up the fibrous/pithy center of the cane and chops it to short lengths. This is then passed through a series of up to six three rolls mills where extraction is effected by squeezing action supplemented with a water wash. As a consequence of these mechanical processes, particle sizes and physical composition vary significantly.

The bagasse is approximately 30 % by weight of the crushed cane (Therdyothim et al, 1992a). However, its composition varies with the variety of the cane, its maturity, the method of harvesting and the efficiency of the milling plant. It consists of woody fiber, water, soluble solids and ash in varying percentages. The moisture content ranges

from 46 % to 52 % after leaving the mill, depending on the milling technique. The sucrose and other soluble impurities are from 2 % to 5 %. Ash accounts for 1 % to 7 % in the sugar cane as delivered to the factory, depending on how clean it is. Finally fiber, the water-insoluble organic portion of the sugar cane is the remainder 36 % to 49 %. One advantage of the bagasse when compared with other fuels is its very low sulfur content (Pavgi and Shinagare, 1988). Usually this amounts to much less than 1 %.



**Figure 2.4: Bagasse Derivatives (Paturau, 1991)**

### **2.2.4 – Cane Trash**

Cane trash is how people commonly refer to the cane tops and leaves. The cane tops and leaves are of no use from the sugar-extraction point of view. Consequently this material is generally left in the field where most of it acts as a soil restituant for the field. A small part is used to feed ruminants.

Sugar cane tops and leaves (cane trash) are made from the immature or growing portion and the green leaves of the sugar cane plant. They represent about 25 % to 40 % (by weight) of the whole plant (DeepChand, 1985). Fresh cane trash has a bulk density of 240 kg/m<sup>3</sup> and is composed of 28 % dry matter of which fiber represents 78 %, fermentable sugars 11%, crude protein 5 %, and other organic and inorganic constituents 6 % (DeepChand, 1985).

Representing such a large part of the sugar plant it has a considerable amount of biomass that could be used as fuel in the sugar mill. A few days after it is cut it has a much lower moisture content, 30 % or less (Therdyothim et al, 1992a; Leal, 1995), thus becoming a good ready-to-use fuel. A problem that has been expected is that due to the fact that cane trash has higher levels of trace alkalai metals than bagasse, serious slagging and ash agglomeration can result inside the boiler with its serious consequences (Leal, 1995). But despite this, cane trash has been used in some sugar mills with positive results. Leal (1995) reports that in Brazil, the Cruangi mill operated some of it boilers fed by 100 % cane trash for more than a month without reporting any damage. Currently diverse

trials are under way to evaluate the recovery, baling and use of cane trash as fuel. In India, Thiru Arooan Sugars, Ltd. and Ugar Sugar Works, Ltd. are already recovering and baling cane trash to mix with bagasse and use as fuel (CCI, 1998; CCI, 1999).

Collecting, storing and later, during the non-milling season, burning the cane trash as a boiler fuel for power generation is very attractive. The problem with the use of cane trash is its collection from the field. When the sugar cane is harvested the cane trash is left in the field where it further dries. At the moment of collection it has to be picked up from the ground to be taken to the baler. Baling is necessary to store, handle and later use as a fuel in the boiler. Most problems come at the time of collecting the trash from the field, because rocks and soil come with it and have negative effects on the baling process and add impurities to the fuel. When the bales are going to be used they are broken up and shredded to reduce their size and are then sent to the conventional boiler fuel feeding system.

Even though no mill has reported the use of 100 % cane trash during normal operations, some of them have reported its use combined with bagasse at about 20 % to 30 % cane trash level. Several studies have been done to evaluate the use of cane trash as a supplemental, or auxiliary, biomass fuel, replacing expensive fossil fuels. These studies have shown that this practice could extend the power generating time of the mill, making the projects more feasible. USAID, the World Bank, and Winrock Foundation have been very active in promoting these studies. Common conclusions of these studies are the collection problems mentioned before, the advantage of having extra fuel and the

recommendation of collecting only from 35 % to 70 % of the available cane trash to let the rest to become part of the soil, maintaining the organic matter level. The investment to collect cane trash is from small to moderate and gives substantial returns. This makes the use of cane trash a viable alternative (Madnaik and Jadhav, 1996; Therdyothim et al, 1992b; Ronco, 1986; Winrock, 1991).

These conclusions only confirm that a careful study should be done by each sugar mill to consider the utilization of cane trash. Since each sugar mill has its own operating characteristics the use of cane trash cannot be recommended to all of them and a detailed analysis of its use is recommended instead.

When the mill is considering the production of power to sell to the utility, the use of cane trash becomes almost always desirable since auxiliary fuel would be needed to have the generating installations run longer than the harvesting season. This reduces the payback period of the investment increasing the profitability of the project.

Great experience has been achieved in Hawaii, Thailand and the Dominican Republic. Under the sponsorship of Winrock Foundation and USAID, studies have shown that the use of cane trash can be feasible. Using modified balers and shredders the cane trash has been successfully collected and used (Winrock, 1991).

One of the problems that seems to be found time after time in the collection of cane trash is the lack of adequate equipment. Equipment that has been designed for other

uses must be extensively modified to be used in cane trash collection. This increases the initial cost and the operational cost of the equipment. This is an area that is calling for further research but apparently has not been able to get it yet.

In any event, many researchers agree that cane trash should be used and represents a very important source of biomass. Not only is its energy as important as that of the bagasse, but since it is pretty similar in composition they can be burned together without any concern.

### ***2.2.5 – Summary***

Two significant developments seem to be occurring in the sugar cane field which will further the exploitation of this plant as a fuel. One is the development of new cane varieties that will yield a lot more fibre, fuel, to generate energy. The other is the use of cane trash as a fuel.

The importance of these developments rest on the fact that they will make the sugar mill more profitable due to the revenues that they will bring when the mill become a year round power producer. Also, the possibility exists that one day some mills will be using “energy cane” not to “mill sugar”, but to “mill energy”.

## ***2.3 – Bagasse Combustion***

### ***2.3.1 – Introduction***

Bagasse is the fibrous residue of the sugar cane which remains after leaving the mill where the cane is crushed and its juice extracted. Bagasse consists of water, fibers and a small quantity of soluble solids. This is a very valuable fuel that can be used to generate energy to satisfy not only the requirements of the sugar mill, but also to export energy. Understanding the combustion of bagasse is important as a first step for the energy generation process. Reviewing the variables in the combustion process and their effects is the principal objective of this section. The variables that most affect the combustion process: bagasse moisture content, percent of excess air, and the temperature of the excess air, were evaluated.

### ***2.3.2 - Basic Combustion Variables***

To properly burn any fuel these basic variables must be considered: the fuel to air ratio, the mixing of fuel and air, the furnace temperature and the furnace volume.

*Fuel to air ratio.* This ratio is of extreme importance to allow the complete combustion of the fuel. A stoichiometric analysis gives the minimum amount of air that has to be provided for a given fuel in order to permit its complete combustion. Often excess air

needs to be provided to improve the mixing of the fuel and air, and to allow complete combustion. To add more excess air than necessary is not desirable because it causes a drop in the adiabatic flame temperature. Because of this, furnace design becomes important to guarantee adequate mixing. If less air than necessary is provided, complete combustion will not be achieved and fuel will escape without being burnt, reducing the efficiency of the cycle. For a particular furnace the optimum fuel-air ratio may be the maximum value at which no combustible appear in the flue gas (Osbourne, 1981).

*Mixing of fuel and air.* As commented in the previous paragraph the proper mixing of the air and the fuel will allow complete combustion. Often a turbulent environment is produced to increase this mixing, and more air than necessary from a stoichiometric point of view may be needed. Under bad mixing conditions the fuel may occupy parts of the furnace while the air flows through another part of it. This will reduce considerably the efficiency of the cycle and will cause a waste of fuel.

*Furnace temperature.* A high furnace temperature is desirable not only to achieve high quality steam, but also to ignite the incoming fuel-air mixture. Also, the higher the flame temperature the higher the rate of heat transferred to the steam and probably the higher the boiler efficiency. The higher the temperature the higher the radiant heat transfer rates.

*Furnace volume.* The furnace volume must be appropriate to ensure that the hot gases will remain inside the furnace long enough to complete the combustion process, and to transfer the heat to the water/steam. Undersized volume furnaces will cause a waste of

energy allowing the hot gases to leave the furnace while they are at a relatively high temperature, or in the worst scenario, while combustion is still occurring.

Ensuring good combustion of the fuel usually means a high efficiency cycle. This can be done by keeping an optimum fuel-air ratio, and it usually happens at a fuel-air ratio slightly higher than the stoichiometric point for the particular fuel (Osbourne, 1981).

### ***2.3.3 - Bagasse Combustion Properties***

Bagasse, like any other fuel, has its physical and chemical characteristics. When burning bagasse, composition, particle size, moisture content, ash content, excess air and heating value must all be considered.

*Bagasse composition.* This varies with the variety of cane, its maturity, the method of harvesting and the efficiency of the milling plant (Upadhiaya, 1991; Paturau, 1989). It is very important to determine this for any design and to define the particular composition for each location. There are many specific analyses available in the literature (Upadhiaya, 1991; McBurney and McBurney, 1997; Luo and Stanmore, 1994). For the calculations required in this thesis an average based on these will be assumed (see Table 2.1).

**Table 2.1: Typical bagasse composition (% by weight-dry basis)**

C	44 %
H	6 %
O	48 %
Ash	2 %

*Particle Size.* Even though usually it is assumed that the particles are small enough to ensure good mixing and fast combustion, this variable can not be overlooked. It is important that the size distribution is compatible with the handling, processing and combustion equipment. A recommended particle size for bagasse combustion was not found in the literature reviewed. When inquiring to other researchers about particle size the information gathered suggested that bagasse fibers are shorter than 50 mm, but in general between 8 and 13 mm.

*Moisture content.* This is probably the single most important variable in bagasse combustion analysis. It varies greatly depending on the degree and quality of the cane processing, ranging from 22% to 58 % after leaving the mill (Luo et al, 1995). On average it is commonly assumed that the moisture content of the bagasse coming out of the mill is around 50 % of the total weight (Upadhiaya, 1991). The higher the moisture content, the lower the heating value of the bagasse, making it a very important variable. A high moisture content not only decreases the heating value of the bagasse, but also for a given energy requirement it requires more excess air to achieve complete combustion of

the greater amount of bagasse needed to supply that energy. This is undesirable as explained earlier both because it reduces the flame temperature and puts a higher load on the fans. The reason why the moisture content affects the heating value is that the moisture within the bagasse needs to be evaporated and the resulting vapor must be raised to the furnace temperature before the combustion of the fuel can begin (McBurney and McBurney, 1997). In order to do this energy must be taken from the combustion process. When the moisture content is higher than 58~65%, burnout decreases considerably, the furnace exit temperature begins to fall alarmingly and the combustion is likely to be unstable (Luo et al, 1995; McBurney and McBurney, 1997).

*Ash content.* In a similar fashion, the presence of ash can lower the heating value of the bagasse. Analysis has shown that the ash in the bagasse is mainly silica with some iron, aluminum, phosphate and potassium carbonate (Upadhiaya, 1991). However, the amount of ash in bagasse is so small that it does not represent a concern as a reduction in the heating value of the bagasse (Luo et al, 1995). Moreover, the ash has no abrasive component. Still, the ash content can be a concern as a refractory that may be formed when melted on the heat transfer surfaces at temperatures over 1600 °C. To avoid this a furnace gas temperature limited to about 1000 to 1500 °C is required (Upadhiaya, 1991). However, since the ash chemistry, like the bagasse composition, is particular to the location of the plant, an analysis of the ash and its melting point is recommended to avoid the problem of furnace slagging and super-heater fouling.

*Excess air.* As explained earlier, excess air is usually necessary to ensure complete combustion and control temperature. However, this has a negative effect since reducing the flame temperature reduces the heat extraction rates, and the increase on excess air increases the power requirement of the boiler auxiliary equipment (fans). Vasudevan and Shivanand (1994) estimated that for every 10 % increase in the excess air there is a 1 % increase in the heat loss. They also estimated that most sugar mills boilers operate at about 50~60 % excess air.

*Heating value.* During the combustion of any fuel the chemical energy in the material is released in the form of heat. The quantity of heat produced per unit weight of fuel is called the heating or calorific value of the fuel.

Like the composition of the bagasse this parameter varies considerable depending on the bagasse composition and its processing. There is sufficient data in literature to develop a good average value. Many authors have suggested equations as function of moisture and ash content (Hugot, 1986; Shewale, 1986; Upadhiaya, 1991; Harel and Baguant, 1992; etc).

In the combustion calculations necessary for this dissertation an approximate, average formula suggested by Hugot (1986) was used:

$$L.H.V. = (4,250 - 12 * i - 48.5 * w) * 4.1868 \quad [kJ/kg] \quad (2.1)$$

Where:  $L.H.V.$  = Low Heating Value of bagasse [kJ/kg]

$w$  = Moisture Content of bagasse [%]

$i$  = Non-fibre impurities content of bagasse (2~3) [%]

The ash content of the bagasse is generally small and practically does not affect the thermal efficiency of the boiler plant (Luo et al, 1995; Upadhiaya, 1991). Consequently it will not be included in the combustion modeling used in this dissertation (see Chapter 3),

## ***2.4 – Equipment Used for Co-generation***

### ***2.4.1 – Introduction***

In this section an overview of the furnaces/boilers, economizers, air pre-heaters and turbines most commonly used by the sugar industry is presented. The boilers that have been, and are being, used by the sugar industry are a “scale down” of the boilers that are used by utility companies for power generation, particularly those designed to use coal as fuel. The bibliographical review showed that the efficiency of the boilers that are currently being manufactured to be used by this industry is around 60 % no matter what type of furnace/boiler is used. This conclusion was confirmed by data from 3 different manufacturers.

Any power generation cycle can be improved by reducing the energy wasted in the cycle. In the case of steam cycles the biggest loss occur in the flue gas leaving the

stack. In order to reduce these losses and improve the efficiency of the cycle, heat recovery equipment is placed in the cycle to extract energy from the flue gas that otherwise would be wasted. The heat recovery equipment commonly used in the sugar industry are the following: Economizer or water pre-heater, air pre-heater or air heater, and bagasse dryer.

The economizer recovers energy from the combustion products by using it to increase the temperature of the water prior to its entrance to the boiler, and reducing the temperature of the combustion products before they leave the system. Economizers are widely used in power generation cycles and have proven to be an effective way to recover energy. The air pre-heater, like the economizer, recovers energy from the combustion products by using it to increase the temperature of the combustion air prior its entrance to the furnace, and reducing the temperature of the combustion products before they leave the system. Some reports suggest that air pre-heaters are chosen instead of economizers when the bagasse has a high moisture content, over 45 %. Hotter air helps the combustion of the bagasse by helping the evaporation of the water in it before it burns. The effectiveness of bagasse dryers is debated due to the lack of a consensus on the positive results from its use. The dryer is designed to reduce the moisture content of the bagasse thus enhancing its calorific value. In other words it increases the available energy present in the fuel.

The first reports on bagasse drying date back to 1911 when E. W. Keer published the results of his experience with bagasse dryers (Boizan Justiz, 1986; Arrascaeta and

Fridman, 1984). Keer conducted a series of tests during the 1910 harvest season at Palo-Alto, Louisiana. In these tests the moisture level was reduced from 54.47 % to 44.45 % and the vapor generation rate was increased from 1.63 to 2.53 kg of vapor per kg of bagasse. Since then several models of bagasse dryers have been developed. In addition to drying the bagasse by thermal means a few technologies have risen that dry the bagasse by chemical means. The interest in bagasse drying is not only to increase its value as a fuel, but also to allow its storage for later utilization. Being a seasonal product, bagasse is produced but not necessarily used at the same time. Storing bagasse as it comes from the mill presents two problems; a high moisture content and a low density, requiring high volumes to store it. To solve these problems the bagasse must be dried and compacted.

A very important condition that has to be watched is the temperature of the flue gas. Since some Sulfur is present in the bagasse, sulfur trioxide is present in the flue gas and if the gas temperature is lower than its dew point temperature, sulfuric acid will form and corrode the heat recovery units.

Steam turbines have been present in the sugar industry for a long time. Initially they were used as the prime movers of the mill and other components of the factory. Later they were used to move the generators and provide electricity to the mill. Today many mills are run with electric motors, but many still use steam turbines as their prime movers. The steam turbines coupled to generators have traditionally been of the back-pressure type. This is understandable since there was no need of extracting more power and bagasse was considered a waste. Now many sugar mills are looking at electricity and

bagasse as commodities that can bring revenues to the factory. This is the reason why condensing steam turbines are, occasionally, being use in the sugar industry. Although, condensing turbines are gaining popularity in the sugar industry, back-pressure turbines still are the most common.

#### ***2.4.2 – Boilers – State of the Art***

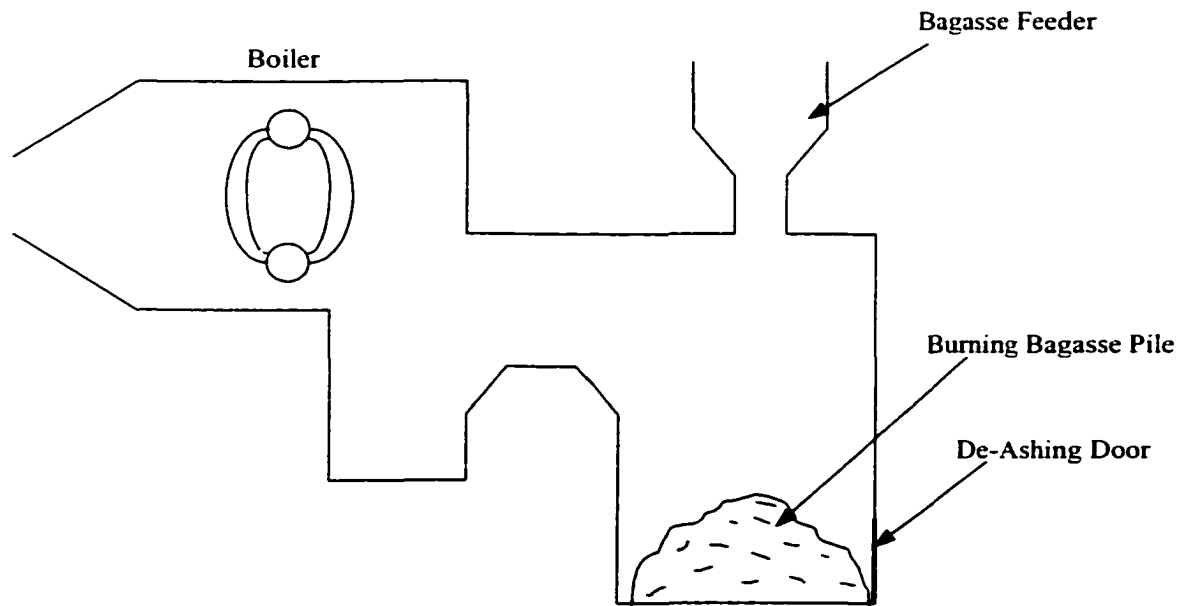
Boilers used for co-generation in the sugar industry are similar to those used by utility companies for power generation, particularly the designs used for coal burning. Minor modifications and a completely different fuel feeding system are the changes that need to be make to a coal burning boiler to get it to burn bagasse. While the steam capacity of bagasse fired boilers may be large by process and biomass standards, it is comparatively small when compared with the boilers used by utility companies for power generation. Bagasse boilers are generally 10 % the capacity of utility boilers (Dixon, 1991).

The temperature and pressure, parameters that basically define a boiler, are constrained by material and economic limitations. Present alloy steels permit operation temperatures of up to 565 °C (Payne, 1991), temperatures above 565 °C imply the use of very expensive alloy steels (Winrock, 1991). This could be used as the limit for co-generation boilers. To avoid extremely expensive materials, a common recommendation is a design temperature between 400 °C and 440 °C (Winrock, 1991).

In general, sugar factories need steam at two different pressures. High or medium pressure steam is needed to operate the mill turbines (if they are not operated by electrical motors), the electric power generation turbines and other components of the factory. Low pressure steam or process steam is necessary for the heat requirements of the sugar processing. The pressure at which the factory needs the steam has an influence on deciding what the boiler operation pressure must be. A small increase in the design pressure design may cause a big increase in the cost of the system, so a careful analysis must be done to ensure an optimum selection. Winrock International studies have led to the conclusion that, from an economic standpoint, steam pressures of 4,000 kPa to 6,000 kPa are desirable for sugar factory steam generators installed for co-generation purposes (Winrock, 1991) . Payne (1991) suggests that the optimum conditions for a co-generation boiler would be 6,000 kPa and 460 °C. Diwakan and Kumra (1994) and Antony et al (1994) also report that the pressure and temperature should be around these values. However, most systems operate at pressures and temperatures lower than these values. Luo and Stanmore (1994) did some simulations of a bagasse fired system that operates at 1,750 kPa and 280 °C. The most common furnaces used in the boilers in the sugar industry are described next.

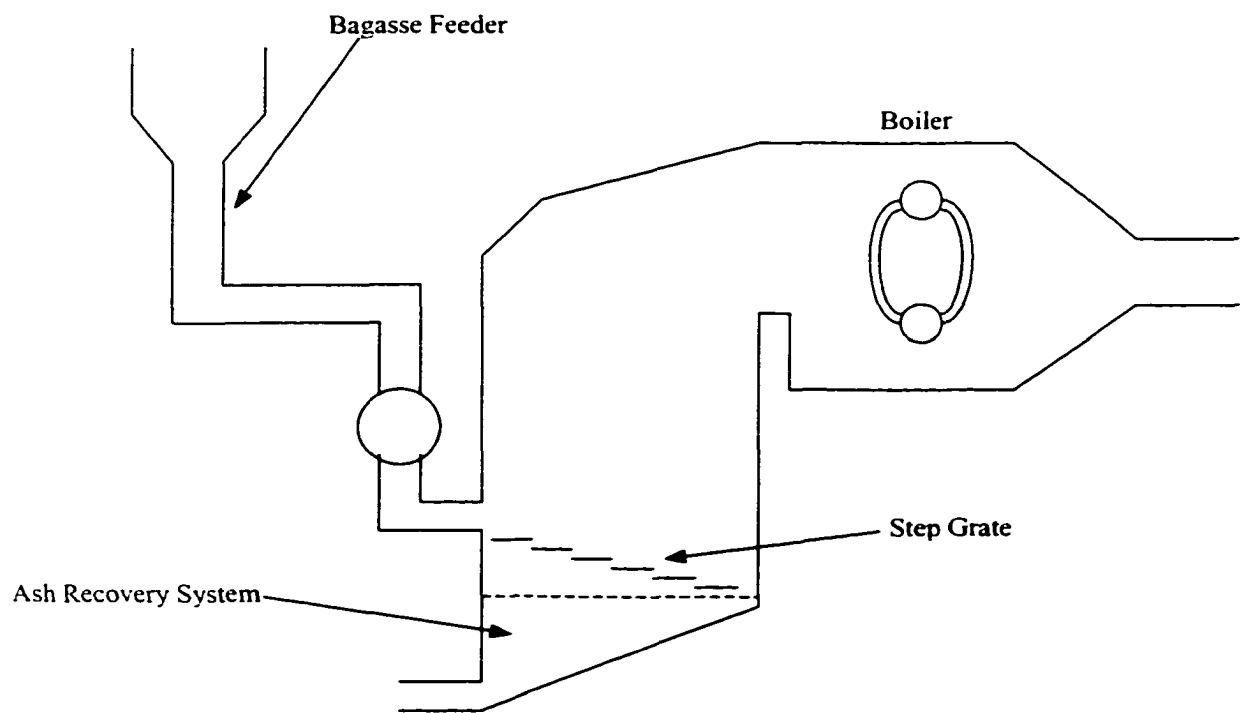
*2.4.2.1 – Cell and Horseshoe furnaces* These are pile burning furnaces and probably the most rudimentary and inefficient furnaces used by the sugar industry. The bagasse is usually fed by gravity into a refractory lined furnace where it forms a pile (see Figure 2.5). Combustion takes place mainly on the outer surface of the pile. The combustion air

is provided through ports around the periphery of the furnace. The advantages of these systems are its simplicity due to the lack of moving parts and the fact that a considerable amount of bagasse is stored inside the furnace, allowing the interruption of supply from the mill without altering the steam generation. The disadvantages are high maintenance cost (usually due to frequent changes of the refractory), poor control, and, considerable air leakage, which results in lower efficiency. Many existing boilers that have had cell or horseshoe furnaces are being, or have been, modified to suspension fire systems (Watkins, 1993), such as spreader stokers, in order to improve their efficiencies.



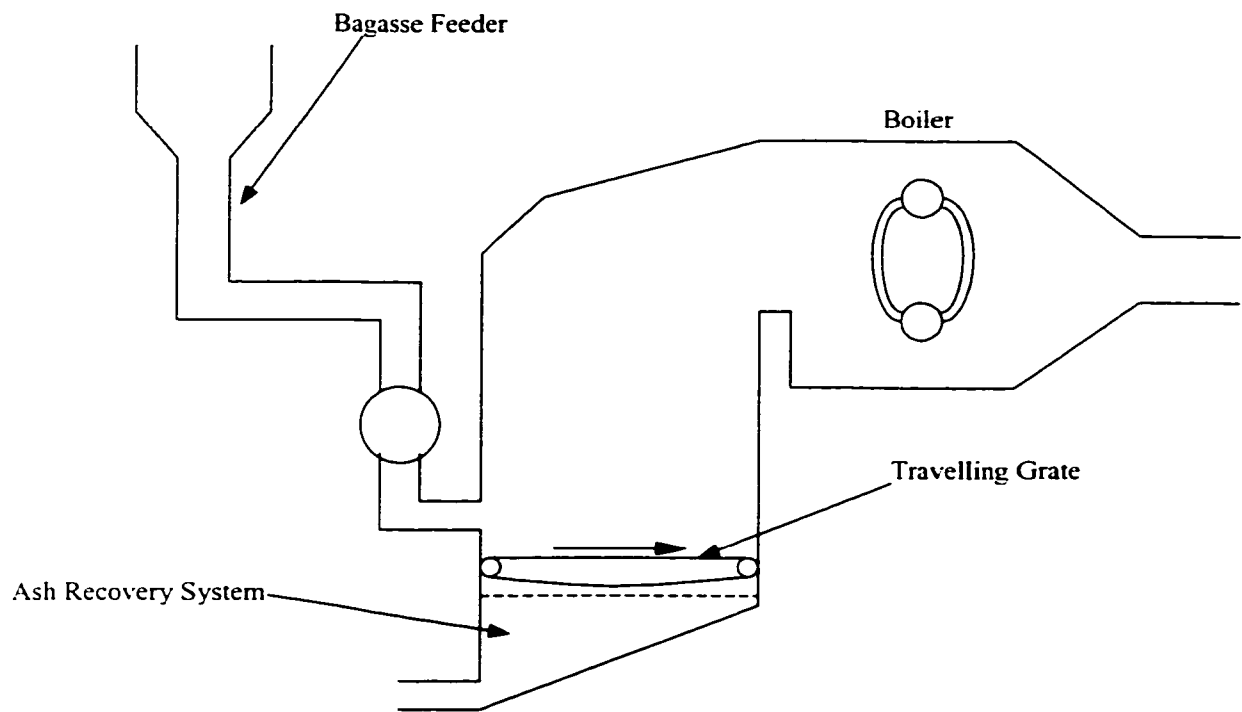
**Figure 2.5: Cell Type Furnace**

2.3.2.2 – Step Grate This kind of furnace has a series of steps (like a stair) where the bagasse burns (see Figure 2.6). The biomass is fed at the highest step and while it burns it also flows to the next step, until the remaining ash reaches the bottom where it is collected by the ash recovery system. The movement of the bagasse is achieved by different designs. Some have fixed inclined steps, where as mass builds up it falls to the next step. Others have moving steps that are moved by a rotating crankshaft, this movement of the step makes the biomass falls to the next step. A boiler like this was commissioned in Jamaica in 1987 and Scheutzlich and Bourzutschky (1988) have reported very satisfying performance.



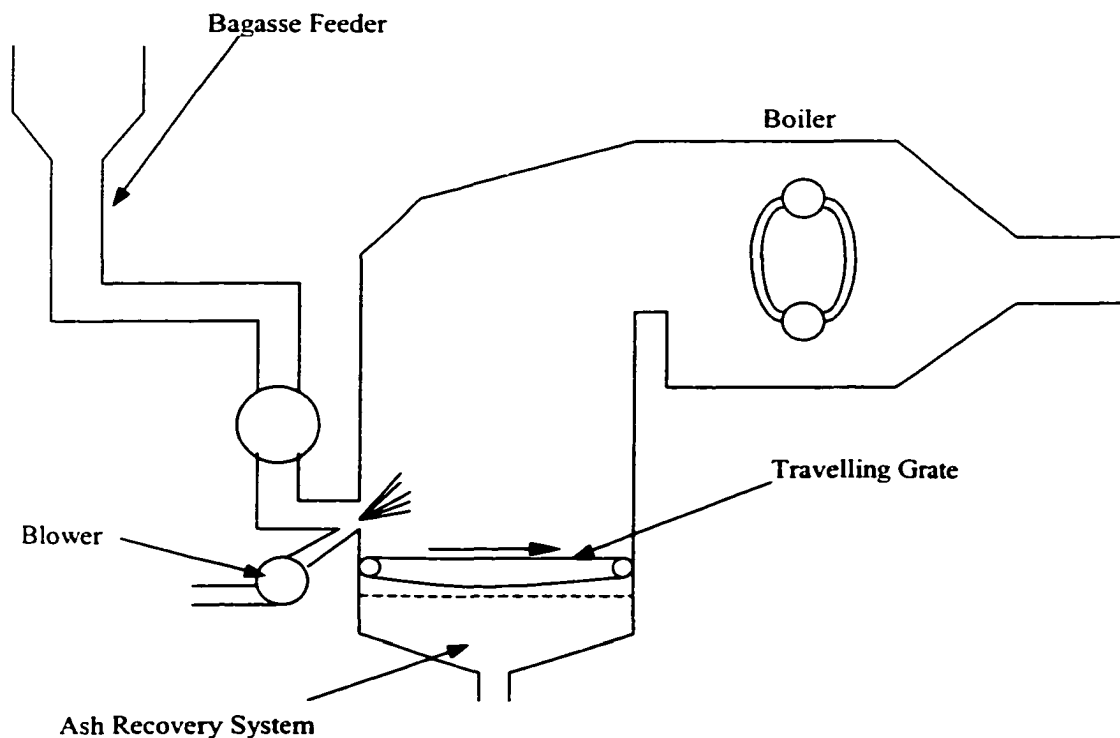
**Figure 2.6: Step Grate**

2.4.2.3 – *Travelling Grate* Similar to the step grate, but the grate is moving away from the fuel feeder and the fuel burns as it “travels”. At the end the ash is dumped to the ash collection system (see Figure 2.7).



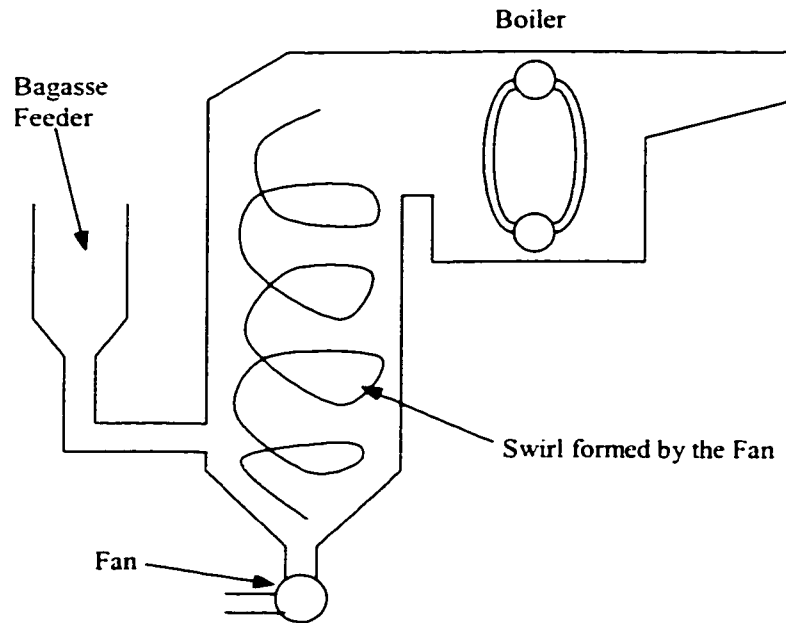
**Figure 2.7: Travelling Grate Furnace**

2.4.2.4 – *Spreader Stoker* Payne (1991) refers to this as the most efficient design to use with bagasse. In this boiler the bagasse is spread into the furnace by means of an air blast. With a large furnace volume, the moisture in the bagasse evaporates while it is in suspension and the combustion occurs for a large fraction of the bagasse while it is falling (see Figure 2.8). The larger particles fall on the grate and burn before the ashes are discharged. One important advantage of this kind of furnace is that it can be used with bagasse that has high moisture content. As an alternative to the pneumatic system, the fuel can also be fed by a mechanical device located above the grate's surface. In either case the purpose is to feed the fuel evenly over the grate area to release an equal amount of energy from each square foot of active air-admitting surface. The air for combustion can be supplied evenly through the grate to provide the oxygen required for combustion.



**Figure 2.8: Spreader Stoker Furnace**

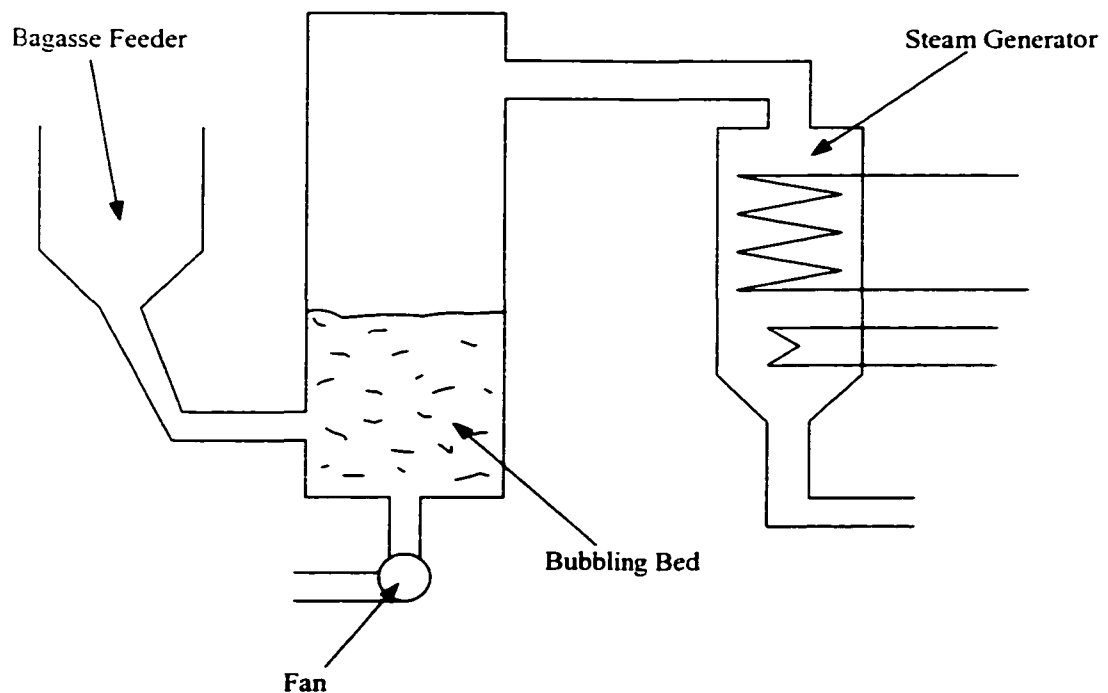
*2.4.2.5 – Swirl Burner Combustion System* This is a derivative of the spreader stoker. A newer design of combustion system, it requires relatively dry bagasse (up to 25% moisture content). The swirl burner generates a stable high intensity flame (the combustor) which burns most of the incoming bagasse within the confines of the flame. Only larger bagasse particles leave the immediate flame zone and move to free furnace space. The most important advantage of this system is that no substantial bagasse is accumulated on the grate. This eliminates the need for auxiliary air usually provided to the grate in order to guaranty the combustion of the bagasse on it. Also, the dryer bagasse does not need to be pulverized to ensure the combustion. The flame stability using this combustor and dry bagasse has been reported to be excellent (Dixon, 1994). On the other hand, the downside of the system is that a dryer is needed to reduce the moisture of the bagasse that normally comes out of the mill with a moisture content around 50 %. Some swirl designs do not limit the combustion to a combustor, but using pneumatic distributors generate a “swirl” current of air and bagasse inside a large part of the furnace where the bagasse burns (see Figure 2.9). Tests done using these systems have shown good combustion and no accumulation of burning bagasse on the grate as well (Silva and Beaton, 1994). The advantage of this swirl system is that it does not require the bagasse be dried, so a dryer is not necessary. The difference between the two swirl systems is that the first limits the combustion to a concentrated zone or combustor, while the second creates a big swirl inside the furnace, creating a bigger combustion zone.



**Figure 2.9: Vertical Swirl Furnace**

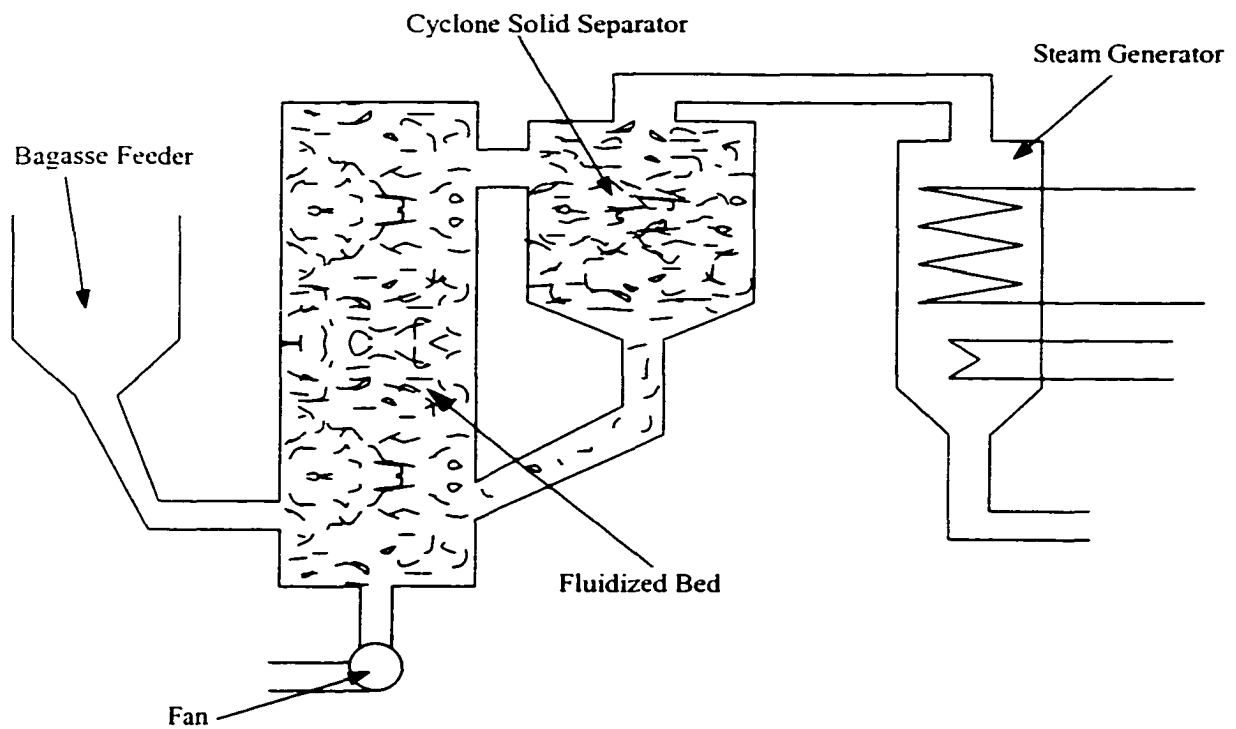
**2.4.2.6 – Bubbling Fluidized Bed Combustion** This technology uses a solid inert substrate (sand/crushed refractory) that is fluidized (suspended in a stream of air flowing upwards) by injecting air on it (see Figure 2.10). This air is also used for combustion and passes through the bed particles at a sufficiently high rate that the pressure drop across the particles in the bed balances their weight, so they become freely suspended, or fluidized. Under this conditions the bed (volume of solids) behaves as a “fluid”. The solids forming the bed stay up to a level inside the combustor and should not leave it. This system produces intense mixing and uniform temperature distribution, avoiding hot spots. The bed gives the combustor a large thermal inertia facilitating control and reduces the sensitivity of the system to changes in the fuel conditions, a situation that is common in

some sugar mills (Chong et al, 1992). In theory this system should accept fuel of different sizes, so crushing it to a particular size is not as important as it is for other systems, such as the spreader stoker. The larger particles will just remain in the bed until they burn. Another advantage of the fluidized bed combustion technology is the lower combustion temperature, around 800 °C, instead of the 1,200 °C for spreader stoker. This causes a lower generation of NO<sub>x</sub> (McBurney and McBurney, 1997).



**Figure 2.10: Bubbling Fluidized Bed Combustor**

2.4.2.7 – Circulating Fluidized Bed Combustion This is similar to the Bubbling Fluidized Bed Combustion system, with the difference that it operates at higher air velocities and the entire combustor is filled with solids. The solids leaving the combustor are collected by solids collection equipment and send back into the combustor (see Figure 2.11).



**Figure 2.11: Circulating Fluidized Bed Combustor**

*2.4.2.8 – Boiler Efficiencies* Different authors have reported similar efficiency values (defined as the amount of energy gained by the steam in the furnace over the amount of bagasse energy that is being fed to the boiler) for boilers using these different furnaces. Scheutzlich and Bourzutschky (1992) report 68 % for step grate, Silva and Beaton (1994) report 65 % to 72 % for furnaces using swirl systems. Magasiner (1996), Babcock (1995) and DeepChand (1985), report efficiencies ranging from 68 % to 76 % for travelling grates. Antony et al (1994) report 70 % for spread stoker and 72 % for fluidized bed combustion. Gehlawat (1994) reports 72 % for a general boiler using bagasse. These efficiencies are based on the global calorific value (GCV), or higher heating value (HHV), of bagasse with 50 % moisture content and using a economizer as heat recovery equipment. Turns (1996) defines the higher heating value as “the heat of combustion calculated assuming that all of the water in the products has condensed to liquid”. In some cases the efficiencies have been adapted to these conditions from the ones reported by the different authors in order to make the comparison. In terms of net calorific value (NCV), or lower heating value (LHV), these efficiencies are around 85 %. The lower heating value can be defined as the heat of combustion calculated assuming that none of the water in the products has condensed to liquid. Watkins (1993) reports efficiencies from 62 % to 65 % for cell and horseshoe boilers. Old boilers that are still in operation have efficiencies as low as 30 %. McIntyre et al (1985) say that typical boiler efficiencies in the sugar cane industry are from 45 % to 70 %.

### **2.4.3 – Economizers**

An economizer is simply a heat exchanger in which heat is transferred from the combustion products (the flue gas) to the water entering the boiler, resulting on the increase of the water temperature and the decrease of the flue gas temperature. Mullen (1984) dates the use of economizers in the sugar industry back to the year 1860, when the bagasse was burned inefficiently and large amounts of energy were wasted through the flue gas. Mullen (1984) also advises that the temperature of the feed-water entering the economizer should never be below 66 °C to avoid sweating of the tubes at the cold end. Whenever this condition exists, external corrosion will result from acid attack caused by the flue gas coming in contact with the wet metallic surface. Internally corrosion of the economizer surfaces is often due to dissolved oxygen in the feed-water. Mullen (1984) advises the deaeration of the water and heating it to above 100 °C before it is sent to the economizer in order to avoid the internal corrosion. Something that should also be avoided is the generation of steam inside the economizer. Except when it is designed for this, it can causes problem in the water flow and solids to precipitate in the water that will accumulate on the tube walls and hurt the heat transfer in the tubes. To avoid this, the designer has to make sure that the temperature of the water exiting the economizer is below the saturation temperature of the water at the operation pressure. Diwakan and Kumra (1994) suggest that the temperature of the water leaving the economizer should be 25 to 30 °C below the saturation temperature in order to ensure adequate circulation.

Li and Priddy (1985) estimate that the efficiency of the boiler will rise 1 % for every 10 °F increase of the water produced by an economizer. Analyzing the performance of several economizers designed to be installed in bagasse-fired boilers (Babcock, 1995; Zum, 1996; John Thompson, 1997) shows that these economizers will raise the efficiency of the boiler from 5 to 14 %.

#### ***2.4.4 – Air Pre-Heaters***

Similar to the economizers, air pre-heaters are designed to recover heat that otherwise could be wasted in the flue gas. Hugot (1986) says that generally the maximum increase of the efficiency is of the order of 12%. Air pre-heaters can use different sources of heat. Flue gas, steam or a separate furnace, are the common sources of heat in an air pre-heater. Traditionally, in the sugar industry the air pre-heater has used the flue gas as source of heat. The air is on one side of the tube and the flue gas in on the other. As in economizers corrosion must be avoided in the air pre-heater. To do this the temperature must be kept above the dew point of the flue gas.

Hot air assists the burning of high moisture content biomass. It improves the burnability of the fuel and the combustion stability, thereby saving fuel and maintaining a stable operation of the boiler. For this reason for very moist bagasse the air pre-heater has been preferred over the economizer.

### ***2.4.5 – Bagasse Drying***

Drying bagasse is done to reduce its moisture content prior to combustion. Before any fuel is burned the moisture it contains must be evaporated. All steam generators designed to burn biomass fuels (bagasse, wood bark, etc) can operate with moisture levels up to 50 to 55 % without problem. But when the moisture level is over 55 % difficulties occur. The most common is flame instability, when the fuel cannot produce enough heat on the grate to sustain combustion. While in the past many bagasse dryers have been used to reduce the moisture levels to where the fuel can be burned without instability, now more bagasse dryers are being used to further reduce the moisture content of the bagasse in order to achieve higher efficiencies and generate more energy.

In addition to increasing the flame stability and the efficiency of the steam generator, bagasse drying can have other advantages. Since less air is required to ensure good turbulence of the lighter bagasse, the excess-air requirements are reduced. This saves fan power and furnace volume. In addition it reduces the amount of fuel required to generate a given amount of power, saving fuel and reducing the size of the fuel handling system. Since reducing the moisture level increases the calorific value of the fuel, drying the bagasse allows us to get the same amount of energy with less dry bagasse than if we use wet bagasse. As Upadhiaya (1991) said when referring to bagasse as a fuel, “from the total available heat must be deducted that portion of heat which is used to evaporate the bagasse moisture and to superheat the resulting steam to the temperature of

the flue gas". If that moisture is removed using waste heat, the available heat that Upadhiaya mentions can be used for steam generation.

Based on cost analysis, English (1987) said that from some studies the optimum moisture levels should be between 35 and 40 %. This is in part to avoid having the temperature inside the boiler get too high. High temperature inside the boiler can lead to problems such as: ash melting and slag formation, failure of pipes due to overheating, increase of the radiation losses, etc. McGaw and Pilgrim (1991) also agree that a normal reduction of the moisture content of the bagasse should be to 35/40 %. For other applications, like pelleting and briquetting, bagasse moisture contents below 20 % are required (Edwards, 1981).

McGaw and Pilgrim (1991) developed a model to simulate bagasse drying using flue gas. Their model assumed a drying rate correlation (Equation 2.2) with an empirical parameter ( $k$ ) determined by the gas temperature, velocity and humidity, and particle size.

$$MR = \exp(-k * \theta) \quad (2.2)$$

Where:

$$k = 0.0019 * e^{(0.0073 * T)} + 0.0292 * e^{(-0.89 * S)} + 0.00078 * V + 0.00057 * H^{-0.57} - 0.00088 * V * e^{(-0.89 * S)} - 0.314 \quad (2.3-a)$$

$MR$  = Moisture Ratio (Final over initial moisture content)

$T$  = Gas Temperature [ $^{\circ}C$ ]

$V$  = Gas Velocity [m/s]

$S$  = Particle size, cross section diameter [m]

$H$  = Gas Humidity [kg water/kg of dry air]

$\Theta$  = Time [sec]

In a previous article, Pilgrim (1987), presented a similar model but the equation (3.41-a) had different coefficients:

$$k = 0.0019 * e^{(0.0073 * T)} + 0.0292 * e^{(-0.89 * S)} + 0.0058 * V + 0.00057 * H^{-0.57} - 0.0065 * V * e^{(-0.89 * S)} - 0.03135 \quad (2.3 - b)$$

Similar models have been suggested by Duggal et al (1988) which include only a temperature dependence.

Several attempts to duplicate the results reported by McGaw and Pilgrim, Pilgrim, and Duggal et al, were unsuccessful except for the equations presented by Pilgrim (1987) (2.2, 2.3-b). The results obtained with the other equations were not only inconsistent with results reported in those articles, but also impossible in the context of moisture ratios. Equations 2.3-a and 2.3-b are very similar and it can be presumed that the factor of ten differences in the 3<sup>rd</sup>, 5<sup>th</sup>, and last coefficients may be a printing error. When the third and fifth coefficients were multiplied by 10 (0.00078 becomes 0.0078, 0.00088 becomes

0.0088) and the last divided by 10 (0.314 became 0.0314) the results from both equations (modified 2.3-a and 2.3-b) were very similar, with differences below 2 %.

A few more complicated models have been developed. Stanmore and Arici (1997) extended a model developed by McIntosh (1976) for brown coal. Their results show good agreement with their experimental data and with other more complex models. The authors adapted the McIntosh model to bagasse fibres which were regarded as cylinders of large aspect ratio, so that drying could be considered as a one dimensional effect. They assumed a cylinder with a moist core surrounded by a dry outer shell. This model is described by the following equations:

$$\frac{MC}{MC_0} = e^{(-n*t)} \quad (2.4)$$

$$n = \frac{2*h*(T_g - T_w)}{\left( \lambda * \rho_w * R * \left[ \frac{MC_0}{(1 + MC_0)} \right] \right)} \quad (2.5)$$

To estimate  $h$ , the heat transfer coefficient, Stanmore and Arici suggest the use of the equation proposed by Whitaker (1972):

$$Nu = (0.4 * Re^{0.5} + 0.06 * Re^{0.667}) * Pr^{0.4} \quad (2.6)$$

$$Nu = \frac{2*h*R}{k_0} \quad (2.7)$$

$$Re = \frac{2 * R * V * \rho_g}{\mu} \quad (2.8)$$

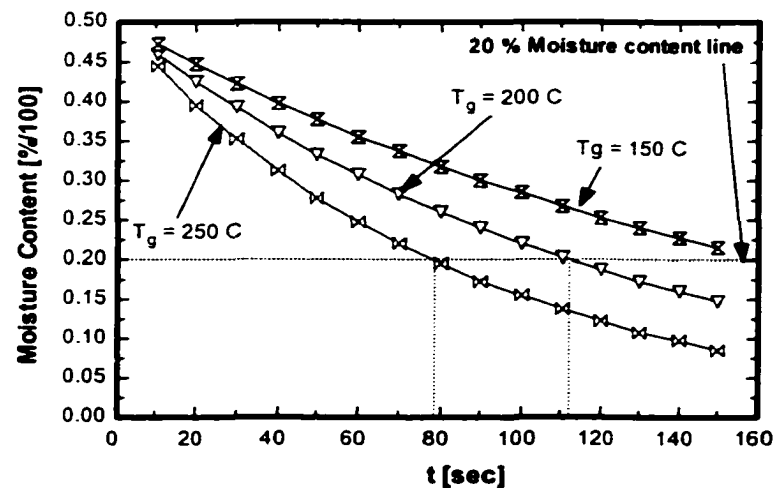
$$Pr = \frac{Cp_g * \mu}{k_g} \quad (2.9)$$

- Where:
- $h$  = External convective heat transfer coefficient
  - $MC$  = Moisture content (mass of water/mass of dry solid)
  - $k$  = Thermal conductivity
  - $R$  = Cylinder Radius
  - $Re$  = Reynolds Number
  - $t$  = time
  - $V$  = Gas velocity
  - $\rho$  = density
  - $\lambda$  = Latent heat of vaporization of water
- Subscripts:
- $0$  = Initial
  - $g$  = bulk gas
  - $w$  = at the surface of the moist core
  - $s$  = dry solid

Aralde et al (1993) went a little bit further and developed an expression to determine the heat transfer coefficient of bagasse particles in relation to the variables involved in the drying process for a pneumatic conveyor bagasse dryer operating with hot gas. Nebra and Macedo (1989) also did an analysis for a pneumatic dryer, specifying the heat transfer equations and other details of it. These models are specific to a particular

type of dryer, and their quite detailed analyses are not included in the dryer models used in this dissertation.

Another important effect of the temperature of the flue gas is on bagasse drying time. Just to illustrate this effect some calculations were done using equation 2.3-b with the corrected coefficients. Figure 2.12 shows these results for three different gas temperatures, 250, 200 and 150 °C. The bagasse particle size used was 3 mm, the gas velocity 3.9 m/s, and the moisture content of the gas 0.016 kgWater/kgAir. Please notice that the temperature of the gas used in these calculations is an estimated average between the flue gas temperature entering and leaving the dryer.



**Figure 2.12: Final moisture content vs. drying time at different gas temperature**

### 2.4.5.1 – Energy Requirement for Drying Bagasse

Ranaiah and Chikhalikan (1986) suggest the calculation of this quantity by using the latent heat of vaporization of water, 540 kcal/kg. However, they did not consider the additional energy necessary to elevate the temperature of the water to 100 °C from its ambient temperature. Table 2.2 shows the amount of energy required to do this depending on the moisture level and assuming an initial temperature of 30 °C. Since the heat is going to be obtained from the flue gas (waste heat), its use will really be a gain in energy, not a loss. Other energy requirements have to be included and are dependent on the dryer used (motors, fans, etc). These parasitic energy requirements should not be underestimated. Kinoshita (1991) suggested that they are about 12 kWh per tonne of bagasse dried, the fan being the highest load of all. Paturau (1989) estimated that the energy requirement of the rotary dryer is around 23 kWh per tonne of water evaporated. His estimate for the pneumatic dryer was 35 kWh per tonne of water evaporated. Viqar (1988) reported that the electricity consumption of the bagasse dryer is about 7.5 kWh per tonne of bagasse (Therdyothin et al, 1992). Correia Maranhao (1986) reports a consumption of less than 4.4 kW per tonne of bagasse dried for some pneumatic dryers.

**Table 2.2: Energy requirement to reduce moisture in bagasse**

<b><i>W</i></b> <b><i>[%/100]:</i></b>	<b><i>m<sub>water</sub></i></b> <b><i>[kg]:</i></b>	<b><i>Deltam<sub>water</sub></i></b> <b><i>[kg]:</i></b>	<b><i>Energy</i></b> <b><i>Requirement [kJ]:</i></b>
0.50	0.50	-----	-----
0.45	0.41	0.09	232.17
0.40	0.33	0.17	425.77
0.35	0.27	0.23	589.99
0.30	0.21	0.29	729.71
0.25	0.17	0.33	851.29
0.20	0.13	0.38	957.85

Table 2.2 shows the energy that is required to remove a particular amount of water, *Deltamwater*, to get it to a moisture level *W*, beginning with a bagasse sample of 1 kg at 50 % moisture content and 30 °C. This energy will be obtained from the flue gas that otherwise will be thrown away or used in another energy recovery devices in the plant.

#### **2.4.5.2 – Bagasse Storage**

Because bagasse is a very bulky material its storage is difficult. Except in very dry locations bagasse not stored under appropriate shelter will ferment and lose a large portion of its value. Atchison (1986) comments about the deterioration of bagasse stored improperly. He reports that losses of up to 30 % occur as the bagasse deteriorates due to its high moisture content and bulk.

To achieve good preservation of the fiber properties and to keep storage losses to a minimum, bagasse moisture content has to be reduced to a level below 20% during storage. By doing this the microorganisms that damage the cellulose fiber cannot live or become inactive if they do survive.

Agblevor et al (1994) report that bagasse stored as it leaves the mill has losses of 10 to 20 % after one year of storage. Madnaik and Jadhav (1996) reference these losses as 25 %. And, the Sugar Industry Research Institute in Mauritius measured 22 % loss of

dry matter in stored bales of bagasse (USAID, 1989). These losses are mainly due to the deterioration caused by biochemical reaction produced by microbial life that proliferates in this fertile environment. These microbial activities result in the loss of structural and non-structural carbohydrates by the destruction of the cellulose fiber. While this happens the temperature may rise to levels where spontaneous combustion can occur. The rise in the temperature is due to the exothermic chemical reactions produced by the microflora (Upadhiaya, 1991).

When stored in big piles Agblevon et al (1994) concluded that the outer layer of the pile provides sacrificial protection for the interior of the pile. The outer layer loses about 23 % of its yield of hydrocarbons while the interior presented some moist areas with microbiological activity but most of it remained in good shape after 26 weeks. Vega Ribera and Garcia Rodriguez (1989) point out that the interior of the storage pile is preserved in an anaerobic condition. This diminishes the degradation because the aerobic microorganisms that attack the fiber cannot be developed.

Despite the fact that it is well known that pile storing is not a good storage method, it is the most common way to store bagasse. Thayer and Roach (1995) suggest this method to be used on a new bagasse powered plant in India, "The bagasse will be stored in the open in piles covered with plastic".

### ***2.4.5.3 – Bagasse Drying Mechanisms***

Bagasse drying can be achieved by different means. The following are mentioned as ways the moisture has been extracted from the bagasse in the past. Some of them are better documented than others, but at least a brief reference is given to all of them.

#### ***2.4.5.3.1 – Hydraulic Presses***

These dryers are not common in the sugar industry but are used in some applications where moisture levels are very high, up to 70 %. Hydraulic presses squeeze water from the biomass reducing its moisture to about 55 %. This is more or less the moisture level at which the bagasse leaves the mill. Due to the large amount of power required by this kind of dryer, they are only used when the moisture content is very high and other means of drying would not be effective. This situation should not happen in the sugar industry, since the objective of the mill is to extract most of the juice present on the cane.

#### ***2.4.5.3.2 – Steam Dryers***

Only one reference to this kind of dryer was found, that was to the so-called exergy dryer described next.

##### ***2.4.5.3.2.1 – The Exergy Dryer***

Developed in Sweden this drying technique dries moist fibrous materials in an atmosphere of steam. The heating medium is steam at high pressures (Pavgi and

Shinagore, 1988). The net energy consumption of this system is supposed to be low when high pressure steam is available and the low pressure generated steam is used as process steam. Pavgi and Shinagare (1988) said that this system could reduce the moisture content of the bagasse to 12 to 15%.

#### *Description of the system*

As described by Pavgi and Shinagare (1988) this is a closed pressurized system, comprising transport pipes, heat exchangers, ducts, cyclones, rotary feeders and a steam fan. The moist material is fed into the system through a rotary valve. This moist material is conveyed up or down the duct/heat exchanger by the transport steam fan. High pressure, high temperature, steam superheats the transport steam carrying the wet material. The wet material is heated when in contact with the steam and the moisture on it evaporates. The transport steam and the dried material are separated in a cyclone. Then the steam can be sent to process and the dried material to the furnace or storage area.

Pavgi and Shinagare (1988) point out the following advantages of this system:

- No fire hazard or danger of explosion
- No particle emission
- Low net energy consumption. No gases are lost through the stack to the atmosphere and the steam generated is used for process.
- Ease of operation and control

### *Energy Requirement*

Pavgi and Shinagare (1988) report the following power demand for a 100 tonnes of cane per hour (TCH) factory (30 tonnes of bagasse per hour (TBH)):

- Blower: 125 kW
- Steam: 8.8 ton/hr at 21 atm saturated (6848 kW). This steam leaves the system as condensate (2235 kW).

#### *2.4.5.3.3 – Hot Gas Dryers*

Hot gas dryers can be fired or unfired. The unfired ones are the most common today because they use waste flue gas as the hot gas or source of energy, increasing the steam cycle efficiency. The fired ones require the combustion of additional fuel, which is not a desirable situation.

Several authors have described installations employing bagasse dryers using flue gas. In India, Bose et al (1994) report their experience with a dryer that reduced the moisture content about 10 percentage points with a payback period of around 3 years and generating more steam while saving bagasse. As will be seen in the following sections good experiences have also been reported in Brazil and Cuba.

##### *2.4.5.3.3.1 – Free Fall Dryer*

This was the earliest type of dryer that has been reported. E. W. Keer designed this kind of dryer in 1910 and reported its performance in 1911. The bagasse and the air travels in counter flow, the bagasse falling against a stream of air whose velocity is less

than the necessary drag velocity so that suspension is avoided. The wet solid is dropped from the top, and from the bottom the drying gas enters at a speed lower than the drag speed for the small particles. The wet gases leave from the top and the dried particles from the bottom. Inside the equipment deflectors may be placed to slow down the fall of the particles and provide longer contact between the gas and the solid. This dryer is not popular and the one built in 1910 was the only reference found.

#### *2.4.5.3.3.2 – Rotary Dryer*

The working principle of these dryers is simple. It consists of a single or multiple passage of hot gases in parallel with the bagasse to be dried inside a rotary drum. The rotary drum has a slight inclination to allow the bagasse to slide down through the equipment. Hernadez Fonseca (1986) describes it as basically a drum or cylinder that rest on four rollers that allow the drum rotation with respect to its axis by means of a motor-transmission system. Through one extreme the bagasse and the hot gas is fed and at the end a cyclone separates them. Inside the drum deflectors distribute the bagasse through the drum increasing the gas-bagasse interaction. After passing through the drum the bagasse is separated from the wet gas in a large cyclone. Correia Maranhao (1994) refer to these dryers as big and with low efficiency, occupying large areas and offering complex manufacturing. They also require a conveyor to take the bagasse to and from the dryer.

Boizan Justiz et al (1986) suggest that the popularity of this type of dryer is due to its universal availability, possibility for different purposes. Also, capacity that may vary from 100's to 1000's of kilos per hour are reported.

Pilgrim (1987) reported that these dryers require 0.68~0.71 kWh per tonne of water evaporated. Hernandez Fonseca (1986) and Patureau (1989) said that they need about 25 kWh per tonne of water evaporated. And, Ranaiah and Shikhalikar (1986) reported that they consume 45 kWh per tonne of water evaporated. As can be seen the values given in the references varied considerable and it is hard to estimate what would be a good value to use.

#### *2.4.5.3.3.3 – Pneumatic Dryers*

Generally pneumatic dryers classify the bagasse in two fractions: a coarse fraction and a fine fraction. The fine fraction is dried by using the flue gas from the boiler, its humidity reduced from 50 to 30 % and then sent to the boiler to be used as fuel (Roca Alarcon and Boizan Justiz, 1993). Depending on the industry the coarse fraction can be used as fuel or as raw material in other industrial processes such as paper pulp preparation, boards, etc.

These systems can be rather simple. A typical pneumatic dryer is a column through which the bagasse and hot air travel until reaching the cyclone where they are separated. Then the hot gases go to the stack and the dried bagasse to the boiler or storage room. To separate the particles several pneumatic classifiers have been developed. But

their objective is the same: separate fine particles from bigger particles. During this process hot gas is used and the particles are not only separated but also dried out. These dryers can dry the bagasse in suspension or while it travels through the air stream. Correia Maranhao (1994) states that these dryers are more efficient than the rotary dryers.

Some applications combine the pneumatic transportation with the drying, achieving two goals in one component (Boizan Justiz, 1986). These dryers are supposed to be of easy construction, low cost and low drying time. Wang et al (1990) recommends pneumatic dryers as the best method available. Arrascaeta and Friedman (1987) also recommend pneumatic dryers, pointing out its simple construction, installation and operation. Also it is low in cost, for their application the dryer had a payback period of half a harvest season. Chen and Tso (1989) also reported the successful construction and operation of a drier of this type. Hernandez Fonseca (1986) and Patureau (1989) coincide estimating the energy consumption of these dryers around 35 kWh per ton of water evaporated.

Good experiences have been reported using these dryers. In Brazil Correia Maranhao (1986) reported a 20 % increase in the steam cycle generation efficiency after adding dryers of this type. In Cuba, Cruz et al (1991) report that the operation of a pneumatic bagasse dryer has resulted in bagasse surpluses of the order of 20~30%.

#### *2.4.5.3.3.4 – Gravity Flow Packed Bed Bagasse Dryers*

McGaw and Pilgrim (1988) clearly explain the design considerations of this type of dryer. They reported that moisture content can be reduced 15 % with a residence time of 10 minutes. Due to its simplicity this system appears to be easy and cheap to build. During operation the bagasse passes down through the dryer by gravity on a vertical column, as a packed bed of material. At the same time the drying gas can be passed cross-flow or counter-flow through the bed.

#### *2.4.5.3.4 – Chemical Drying*

##### *2.4.5.3.4.1 – Isoelectric Dehydration*

This technique was well explained by Hidi (1983) at the Eleventh Australian Conference on Chemical Engineering. It is based on the fact that particles hold less aqueous solution at their isoelectric point. The isoelectric point is defined by the pH at which the particle has a net zero charge. This is because hydration of colloid particles depends on their electrokinetic charge. Hidi reported that bagasse particles normally carry a negative charge, and this charge decreases with decreasing pH. The bagasse particles are isoelectric in the 4.5~5.1 pH range and are positively charged at lower pH values.

The tests conducted by Hidi show reduction of the bagasse moisture content of around 1.4 % by chemically adjusting the pH of the bagasse before its last run through the mill. Even though this is very small the investment required to achieve it is very small too,

representing a simple and economical way to reduce slightly the moisture content with little investment.

#### *2.4.5.3.4.2 – Bagatex-20*

This is a process developed in Brazil for treating, baling and storing bagasse. Atchison (1986) explains that Bagatex-20 produces the rapid drying of bagasse in large 600 to 900 kg bales from 50 % to 20 % moisture content, or less, by adding a bio-chemical catalyst that accelerates the micro-biological fermentation of residual sugar in bagasse without the use of external fuel.

Bagatex-20 promises that the bagasse can be stored for more than 24 months without deterioration of the fiber or serious losses in storage. This will allow the mill to store the bagasse year round and to use it for electricity generation or other by-products whenever needed. Bagatex-20 reduces the moisture content to 20 % after the 20<sup>th</sup> day and usually goes down to 15 % after the 30<sup>th</sup> day.

The Bagatex-20 process, as explained by Atchison, includes the following operations:

- Preparation of especial catalytic bio-chemical fluid
- Metering and feeding the raw bagasse to the baling system at 50% moisture content
- Spraying the catalytic fluid on the bagasse at the entrance to the baler feed chute
- Baling the bagasse into dense bales of 600 kg/M<sup>3</sup> density and 0.8 M x 1.5 M x 1.2 M or 1.8 M size, weighting 600 kg or 900 kg, respectively
- Palletizing the bales in units of three in height

- Transporting by fork lift truck to the conditioning or maturing warehouse
- Conditioning for a period of 20 days
- Transporting to final storage and covering with plastic tapaulins to protect the bagasse from the weather
- Loading and transporting to the place of use
- Wire removal and bale breaking
- Feeding the bagasse to the boiler

Dawson et al (1990) reported that there is no substantial difference between the drying of bagasse using the Bagatex-20 process or using the baling and storing procedure recommended by the Bagatex-20 process but without adding the chemical conditioners of the process, or just adding a solution of 10 % yeast. Private communications with Dr. T. F. Dixon, coauthor of the mentioned report, confirm these conclusions. Many attempts were made to contact any Bagatex-20 officials, with no success. But several researchers in Brazil suggested that the company is out of business.

Considering this, the baling of bagasse can be recommended without the addition of any chemicals. This will only add the cost of baling the bagasse, but not the expensive licensing fees for the process. During the private communications with Dr. Dixon, from the Sugar Research Institute in Mackay, Australia, he confirmed that baling of bagasse would be worth considering when the bagasse is going to be used as fuel.

#### *2.4.5.3.4.3 – FERLAB Process*

The FERLAB process was developed in Venezuela. This is a biological treatment that quickly ferments the soluble residual sugars and other low molecular weight extractables, while maintaining low bale temperature and reducing the fermentation losses in storage (Grozdzits, 1997; Warren and Grozdzits, 1990). The FERLAB treatment is combined with a self ventilating piling methods of the bales to reduce fermentation losses and moisture contents. This process promises the reduction of the moisture content to 25 %. The time required was not included in the report where the information was found, but it is said that this process will allow year round storage of the bagasse.

Despite the fact that details about the bio-chemicals used for both FERLAB and Bagatex-20 are not disclosed in the literature these two processes look very similar to each other. Decision of which one to use or whether to consider either will probably be determined by the licensing fees and operating cost of each.

#### ***2.4.6 – Corrosion in Heat Recovery Equipment***

Fuels containing Sulfur will generate sulfur trioxide ( $\text{SO}_3$ ) in the flue gas. The amount formed varies depending on the amount of sulfur present in the fuel, the amount of excess air and the firing method.

The combination of sulfur trioxide and condensed water vapor forms sulfuric acid ( $\text{H}_2\text{SO}_4$ ). Thus, if the temperature of the flue gas is below the water dew point, water will

start condensing on the heat exchanger surfaces, combine with the sulfur trioxide to form sulfuric acid, and corrosion will start. The corrosion would not be limited to the heat recovery equipment but may also attack other structures that are in contact with the cooler flue gas. Attack of acids in the stack has often been observed. To prevent this, two measures are commonly taken. First, as just discussed, prevent the gas temperature from reaching the dew point temperature. Second, provide cast iron jackets over the steel exposed to this cooler gas when is condensation is unavoidable.

#### ***2.4.7 – Steam Turbines in the Sugar Industry***

Steam turbines in the sugar industry have been present almost since steam started to be produced at the mills. They were used to power the mills and extract the juice from the sugar cane. Steam turbines for the sugar industry do not differ from the steam turbines used by other industries. Steam turbines can be classified in four basic types described below.

- Back Pressure
- Condensing
- Extraction (pass out)
- Mixed Pressure (pass in)

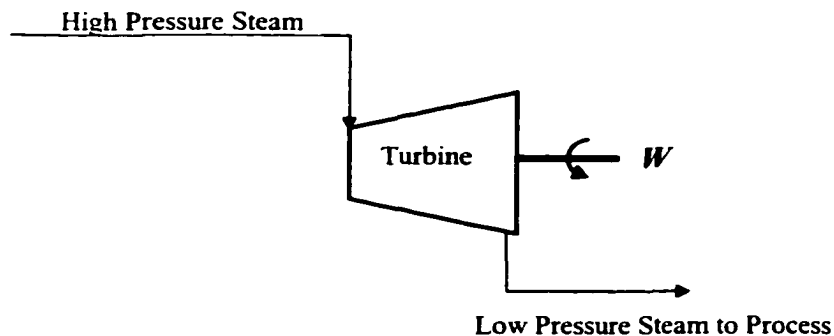
#### 2.4.7.1 – Back Pressure Turbines

Due to their simplicity and lower cost they have been the most common in the sugar industry (Platt, 1980; Bowell, 1996). High-pressure steam coming from the boiler passes through the turbine where it is expanded down to the process pressure (see Figure 2.13). The process pressure varies from mill to mill. Bowell (1996) suggested that this pressure varies from 2 to 3 bars (200 to 300 kPa).

The advantages of these turbines are:

- Relatively low cost
- Operators are already familiar with back pressure turbines
- Atmospheric condensers require less cooling water than vacuum condensers.

Simple devices like dump condensers can be used and only water make up is needed to replace the water that is evaporated.



**Figure 2.13: Back Pressure Turbine**

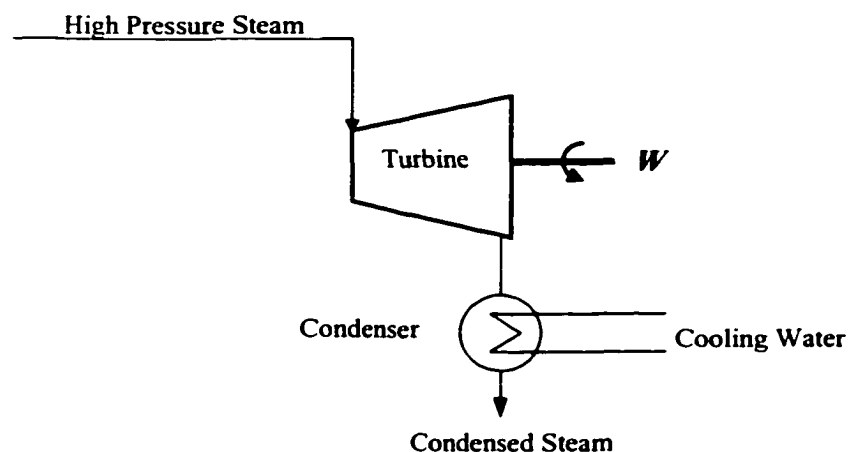
The big disadvantage of these turbines is that the steam condensed in the atmospheric condenser wastes all the energy that it contains.

#### 2.4.7.2 – Condensing Turbines

These turbines take the high-pressure steam from the boiler and expand it to a vacuum pressure much lower than the atmospheric pressure (see Figure 2.14). The steam exhaust the turbine into a vacuum condenser at a pressure of 0.1 to 0.2 bar (10 to 20 kPa) (Bowell, 1996). The condenser can be incorporated within the turbine (for small units) or it can be a separate heat exchanger. Generally water to water condensers are used, but if cooling water is a concern and space is available water to air condensers can be used.

The advantages of condensing turbines are:

- Maximize the power generation
- Waste less energy, making the cycle more efficient



**Figure 2.14: Condensing Turbine**

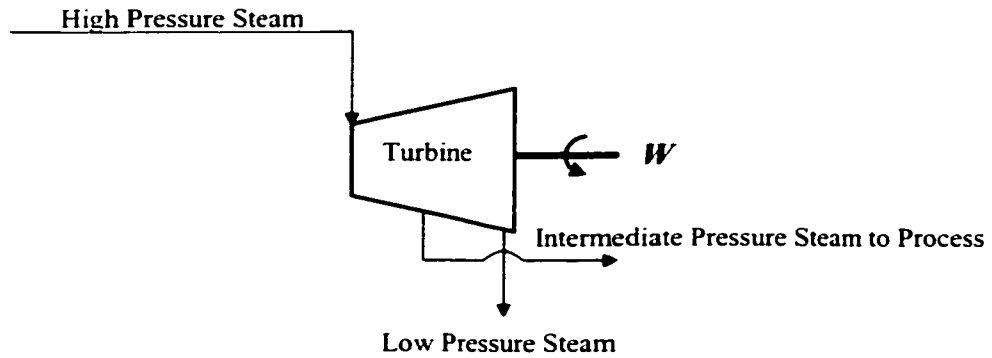
The disadvantages are:

- Higher cost than back pressure units (Ablitt, 1993)
- Higher volume. Since it has to accommodate the condenser and has more stages
- Majority of mill personal may not have experience with this type of turbine
- Requires more cooling water and equipment. **Bowell (1996) reports that condensing sets require approximately 20 times more water than back pressure ones. Since dump condensers cannot be used all the cooling must be provided by cooling water.**

In RONCO (1987) Kehin Kai mentioned the great benefit of using a modern condensing turbine that could have an efficiency of 78 % compared with the earlier non-condensing turbines used in the sugar industry that have efficiencies of less than 50 %. Defining efficiency as the fraction of the steam energy that enters the turbine that is effectively converted into power.

#### *2.4.7.3 – Extraction Turbines*

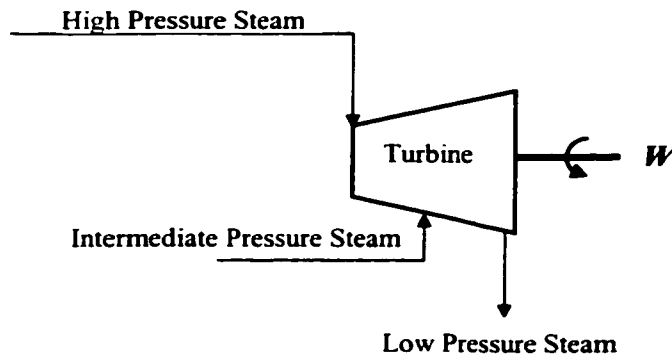
The extraction turbine could be either an extraction back-pressure or an extraction condensing type. In this kind of turbine a fraction of the steam leaves the turbine, “it is extracted”, between the inlet and the exhaust at the pressure required by the process (see Figure 2.15). The pressure and flow of the extracted steam are regulated by a set of valves and determined by the process demand.



**Figure 2.15: Extraction Turbine**

*2.4.7.4 – Mixed Pressure Turbines*

These turbines have two steam inlets. The regular high-pressure inlet and another inlet at a lower pressure somewhere down the machine (see Figure 2.16). They are usually used when steam generated at a different pressure is available. **Bowell (1996)** says that these are unlikely turbines to be used in the sugar industry.



**Figure 2.16: Mixed Pressure Turbine**

### ***2.4.8 – Turbine Selection***

Traditionally back-pressure turbines have been the preferred choice of sugar mills. This occurred because these turbines are the most economical option to generate electricity only for the mill's consumption. One of the options that mills now look at to improve their revenues is to sell electricity to the grid instead of buying from it. This is usually achieved by improving the efficiency of steam use. Reducing leaks, recovering condensate, eliminating blow downs, etc.

Conducting an energy audit is the first step towards increasing the steam availability to produce electricity. After this is done, turbine options have to be evaluated. When there is a market for electricity, condensing turbines may become an interesting option, over the back pressure type, since they allow us to transform more of the energy available in the steam into electricity. The big challenge that mills have experienced when trying to do this is to be able to sign a long-term agreement with the utility to sell the generated electricity at profitable prices. So far this has been difficult, and has led to the installation of less expensive but also less productive back-pressure turbines over condensing turbines. This happened in Fiji, where a back pressure turbine was recently selected and installed instead of a condensing one because a long-term agreement with the local utility was not reached (Komen and Koster, 1994; **Bowell and Abblit, 1995**). This agreement must basically guarantee the purchase of the amount of energy that the condensing system could produce for a period of time that allow the amortization of the cost of the equipment.

Despite these experiences, since every project has its own particularities, the selection of condensing turbines cannot be ruled out. They must be considered in every project, particularly on those where there is a possible market for surplus electricity.

#### ***2.4.9 – Some Conclusions from the Literature Review***

A surprise from the review of boiler technology, was that despite of the type of furnace design, all of them can achieve around the same overall efficiencies.

From the literature review of economizers and air pre-heaters it was concluded that they are often used in the sugar industry. In many cases they have been installed long after the plant has been put into operation as a way to improve the efficiency of the cycle with minor investment.

Air pre-heaters have been preferred over economizers when the fuel has a high moisture content. The reason for this is the assumption that the hotter air helps the combustion process thereby improving the efficiency of the cycle.

Further research, experimentation and development needs to be done to design an optimum bagasse dryer. Even though several types of dryers are being used commercially there is no agreement on their relative advantages, costs and positive consequences.

Although not a standard practice, the use of bagasse dryers has become more common in the past 10 years. Pneumatic and rotary dryers seem to be the popular ones, but no consensus has been found about their advantages and disadvantages. Something that is clear is that for a bagasse dryer to be useful a considerable amount of heat has to be available as waste. In other words a system that has good steam generation efficiency will not be greatly improved by the use of a bagasse dryer.

The different means of drying bagasse are economically feasible under different economic conditions. For example, Bagatex-20 has been used in Brazil successfully but when evaluated under particular conditions for a project in Jamaica (USAID, 1989) and another in Costa Rica (Tuwell et al, 1988) it was not feasible, particularly due to the high initial licensing cost.

An area that needs further research is the drying characteristics of the bagasse. Some work has been done to simulate this, but when the equations reported were used the results obtained were inconsistent.

Corrosion can happen in the heat recovery equipment and other parts of the boiler if the temperature of the flue gas is lower than its dew point temperature. This can be avoided by keeping the temperature of the gas always above its dew point. This will ensure that sulfuric acid will not form on the equipment surfaces.

Condensing turbine will maximize the power generation for a given amount of steam. The problem is that condensing turbines are much more expensive than simple back-pressure turbines. The large cost difference is the reason why back-pressure turbines are the most popular in the sugar industry. To ensure the recovery of the additional investment necessary to install a condensing turbine for extra-power generation, a long-term contract between the mill and the electric company must be signed. **Bowell and Ablitt (1995)** explain that this is something that most utility companies are not willing to do, mainly because it will be a seasonal generation.

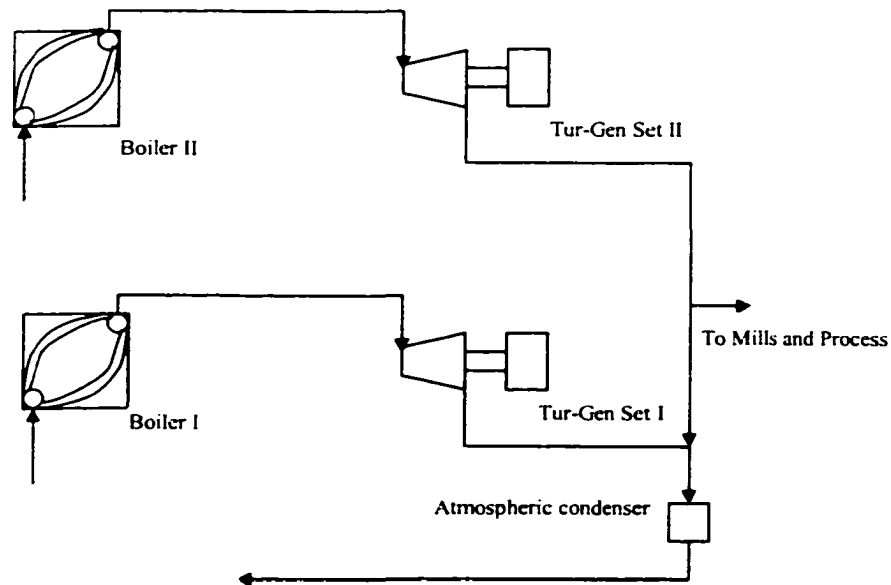
## ***2.5 – Co-generation Cycles in the Sugar Industry***

After an extensive literature review it can be concluded that no particular cycle configuration is used for co-generation in the sugar industry. Different arrangements of condensing and non-condensing turbines are used mainly depending on the electrical demand. If the mill has a high electricity demand it may choose condensing turbines, if not back pressure turbines may be chosen.

Many mills show arrangements of several boilers and turbines operating at the same conditions (Figure 2.17). This duplicating practice is presumed to be justified as a preventive measure in the case of failure of any of the components. However, **Robert H. Hughes**, a well know sugar specialist, reports that the experience of the sugar industry in Hawaii has shown that the equipment is so reliable that this practice does not add any

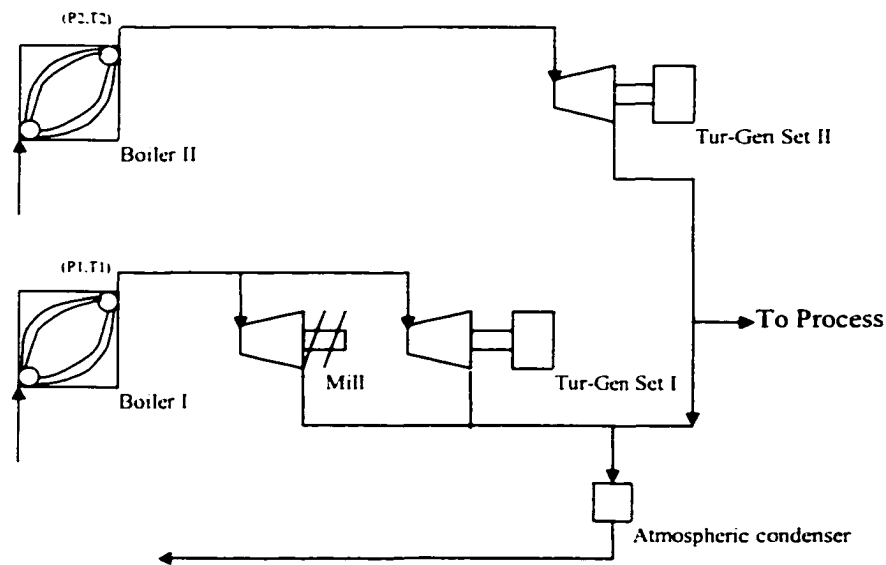
advantage to the systems (USAID, 1986). On the other hand, duplication requires a considerably higher initial capital investment. Another common situation found in the literature is the configuration of systems generating steam at different conditions (P, T) and overlapping others, this is probably a consequence of the growth or upgrade of the mill (Figure 2.18). Instead of taking the old system off line a new one was added to use the new resources.

Throughout the literature review no reheat or regeneration systems were found. It seems that these cycles are not used in this industry. The reason for this may be the relatively higher capital cost of these systems at these sizes when compared with utility power plants.



**Figure 2.17: Duplicated System**

From any of the co-generation cycles three products are demanded. High-pressure steam for power generation, medium pressure steam to move the mills, and low pressure steam for process. Generally the milling is done using steam at medium pressure, this steam is expanded to a lower pressure and then used as a source of heat for the sugar production (process steam).

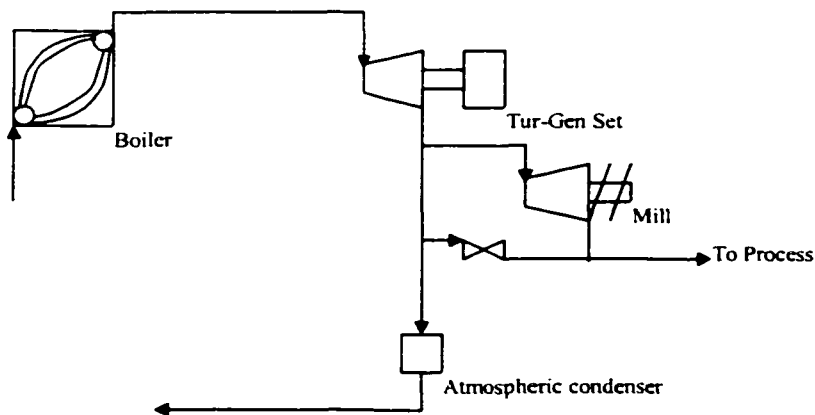


**Figure 2.18: Upgraded System**

### 2.5.1 – Common Co-generation Cycles

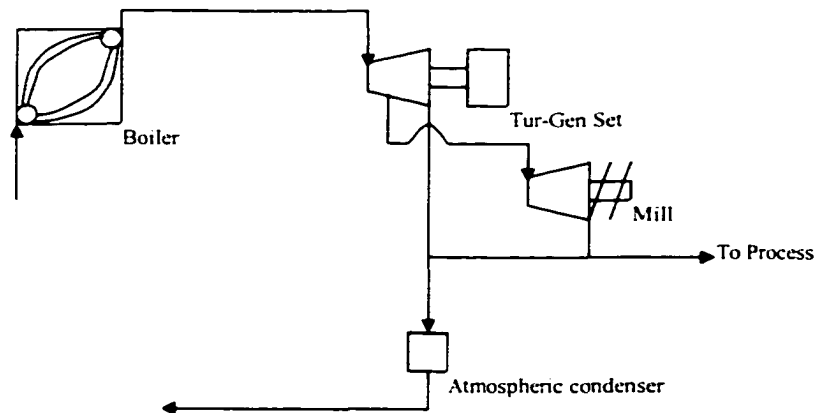
The following is a description of the cycles most commonly found in the literature. It is noticeable that the main differences are the steam turbines used and their location in the cycle. A detailed description of the arrangement of the heat recovery equipment is not given.

Figure 2.19 shows the simplest and therefore the most common cycle used. In this configuration a back-pressure turbine topping the mill is employed. The steam is generated at a high pressure and exhausts the turbine at the pressure required by the milling turbines. After the steam exhausts the milling turbines it is used as a heat source at the mill (process steam).



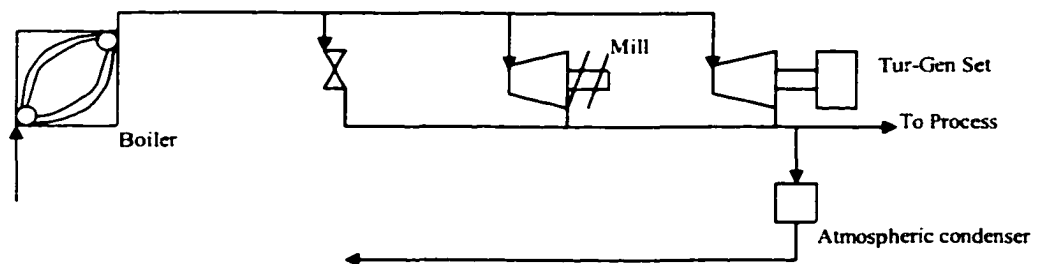
**Figure 2.19: Common Arrangement**

Aiming to achieve higher efficiencies steam turbines with steam extraction are used. The single extraction steam turbine (SEST), Figure 2.20, allows only the extraction of the steam necessary for the milling turbines and expands the rest of the steam to a lower pressure in order to obtain more energy from it. Finally, the steam is exhausted and used as process steam.



**Figure 2.20: Single Extraction System**

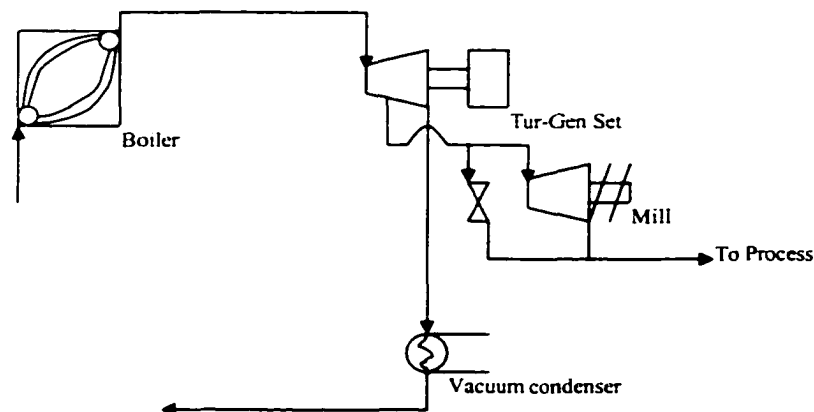
On systems where steam is generated at a low pressure (1.0 to 3.0 MPa) the milling and power generation turbines are generally in parallel, see Figure 2.21.



**Figure 2.21: Milling parallel to power generation**

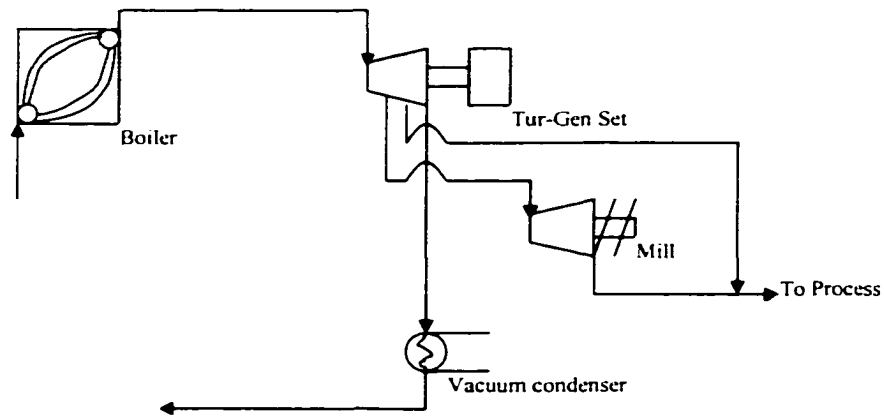
Even though the configuration of Figure 2.21 is usually found using medium pressure steam it could also be used with high-pressure steam. In this way the steam required for milling would be reduced and that available for power generation increased.

Similar configurations but using condensing turbines are present in the industry. These permit a substantially higher power generation capacity. However, the initial capital investment is also substantially higher. The successful implementation of these systems is based on the capacity of the mill to sell all the surplus power. Figure 2.22 shows a single extraction condensing steam turbine (SECST) where the extraction goes to satisfy the milling and process steam demands and the rest of the steam is expanded to a much lower pressure through the power generation turbine.



**Figure 2.22: Single Extraction Condensing System**

Figure 2.23 shows a double extraction condensing steam turbine (DECST) that is meant to get even more energy out of the steam. Here a medium pressure extraction gets the steam necessary for milling and a lower pressure extraction gets that necessary to satisfy the steam process needs.



**Figure 2.23: Double Extraction Condensing System**

### ***2.5.2 – Summary***

Four cycles have been identified as being representative of the current state of the art for co-generation cycles used in the sugar industry. In addition to them reheat and regeneration cycles are going to be included in the cycle simulation discussed in Chapter 3 even though no reference to them was found in the literature review because of their potential for enhanced performance.

Using data from typical sugar mills the simulations were carried out using these cycles and diverting the flue gas through the different heat recovery components. This permitted the assessment of the effect of using back pressure turbines versus condensing ones. single extraction versus double extraction, high pressure for milling versus medium pressure, and recovering heat in different proportions in the different heat recovery components.

## ***2.6 – Gasification of Bagasse***

Even though gasification is not going to be covered during the simulations it is an area of ongoing research and there are high expectations for the outcomes of them. This section summarizes a literature review done on the subject and it is provided as a background since it may be of interest for some people in the co-generation industry.

Gasification of a fuel is its partial oxidation to convert a solid fuel into a gaseous fuel or combustible gas. As on any other conversion system the goal is to maximize this conversion. To do this the solid carbon conversion and the heating value of the combustible gas should be maximized.

Joyce et al (1999a) stated that the first step in gasification is the devolatilisation which happens in a fraction of a second at a typical gasification temperature of 800~900 °C, and in the case of bagasse results in a conversion of approximately 85% of the

feedstock (on an ash free mass basis). Second, cracking of the tar happens rapidly and converts the material that became tar (about 5% of the initial mass) to light gases. Last, the remaining char gasifies at a rate 100~1000 times slower than the devolatilisation step.

In order to maximize the energy conversion, the Integrated Gasification Combined Cycle (IGCC) is used. This cycle involves the thermal conversion of the bagasse into a gas and its combustion in a gas turbine. The thermal energy contained in the exhaust gas of the turbine is used to produce high pressure steam for further power generation in a steam turbine and low pressure steam for process heating applications.

Despite the fact that the research on gasification of bagasse is a hot topic in this industry, “it is obvious that the technology is still very much in its infancy”, stated Hobson and Dixon (1998). Turn (1999a) also states that this technology is still in the development stage and no commercial system is available.

The interest in this technology is due to the promising estimates on the power generation that range from 285 to 460 kWh per tonne of cane (Lobo et al, 1992; Hurtado, 1989; Vasudevan and Shivanand, 1994). So far the systems that have been built to demonstrate/develop the technology have not been economically feasible, despite the higher generation potential.

### **2.6.1 – Biomass Integrated Gasification Combined Cycle (BIGCC)**

As stated before this cycle combines the use of a gas turbine and a steam turbine to maximize the energy use and power generation. This cycle can come in many variations with designs that use considerably different equipment. For example the gasifier can be classified into at least 6 different physical configurations (Joyce et al, 1999):

- Fixed bed
- Mechanically moving bed
- Bubbling fluidized bed
- Circulating fluidized bed
- Multiple fluidized bed
- Entrained flow

But as stated by Turn (1999a) most of the development efforts currently under way have selected bubbling or circulating fluidized bed technologies.

The heating of the gasifier can be direct or indirect. Referring to the heat necessary for gasification, it can be provided by partial oxidation of bagasse (direct) or from some other process which takes place externally to the gasification vessel (indirect). The gasification agent also varies, it can be air, oxygen, steam, or any combination of them. Oxygen is the least preferred since the additional expense to produce a concentrated oxygen stream can make the project economically not feasible. Finally,

pressure is another variable since there are designs that operate in pressurized or unpressurized mode.

As expected, each of these options and variables, has its advantages and disadvantages. What is surprising is to find so many options, variations, on the proposed designs and so few projects developed. Later in this section a brief overview of currently developed projects and projects under development is included.

### ***2.6.2 – Gas Conditioning***

All authors concur that gas conditioning is a necessary component of the BIGCC system so that the gas can be used in combustion turbines. Turn (1999a) points out the need to manage higher hydrocarbons (mostly benzene and naphthalene) present in the gas. These compounds may condense at lower temperature (<400 °C) forming tar and oil that may create operating difficulties and damage to equipment. Two methods of managing higher hydrocarbons are to maintain all system components at temperatures in excess of their condensation point, or, to convert them into permanent gas species (CO, H<sub>2</sub>, CH<sub>4</sub>, etc) by thermal or catalytic reforming techniques.

Ash and incompletely converted carbon also need to be removed from the gas before it can be admitted in the combustor of the turbine. A cyclone can remove the bigger particles, but a secondary system needs to be used to remove the finest ones and

get the gas into specifications. Two methods of filtering the gas are hot gas filters (for gas at temperatures over 500 °C) and bag house filters (for gas temperatures below 300 °C) at atmospheric conditions. Hot gas filters use ceramic candle filters which can stand temperatures above 1000 °C and are cleaned by back pulsing with high pressure gas. When using bag house filters some type of wet scrubbing device is used to further reduce the gas temperature and remove condensable hydrocarbon species and water vapor before compressing it to the turbine inlet pressure. Since biomass ash contains a small amount of alkali additional care must be taken to remove it. These species are responsible for deposition and hot corrosion inside the turbine.

### ***2.6.3 – Gas Turbine***

There are two type of gas turbines considered for BIGCC systems. The first is the heavy duty, industrial turbine designed for stationary utility power generation. The second is the light weight, compact aeroderivative gas turbines used for both stationary and marine applications. Since gas turbines are designed to burn fuels with higher heating values than gas from biomass, the combustors and control valves of the turbines often need to be modified for efficient operation. Turn (1999a) reports that this has been successfully accomplished.

#### ***2.6.4 – Feeding Systems***

One of the most problematic components of the BIGCC is the feeding system. Those systems that are pressurized are the ones with the most problems. In general fuel drying has been incorporated into most BIGCC systems under development. Drying has been done using flue gas from a boiler or using the producer gas that later is combusted. The latter uses the dryer to remove moisture from the bagasse, so that it reaches the moisture content desired for the gasification. At the same time moisture is added to the producer gas, increasing its mass and resulting in a higher power output from the turbine. Drying to levels below 20 % moisture has been associated with feeding problems, especially in extrusion type feeders, due to reduced lubricity and the increased frictional characteristics and abrasiveness of the fuel (Turn, 1999b).

Turn (1999b) found out that the most common feeding system uses lock hopper devices. This device basically isolate the feeding system from the reactor pressure, allowing solids to be introduced without permitting back flow of the material in the gasifier (fuel and oxidizer). Most designs also include a fuel reservoir and metering devices to allow the controlled operation and control of the gasifier.

### **2.6.5 – Gasification Projects**

As mentioned earlier, despite the many designs suggested only a few projects of biomass gasification has been built. Joyce et al (1999) said that there have been 21 gasification projects with possible large scale application using bagasse as fuel. Joyce also said that only one of these projects has been operated commercially.

Turn (1999b and 1999c) gave a good review of the gasification projects that potentially could use bagasse as fuel, identifying four major efforts summarized below:

- **The Renugas Process:** Developed by the Institute of Gas Technology, this process uses a pressurized, bubbling fluidized bed gasifier with air and steam as fluidizing agents. A pilot plant was built in Tampere, Finland, as a demonstration plant and currently operates as a test facility. The plant has operated on paper mill waste, straw and coal mixtures, alfalfa stems, and a variety of wood fuels. The Renugas process was also put to the test in Paia, Hawaii. The project begun in 1993, known as the Biomass Gasifier Facility, is the only project that has been intended to use bagasse as its only fuel. It is a pressurized system with a plug screw feeder which serves as a pressure seal. The gasifier agent is air and steam. The feeding system has been the component that has given the most trouble in this project. The latest information suggest that this project has been terminated (Joyce et al, 1999) and that offers to remove and use the equipment elsewhere are welcome (CCI, 1999).

- **Bioflow Ltd.:** Bioflow Ltd. Is a joint venture company formed in 1992 to market the pressurized, circulating fluidized bed gasifier technology developed by Sydkraft AB of Sweden, and Foster Wheeler Energy International Inc. The technology development began in 1991. An installation was built in Värnamo, Sweden, and operates at 20 bars and 950 to 1000 °C. Provided with a lock hopper system and injection screw feeding system it has used wood chips and bark as fuel. This BIGCC system is designed to operate generating 6 MWe and 9 MWth for district heating. The electrical efficiency of this plant is reported to be 32 %.
- **TPS Termiska Processor AB:** This company based in Sweden has developed an atmospheric pressure gasification system using air-blown, circulating fluidized bed technology for biomass and refuse fuels. In 1986 a pilot plant was built with a capacity to generate 2 MW of thermal energy and tested the use of wood bark, mixtures of wood and polyvinyl chloride (PVC), and refuse fuels. Later developments in 1987 added a 500 kW modified diesel engine that operated during 750 hours to demonstrate the technology. The TPS technology was put to use in Greve-in-Chianti, Italy, where two 15 MW thermal units have been built. The facility has a boiler where gas is burned and raise steam which powers a 6.7 MWe condensing steam turbine. This technology has also been proposed for a demonstration project in Brazil, where it will use wood as fuel. Finally a location in Macuri, Southern Bahia state, was selected to build a 32 MWe plant, this is the latest information found about this project and its current status is unknown. In Eggborough, U.K., a 10 MWe project is also under development, this project will use wood supplied by a willow plantation.

- **Battelle Process:** In the late 1970's, Batelle began development of indirectly-heated gasification technology to produce a medium heating value gas from bagasse. The process features circulating fluidized beds, and steam as gasification agent. A research unit using 9 tonnes per day of fuel was constructed and a 200 kWe turbine was installed. Among the fuels tested are pine, poplar, and switch grass. This technology was scaled up into a demonstration plant at the McNeil Generating Station in Burlington, Vermont (USA). This facility is wood-fired and will generate 5 MWe in a gas turbine.

Not included in the work done by Turn and Joyce are the gasification systems built by PRN Energy Systems in the USA. They have successfully built several gasification units worldwide using rice husks as fuel. In conversation with the president of this company he ensured that the system will operate without problem if bagasse is used as fuel. This is an atmospheric, fixed bed system that uses modified diesel engines as generation units. This system is suspected to be less efficient than the BIGCC system mentioned earlier since the remaining ash has a composition of about 30 % carbon (by weight) when using rice husk as fuel.

### **2.6.6 – Summary**

Despite the expected higher efficiency of the BIGCC process when compare with the Rankine cycle, the economic barriers have not been overcome and none of the demonstration projects have been able to generate electricity at lower cost. This is a young technology born in the 70's that is still in the research phase. It is not expected, in the short term, that it will be economically more competitive than conventional systems. Therefore, it is not expected that a commercial facility will be built in the near future. One of the technical aspects that needs further development is the bagasse feeding system. This was the source of the biggest problems encountered by the designers of the Biomass Gasifier Facility built in Hawaii. The research seems to be focused in both pressurized and atmospheric-pressure gasifiers using bubbling and circulating bed reactors.

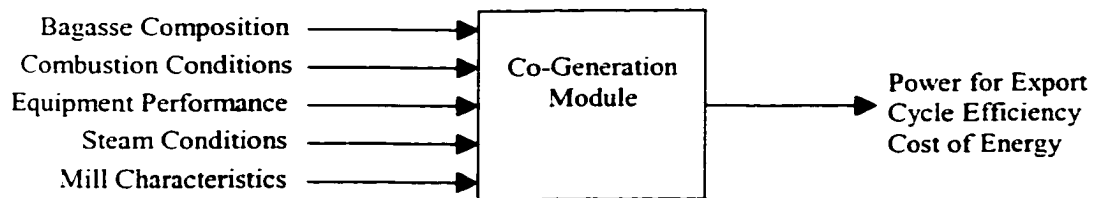
## *Chapter 3*

# **Modeling and Plant Simulation**

In this chapter the models used for the simulation of a sugar plant with co-generation are described. First a global view of the complete simulation is given, along with required inputs and overall outputs. Next, models for each component of the plant are described as independent modules of the simulation. For each of these the required inputs and outputs are given. Also included are results from modeling the combustion process, drying process and the boiler performance, obtained by doing simulations independently from the overall plant simulation. These results are included to quantify and understand these processes that are not look in detail during the overall plant simulation, and they were also used to validate the models by comparing with results reported by others. The results confirmed conclusions obtained by other authors, like the benefits of bagasse drying, negative effect of bagasse moisture content and excess air on the adiabatic flame temperature, etc.

### ***3.1 – Plant simulation***

The modeling of the co-generation plant was accomplished through a combination of component models that simulate each one of the components of the plant. After all these components models are connected, the result is a global model for the plant requiring several user inputs and giving the outputs of the simulation (Figure 3.1). The Co-Generation Module and its inputs and outputs are identified by using the word “main” to distinguish them from the inputs and outputs of the component models.



**Figure 3.1: Co-Generation Module**

#### ***Main Inputs***

The inputs of the main module (Co-Generation Module) can be divided into five groups; bagasse composition, excess air requirement, equipment efficiencies, steam condition, and, mill characteristics. All of these are user supplied. Values from the literature review presented in Chapter 2 can be used.

*Bagasse Composition Inputs* These provide the composition of the bagasse necessary to simulate bagasse combustion. The dry basis composition and the moisture of the bagasse must be supplied. From the literature review, as discussed in Chapter 2, a typical bagasse composition is presented on table 3.1. With a moisture content of 50 %, this composition was used in the simulations performed in this research with the belief that it represents a realistic sample of sugar cane bagasse.

**Table 3.1: Typical Bagasse Composition (% by weight dry basis)**

<b>Specie</b>	<b>Content [%]</b>
C	44
H	6
O	48
Ash	2

*Excess Air Requirement* The amount of excess air provided for the combustion process depends on the boiler equipment. Values from 50 to over 100 % are commonly found in the industry. On a well designed and operated boiler the burning of wet bagasse is normally carried out with excess air which varies between 60 and 50 %. With dried bagasse it may fall to 20 % (Correia Maranhao, 1994). Albert-Thenet (1991) also suggested similar reductions in the excess air when the moisture is reduced. In his investigation Correia Maranhao concluded that the higher the moisture content in the bagasse the higher the need of excess air. This is due to the need for more air to keep a high combustion efficiency and remove water vapor. During the simulations the excess air was determined by the moisture content following the table below (Table 3.2). The values of the excess air in this table were generated by linear interpolation between the

excess air values suggested by Correia Maranhao at 20 and 50 % moisture content. For bagasse moisture contents below 20 % the excess air rate was fixed at 20 %.

**Table 3.2: Excess Air relative to Moisture Content of Bagasse**

<b>%H2O</b>	<b>ExAir [%]</b>
50	60
45	53
40	47
35	40
30	33
25	27
20	20

*Equipment Efficiencies* The boiler, steam turbine, air pre-heater, and economizer efficiencies must be supplied. Typical values for all of these and ranges for them are discussed in Chapter 2. For the simulations values obtained from the literature review and manufacturers quotes were used, they are summarized on table 3.3 below.

**Table 3.3: Equipment First Law Efficiencies**

<b>Equipment</b>	<b>Efficiency [%]</b>
Boiler	70
Economizer	95
Air Pre-Heater	95
Turbine	75
Generator	95

*Steam Condition* The pressure and temperature of the steam after the boiler is required as an input. The pressure of the steam leaving the turbine is also required. As will be explained in the following sections, three different temperature and pressure sets were considered in the simulations. Also, two condensing pressures (turbine exit pressures), atmospheric pressure and 0.01 KPa were treated. The values used are in table 3.4.

**Table 3.4: Steam Conditions**

<b>Steam (Pressure, Temperature) [MPa, °C]</b>	(2,320), (4.5,460), (6,460)
<b>Pressure after Turbine [KPa]</b>	101 (Atmospheric) 0.01 (Condensing System)

*Mill Characteristics Inputs* These define the power and steam required by the sugar mill. Mill steam consumption and power requirements, determine how much steam will be available for power generation, and how much power is available for export. The values used during the simulations can be found in the following table and more about these values is explained in the Mill Characteristics section of this chapter.

**Table 3.5: Mill Characteristics Inputs**

<b>Milling Capacity</b>	217 TC/hr
<b>Milling Pressure</b>	2.0 MPa
<b>Process Pressure</b>	0.2 MPa
<b>Steam for Milling</b>	15 kWh/TC
<b>Steam for Process</b>	420 kg/TC
<b>Electric Requirements</b>	15 kWh/TC

### ***Assumptions***

*After Process Condensed Return* The steam sent to the mill for processing the cane is assumed to return back to the cycle as water at atmospheric pressure (101.325 kPa) and at 90 °C (363.15 K).

*Make Up Water* Make up water is always necessary in this kind of factory to replace the steam blown to the atmosphere and the steam lost during cane processing. Despite this, and because it varies greatly from one factory to another it was not considered in the analyses done. One of the first upgrades that a mill needs to undertake is to minimize the waste of steam, therefore, it is expected that the amount make up water should be small if the factory is co-generating. Because of this it is not expected that this have an effect the simulations done.

### ***Main Outputs***

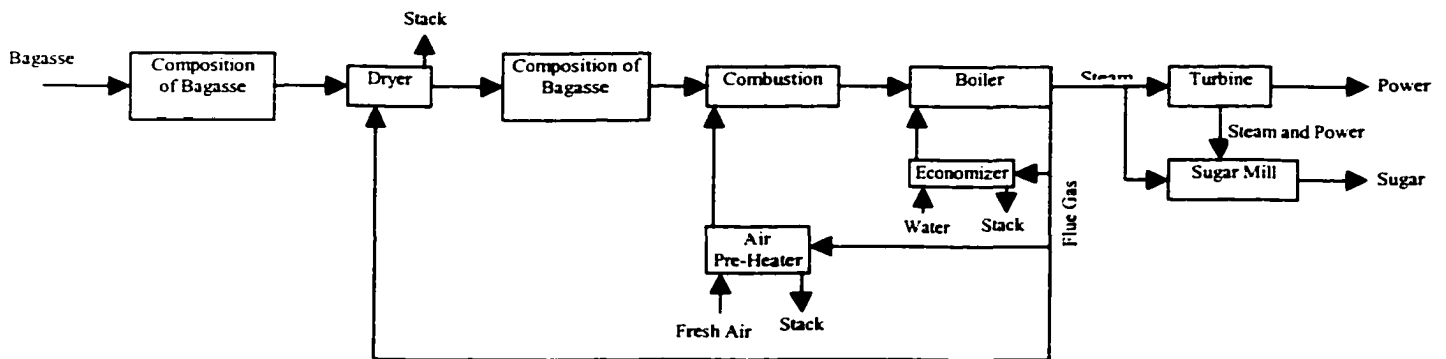
The outputs from the Co-Generation Module are used to evaluate the overall performance of the cycle from the technical and the economic points of view (the later only when economic data is provided). These outputs indicate how much power is generated, how much power is available for export, the overall efficiency of the plant (defined as power produced plus thermal energy used in the plant over energy in the

bagasse burnt), and the cost of the energy generated (power), expressed as US \$ per kWh generated.

### 3.2 – Components Models

Each component was modeled in an independent module with its own inputs and outputs. All modules are related since the outputs of one module are the inputs of others. Figure 3.2 shows the relationship among the different modules. The complexity of their inter-relationships can be appreciated. Note the loops formed after the boiler module when the flue gas is sent to the heat recovery components.

A detailed description of each of the modules (component models) used in the simulations will be presented next. The descriptions include a brief explanation of the inputs, the assumptions, and the outputs of each module, along with a figure representing them.



**Figure 3.2: Modules Relationships**

### **3.2.1 – Bagasse Combustion**

This module simulates the combustion process, using as input the dry basis composition and the moisture content of the bagasse. First, the wet basis composition is calculated. Following this, the wet basis composition of the bagasse and the combustion air supplied are used to calculate the flue gas conditions (temperature, flow rate per kg of bagasse and composition). The equations used to simulate the combustion were developed using basic combustion theory, and include the specific characteristics of bagasse combustion (such as the heating value) whenever needed.

#### ***Inputs***

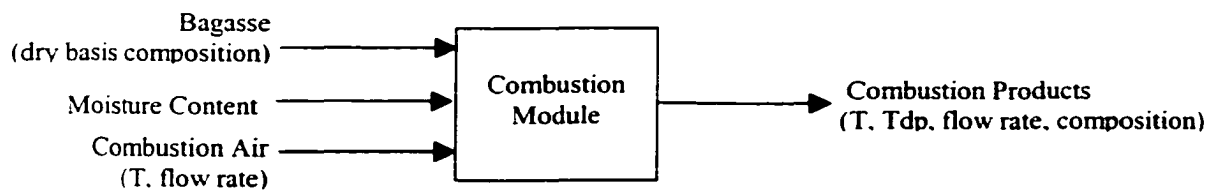
- **Dry Basis Composition of Bagasse:** Percentage of Carbon, Hydrogen and Oxygen present in the bagasse.
- **Bagasse Moisture Content:** This plus the dry composition of the bagasse permits the calculation of the wet basis composition of the bagasse.
- **Combustion Air:** In order to simulate the combustion process, the amount of combustion air provided and its temperature are required. The amount of combustion air is a main input, while its temperature is assumed to be ambient temperature initially. If an air pre-heater is used the actual combustion air temperature is calculated later during the modeling.

### ***Assumptions***

- Ash Content: The ash content of the bagasse is assumed to be the remainder after subtracting the Carbon, Hydrogen and Oxygen contents from 100 %.

### ***Outputs***

- Combustion Products: The temperature (adiabatic flame temperature), flow rate per kg of bagasse, composition, and the dew point temperature of the combustion products are the outputs of this module.



**Figure 3.3: Combustion Module**

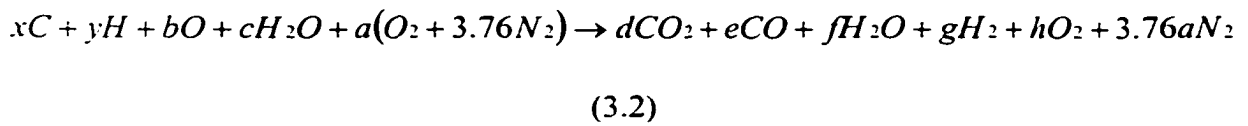
### ***Equations***

From Turns (1996) the lower heating value is:

$$LHV = -\Delta H_R = H_{React}(298.15K) - H_{Prod}(298.15K) \quad [\text{kJ/kgmol}] \quad (3.1)$$

Where:  $LHV$  = Lower Heating Value (*see chapter 2*)  
 $H_{React}$  = Enthalpy of the reactants (bagasse and air)  
 $H_{Prod}$  = Enthalpy of the Products

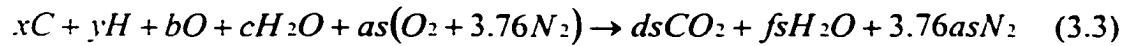
The LHV is calculated directly from Equation 2.1. The composition of the products requires the application of chemical thermodynamics. The general chemical reaction is:



Where:  $a$  = Actual amount of air used (number of moles)  
 $d$  = Number of moles of CO<sub>2</sub> in the combustion products (per kg of bagasse)  
 $e$  = Number of moles of CO in the combustion products (per kg of bagasse)  
 $f$  = Number of moles of H<sub>2</sub>O in the combustion products (per kg of bagasse)  
 $g$  = Number of moles of H<sub>2</sub> in the combustion products (per kg of bagasse)  
 $h$  = Number of moles of O<sub>2</sub> in the combustion products (per kg of bagasse)

The coefficients,  $x$ ,  $y$ ,  $b$ ,  $c$ , and  $a$  can easily be determined from the wet basis composition. However  $d$ ,  $e$ ,  $f$ ,  $g$ , and  $h$  need to be determined. This will be done applying the principles of stoichiometry and chemical equilibrium.

To determine  $a$  the stoichiometric coefficient  $as$  will be determined first:



Where:  $as$  = Stoichiometric amount of air (number of moles)  
 $ds$  = Number of moles of  $CO_2$  in the combustion products (per kg of bagasse)  
 $fs$  = Number of moles of  $H_2O$  in the combustion products (per kg of bagasse)

Balancing all the species we have:

$$\text{Bal. of C:} \quad x = ds \quad (3.4)$$

$$\text{Bal. of H:} \quad y + 2c = 2fs \quad (3.5)$$

$$\text{Bal. of O:} \quad b + c + 2as = 2ds + fs \quad (3.6)$$

Solving these equations we find that the amount of air required for stoichiometric combustion is:

$$as = x + \frac{y}{4} - \frac{b}{2} \quad (3.7)$$

The actual amount of air used,  $a$ , and the equivalence ratio,  $\phi$  are determined as follows:

$$a = \frac{as}{\phi} \quad (3.8)$$

$$\phi = \frac{100}{100 + ExAir} \quad (3.9)$$

Where:  $\phi$  = Equivalence ratio (Fuel/air)/(Fuel/Air)<sub>stoich</sub>

$ExAir$  = Percent Excess Air [%]

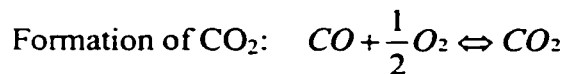
The coefficients d, e, f, g, h, and the flame temperature are found using the species balance implied in Equation 3.2 along with two equilibrium equations, and the energy equation.

$$\text{Bal. of C:} \quad x = d + e \quad (3.10)$$

$$\text{Bal. of H:} \quad y + 2c = 2f + 2g \quad (3.11)$$

$$\text{Bal. of O:} \quad b + c + 2a = 2d + 2h + e + f \quad (3.12)$$

The equilibrium chemical reactions used are:



$$k_{p1} = \frac{d}{e * h^{1/2}} * n_{tot}^{1/2} * \left(\frac{P_o}{P}\right)^{1/2} \quad (3.13)$$

$$n_{tot} = d + e + f + g + h + 3.76a \quad (3.14)$$

$$k_{p1} = \exp\left(\frac{-\Delta G_{T1}}{R * T_{af}}\right) \quad (3.15)$$

$$\Delta G_{T1} = gfCO_2 - gfCO \quad (3.16)$$

Where:  $k_{p1}$  = Equilibrium constant of the reaction  
 $n_{tot}$  = Total number of moles [kmol]  
 $P_o$  = Standard-state pressure, by convention taken to be 1 atm  
 $P$  = Pressure at which the reaction happens [atm]  
 $\Delta G_{T1}$  = Standard-state Gibbs function change [kJ]  
 $g_f x$  = Gibbs function of formation for specie  $x$  [kJ/kmol]

Water Formation Reaction:  $H_2 + \frac{1}{2}O_2 \Leftrightarrow H_2O$

$$k_{p2} = \frac{f}{g * h^{1/2}} * n_{tot}^{1/2} * \left(\frac{P_o}{P}\right)^{1/2} \quad (3.17)$$

$$n_{tot} = d + e + f + g + h + 3.76a \quad (3.18)$$

$$k_{p2} = \exp\left(\frac{-\Delta G_T}{R * T_{af}}\right) \quad (3.19)$$

$$\Delta G_{T2} = g_f CO_2 - g_f CO \quad (3.20)$$

Where:  $k_{p2}$  = Equilibrium constant of the reaction  
 $n_{tot}$  = Total number of moles [kmol]  
 $P_o$  = Standard-state pressure, by convention taken to be 1 atm  
 $P$  = Pressure at which the reaction happens [atm]  
 $\Delta G_{T2}$  = Standard-state Gibbs function change [kJ]  
 $g_f x$  = Gibbs function of formation for specie  $x$  [kJ/kmol]

In writing the energy equation, the enthalpy of the reactant (in this case is the bagasse) is obtained from the definition of the lower heating value, equation 3.1. However, we need to include the effect of the radiation losses (1~5%) and the unburned bagasse (0.5~6%) (Dixon, 1999; Harel and Baguant, 1992; Ramajah and Chickhalikar, 1986). This is done by introducing efficiencies which reduce the LHV as follows:

$$H_{Bag} = LHV * \eta_R * \eta_U + H_{Prod(298)s} \quad (3.21)$$

Where:  $\eta_R$  = Radiation losses efficiency (95~99 %)

$\eta_U$  = Losses due to unburned bagasse efficiency (94~99.5 %)

$$H_{Prod(298)s} = d_s * h_{CO_2(298)} + f_s * h_{H_2O(298)} + 3.76 * a_s * h_{N_2(298)} \quad (3.22)$$

Assuming that the bagasse is at 298.15 K the energy conservation equation can be written as follows:

$$H_{React} = H_{bag(298K)} + H_{air(T_{air})} = H_{Prod(T_f)} \quad (3.23)$$

$$H_{Prod(T_f)} = d * h_{CO_2(T_f)} + e * h_{CO(T_f)} + f * h_{H_2O(T_f)} + g * h_{H_2(T_f)} + h * h_{O_2(T_f)} + 3.76 * a * h_{N_2(T_f)} \quad (8.24)$$

$$H_{Prod(T_f)} = d * h_{CO_2(T_f)} + e * h_{CO(T_f)} + f * h_{H_2O(T_f)} + g * h_{H_2(T_f)} + h * h_{O_2(T_f)} + 3.76 * a * h_{N_2(T_f)} \quad (8.24)$$

$$H_{air(T_{air})} = a * (h_{O_2(T_{air})} + 3.76 * h_{N_2(T_{air})}) \quad (3.25)$$

Where:  $h_x(T_f)$  = Enthalpy of the specie x at the flame temperature

Using these equations, several calculations were done to evaluate the effect of the excess air, and the moisture content, on the flame temperature and combustion products. The results of these calculations are presented later in this chapter (section 3.4). The flame temperature was chosen as the parameter of reference for the combustion, because it represents the maximum possible temperature inside the boiler at any time. This is a very relevant parameter for the design and it can be used as a limit to prevent:

- Reaching the maximum temperature allowed by the equipment specification
- Reaching the melting point temperature of the bagasse ash

### ***3.2.2 – Boiler Modeling***

The objective here was not to model all the events that happen inside a boiler and with that determine its efficiency. Instead, using data provided by manufacturers and from the different bibliographical sources consulted, the efficiency of the boiler was assumed, and the steam production was determined that way. In this module the energy transfer from the hot combustion products to the water/steam in the boiler is simulated. By assuming a characteristic efficiency for the boiler and with the steam conditions given, we can determine the steam flow rate and the conditions of the combustion products leaving the boiler.

Though this module is called the boiler module, what is really modeled here is only the heat transfer process inside the furnace. It is assumed that combustion has

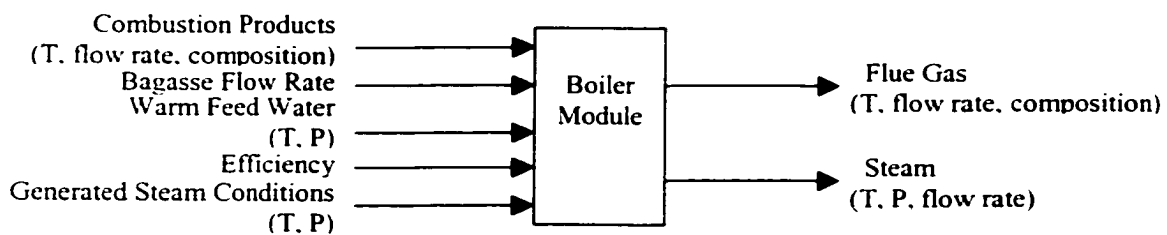
already occurred and only the interaction between combustion products and water/steam is modeled. The economizer and air pre-heater also affect the boiler performance. Both are analyzed in following sections and their effects on the boiler are reflected on the feed water temperature and the temperature of the fresh air entering the combustion module.

### ***Inputs***

- **Combustion Products Conditions:** The temperature (flame temperature) and flow rate of the hot gas entering the boiler.
- **Feed Water Conditions:** The temperature and pressure of the feed water entering the boiler. Initially it is assumed that the water is at the same temperature as the water coming from process (90 °C). As the simulation progress this temperature is recalculated to include the effect of the water coming from the condenser. When an economizer is used, the temperature is recalculated.
- **Efficiency of the Boiler:** This is a main input obtained from the bibliography review. As discussed earlier in this chapter a value of 70 % based on LHV was assumed.
- **Generated Steam Conditions:** The pressure and temperature of the steam leaving the boiler are supplied by the user as main inputs.
- **Bagasse Flow Rate.** Determined assuming that bagasse accounts for 30 % of the sugar cane input to the sugar mill.

## ***Outputs***

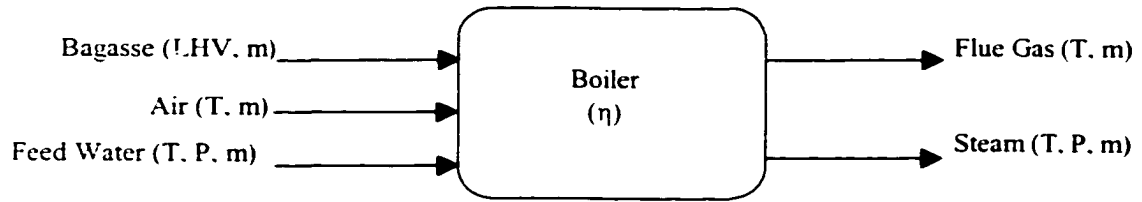
- **Flue Gas Conditions:** The flue gas temperature and flow rate leaving the boiler and entering the heat recovery components is calculated.
- **Steam Flow Rate:** The steam flow rate is the other important output of this module. As input we had the pressure and temperature desired. During the calculations the amount of steam generated at this condition is calculated.



**Figure 3.4: Boiler Module**

## ***Equations***

Figure 3.5 shows the energy balance of a boiler. The variables to be accounted for are the flow rates, temperatures and pressures of the incoming water and fuel/air mixture, and the outgoing steam and flue gas generated.



**Figure 3.5: Boiler Energy Balance**

The energy transferred to the steam is calculated using an overall boiler efficiency:

$$\eta = \frac{m_{water} * (h_{out} - h_{in})_{water}}{m_{Bagasse} * (LHV) + H_{air}} \quad (3.26)$$

Where:

$m_{Bagasse}$  = Mass flow rate of the Bagasse [kg/sec]

$m_{water}$  = Mass flow rate of the water/steam line [kg/sec]

$LHV$  = Lower Heating Value of the Bagasse [kJ/kg]

$h_{out}$  = Enthalpy of the water/steam leaving the furnace at given  
P and T [kJ/kg]

$h_{in}$  = Enthalpy of the water/steam entering the furnace at given  
P and T [kJ/kg]

$H_{air}$  = Enthalpy of the air coming from the air pre-heater [KJ]

As seen in chapter 2 the First Law efficiencies (Based on HHV) of the boilers using economizers varies from 65 % to 76 %. Li and Priddy (1985) stated that for every

10 °F increase of the feed-water temperature, the boiler First Law efficiency increases 1 %. Based on this, and from manufacturer performance data (Babcock, 1995; Zurn, 1996; John Thompson, 1997) on economizers designed to be installed on bagasse fired boilers, the furnace efficiency was be estimated. The economizer performance data show increases of the feed water temperature from 49 °F to 135 °F. From this it can be expected that the economizer increases the efficiency of the boiler by 5 % to 14 %. Considering this performance, a furnace efficiency of 60 % based on the HHV of the bagasse was adopted in the simulations for boilers without economizers. This is consistent with what was reported by Hugot (1986). He reported that boiler efficiency (based on HHV) varies from 50 to 65% for systems without economizer. An efficiency of 60 % based on HHV is equivalent to an efficiency of about 70 % based on LHV for a bagasse fired boiler.

Calculations were done to see the effect of the steam temperature and pressure on the Second Law efficiency of the boiler. Results from these calculations are presented in section 3.3. The Second Law efficiency of the boiler,  $\epsilon$ , was calculated using Equation (3.26-a), on which it is assumed that the change in the availability of the bagasse equals the HHV of it.

$$\epsilon = \frac{\Delta A_{steam}}{m_{Bagasse} * HHV} \quad (3.26 - a)$$

Where:  $\Delta A_{steam}$ = Rate of change of the availability flow of the  
water/steam in the boiler [kW]

$HHV$ = Higher Heating Value of bagasse [kJ/kg]

In order to estimate the temperature of the flue gas leaving the boiler the energy transferred to the steam is subtracted from the energy available in the combustion products at the time it enters the boiler. Doing this results in the following equation.

$$h_{prodleaving} = (1 - \eta_b) * (H_{react} - H_{prod 298}) + H_{prod 298} \quad (3.27)$$

Where:

- $h_{prodleaving}$  = Enthalpy of the flue gas leaving the boiler  
[kJ/kgBagasse]
- $\eta_b$  = First Law Efficiency of the boiler
- $H_{react}$  = Enthalpy of the reactants (Bagasse and air)  
[kJ/kgBagasse]
- $H_{prod298}$  = Enthalpy of the flue gas at 298 K [kJ/kgBagasse]

Another parameter to evaluate the performance of the boiler is its effectiveness. The effectiveness,  $Eff$ , is the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer. Even though this parameter was not used for the modeling of the boiler it was calculated for some scenarios to have an idea of its value.

$$Eff = \frac{C_h(T_{h,i} - T_{h,o})}{C_{min}(T_{h,i} - T_{c,i})} \quad (3.28)$$

Where:

- $Eff$  = Effectiveness
- $C_h = m * C_p$  of the hotter fluid
- $m$  = Mass flow rate
- $C_p$  = Specific Heat
- $T_{h,i}$  = Temperature of the hotter fluid entering the heat exchanger

$T_{h,o}$  = Temperature of the hotter fluid leaving the heat exchanger

$C_{min}$  = Smaller between  $C_h$  and  $C_c$

$C_c = m * C_p$  of the colder fluid

$T_{c,i}$  = Temperature of the colder fluid entering the heat exchanger

### ***3.2.3 – Economizer Modeling***

This module simulates the heat transfer from the hot flue gas to the water that goes through the economizer before entering the boiler.

#### ***Inputs***

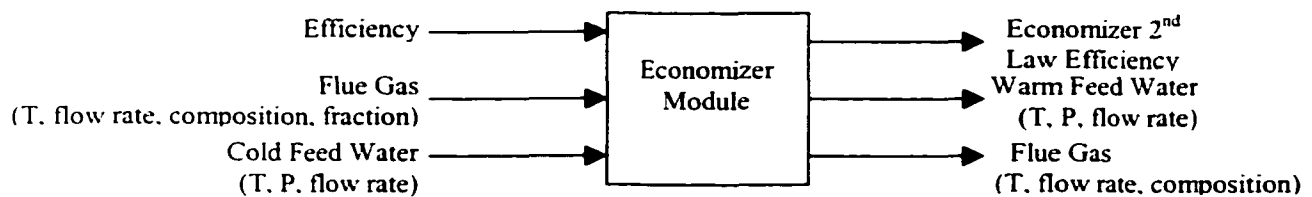
- **Economizer Efficiency:** The economizer first law efficiency is taken to be a characteristic value obtained from the literature reviewed.
- **Flue Gas Temperature:** The temperature of the flue gas as it leaves the boiler.
- **Cold Feed Water Conditions:** The temperature and pressure of this water are assumed to be those of return water as it comes from the sugar mill.
- **Flue Gas Fraction:** The fraction of the total flue gas generated that is diverted to the economizer

### ***Assumptions***

- Flue gas Temperature leaving the economizer: This is set to be at least 25° C above the dew point temperature to avoid corrosion in the equipment.

### ***Outputs***

- Warm Feed Water: The temperature of the water leaving the economizer



**Figure 3.6: Economizer Module**

### ***Equations***

The efficiency of a heat exchanger ( $\eta$ ) can be measured using the First Law of Thermodynamics. Efficiency,  $\eta$ , in this case is the fraction of the heat extracted from the flue gas which is transferred to the feed water. The remaining heat is assumed to be lost to the surroundings. Thus,  $1 - \eta$  is the portion of heat lost. Applying an energy balance to an economizer its efficiency can be determined with the following equation (3.29).

$$\eta = \frac{m_{\text{water}} * (h_{\text{out}} - h_{\text{in}})_{\text{water}}}{m_{\text{g}} * (h_{\text{in}} - h_{\text{out}})_{\text{g}}} \quad (3.29)$$

Where:  $\eta$  = First Law Efficiency  
 $m_{water}$  = Water/steam flow rate [kg/sec]  
 $m_g$  = Flue gas flow rate [kg/sec]  
 $h_{out}$  = Enthalpy of fluid leaving the economizer [kJ/kg]  
 $h_{in}$  = Enthalpy of fluid entering the economizer [kJ/kg]  
 $_{water}$  = This subscript indicate that the fluid is water  
 $_g$  = This subscript indicates that the fluid is flue gas

The efficiency of a typical heat exchanger was assumed to be 95 % based on the examples given by Li (1995). If one assumes that the equipment is well insulated losses of 5 % are reasonable.

Since the First Law of Thermodynamics only requires conservation of energy, a Second Law of Thermodynamics analysis is necessary in order to evaluate how well the energy is transferred in the economizer. In order to estimate this, the irreversibility, or loss of availability in the economizer was calculated. Irreversibility was the parameter used to compare the performance of the heat recovery components considered in this research. For the economizer the equation giving the irreversibility generated during energy transfer can be written as follows:

$$I_{ec} = m_{bag} * X_e * (a_{in} - a_{out})_g - m_{water} * (a_{out} - a_{in})_{water} + \dot{Q}_{surr} \quad (3.30)$$

Where:  $I_{ec}$  = Irreversibilities of the Economizer [kW]

$m_{water}$  = Water/steam flow rate [kg/sec]

$m_{bag}$  = Bagasse flow rate [kg/sec]

$X_e$  = Fraction of the flue gas diverted to the economizer

$a_{outg}$  = Availability of flue gas leaving the economizer  
[kJ/kg of bagasse]

$a_{ing}$  = Availability of flue gas entering the economizer  
[kJ/kg of bagasse]

$a_{outwater}$  = Availability of the water leaving the economizer  
[kJ/kg]

$a_{inwater}$  = Availability of the water entering the economizer  
[kJ/kg]

$Q_{surr}$  = Loss due to heat transfer to the surroundings [kJ]

The exit temperature of the flue gas leaving the economizer was determined by setting it to be 25° above the dew point temperature of the flue gas. Also the temperature of the water leaving the economizer was limited to not exceed the temperature of the flue gas entering the economizer minus 25°. This was done to avoid a design that would suggest an economizer that needed to be really big and could be economically not a good choice. Even though not used as a parameter of the simulations, the effectiveness (as defined in previous section) was calculated for some scenarios to have an idea of its range.

### ***3.2.4 – Air Pre-Heater Modeling***

Here the temperature of the combustion air entering the boiler is determined. The heat transfer from the hot flue gas to the fresh combustion air is simulated and the temperature of both leaving the air pre-heater is determined.

#### ***Inputs***

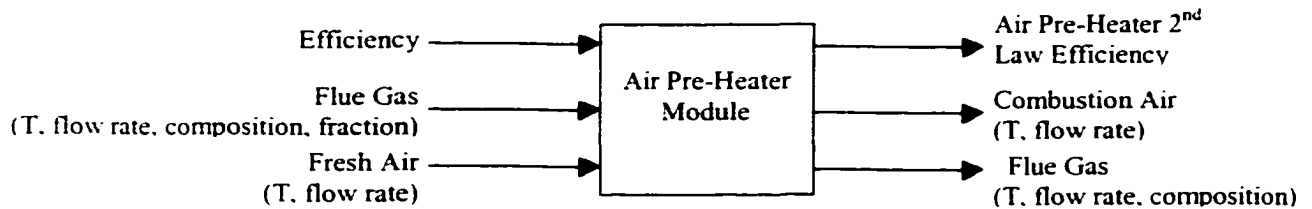
- Air Pre-Heater Efficiency: The air pre-heater First Law efficiency is given as a characteristic value obtained from the literature reviewed
- Flue Gas Temperature: The temperature of the flue gas as it leaves the boiler.
- Fresh Air: The amount of fresh air is determined by the combustion conditions given as main input.
- Flue Gas Fraction: The fraction of the total flue gas generated that is diverted to the air pre-heater

#### ***Assumptions***

- Temperature of the Fresh Air: It is assumed to be ambient temperature.
- Flue gas Temperature leaving the air pre-heater: This is set to be at least 25° C above the dew point temperature to avoid corrosion in the equipment.

## Outputs

- Combustion air Temperature: The temperature of the air leaving the air pre-heater.



**Figure 3.7: Air Pre-Heater Module**

## Equations

The efficiency for the air pre-heater ( $\eta$ ) is calculated in a similar fashion as it is calculated for the economizer. In this case the efficiency,  $\eta$ , is the fraction of the heat extracted from the flue gas which is transferred to the air. As in the economizer the remaining heat is assumed to be lost to the surroundings. The following equation defines this efficiency:

$$\eta = \frac{m_{ca} * (h_{out} - h_{in})_{water}}{m_g * (h_{in} - h_{out})_g} \quad (3.31)$$

Where:  $\eta$  = First Law Efficiency

$m_{ca}$  = Combustion air flow rate [kg/sec]

$m_g$  = Flue gas flow rate [kg/sec]

$h_{out}$  = Enthalpy of fluid leaving the air pre-heater [kJ/kg]

$h_{in}$  = Enthalpy of fluid entering the air pre-heater [kJ/kg]

$ca$  = This subscript indicate that the fluid is combustion  
air

$g$  = This subscript indicates that the fluid is flue gas

In order to estimate the irreversibility in the air pre-heater equation (3.32) below was used. In this case irreversibility measures how much of the available energy in the flue gas is loss during the energy transfer to the combustion air in the air pre-heater.

$$I_{ap} = m_{bag} * X_{ap} * (a_{in} - a_{out})_g - (A_{out} - A_{in})_{ca} + \dot{Q}_{surr} \quad (3.32)$$

Where:  $I_{ap}$  = Irreversibilities of the Air Pre-heater [kW]

$X_{ap}$  = Fraction of the flue gas diverted to the air pre-heater

$A_{outca}$  = Availability flow of the combustion air leaving the  
air pre-heater [kW]

$A_{inca}$  = Availability flow of the combustion air entering the  
air pre-heater [kW]

$Q_{surr}$  = Loss due to heat transfer to the surroundings [kJ]

Similar restrictions were imposed on the fluids in the air pre-heater as the ones imposed in the economizer. The temperature of the flue gas leaving the air pre-heater was

25° above its dew point. Also, the temperature of the fresh air leaving the air pre-heater was limited to be no warmer than 25° below the temperature of the flue gas entering the air pre-heater. Similarly, some calculations of the air pre-heater effectiveness were carried out to get an estimate of its range.

### ***3.2.5 – Dryer Modeling***

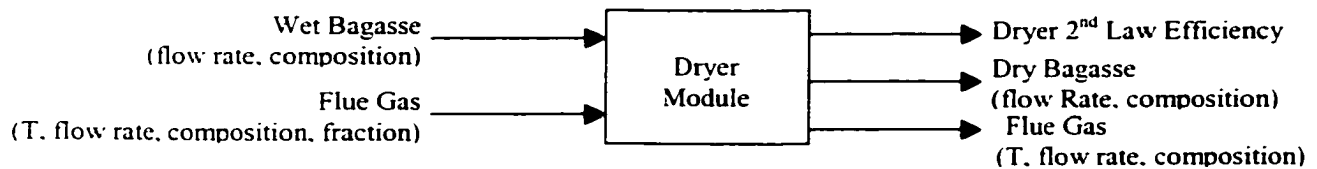
The dryer module simulates the heat transfer from the gas leaving the boiler to the wet bagasse entering the dryer. Here, due to the heat transferred to the bagasse, some water will evaporate and become part of the flue gas.

#### ***Inputs***

- **Wet Bagasse Composition:** The composition of the wet bagasse is necessary to determine the amount of water in it.
- **Flue Gas Temperature:** The temperature of the flue gas as it leaves the boiler.
- **Flue Gas Fraction:** The fraction of the total flue gas generated that is diverted to the dryer

## ***Outputs***

- Dried Bagasse Moisture Content and Flow Rate: A new bagasse moisture content is calculated as well as its flow rate. The output composition of the bagasse in terms of mass fractions must also change and is calculated.
- Flue Gas Composition: Flue gas leaving the dryer has increased its water content and its composition on a mass basis is calculated



**Figure 3.8: Dryer Module**

## ***Equations***

The methodologies of McGaw and Pilgrim, Pilgrim, Duggal et al, etc, described in the literature review are useful to determine evaporation time, but they do not seem to consider the amount of available gas. Since the focus of this research was on the energy used for the drying, not the time for drying, a simple methodology was developed to calculate the drying of bagasse. This methodology is based on the assumption that in the dryer heat extracted from the flue gas is used to evaporate moisture from the bagasse as well as raise its temperature. Part of the heat will raise the temperature of the water and the fibre to the wet bulb temperature of the gas, and the rest is going to actually evaporate

water from the bagasse. The wet bulb temperature of the gas is determined using the equations described in the ASHRAE Journal of Fundamentals (1993). The energy conservation statement (First Law of Thermodynamics) can be written:

$$\dot{Q}_{av} = \dot{H}_{gin} - \dot{H}_{gout} - \dot{Q}_{loss} \quad (3.33)$$

Where:

$\dot{Q}_{av}$  = Rate of heat transfer from the flue gases available for drying the bagasse [kW/kgBagasse]

$\dot{H}_{gin}$  = Enthalpy flow associated with the flue gases entering the dryer [kW/kgBagasse]

$\dot{H}_{gout}$  = Enthalpy flow associated with the flue gases leaving the dryer [kW/kgBagasse]

$\dot{Q}_{loss}$  = Rate of Heat loss to the surroundings [kW/kgBagasse]

By definition  $\dot{Q}_{av}$  is equal the amount of heat transferred to the bagasse:

$$\dot{Q}_{av} = \dot{Q}_b = \dot{Q}_f + \dot{Q}_{wb} + \dot{Q}_{vap} \quad (3.34)$$

$$\dot{Q}_f = \dot{m}_b * C_{p_f} * (T_{wb} - 30) \quad (3.35)$$

$$\dot{Q}_{wb} = \dot{m}_w * (h_{wb} - h_{30}) \quad (3.36)$$

$$\dot{Q}_{vap} = \dot{m}_w * X * h_{fg} \quad (3.37)$$

Where:

$\dot{Q}_b$  = Rate of heat transfer to the bagasse [kW]

$\dot{Q}_f$  = Rate of heat transfer necessary to increase the temperature of the bagasse fiber from ambient temperature, 30 °C, to its dew point [kW]

$\dot{Q}_{wb}$  = Rate of heat transfer necessary to increase the temperature of the water from ambient temperature, 30 °C, to its wet bulb temperature [kW]

$\dot{Q}_{vap}$  = Rate of heat transfer necessary to evaporate the water [kW]

$\dot{m}_b$  = Mass rate of flow of dry bagasse entering the dryer [kg/sec]

$\dot{m}_w$  = Mass rate of flow of water in the wet bagasse entering the dryer [kg/sec]

$H_{\Gamma_{wb}}$  = Enthalpy of water at 1 atm, wet bulb temperature [kJ/kg]

$h_{30}$  = Enthalpy of water at 1 atm, 30 °C [kJ/kg]

$h_{fg}$  = Enthalpy of vaporization of water at its dew point temperature [kJ/kg]

$X$  = Fraction of water evaporated

$C_{p_f}$  = Specific heat of dry bagasse [kJ/(kg\*K)]

From the last equation the fraction of water that is evaporated,  $X$ , can be determined:

$$X = \frac{\dot{Q}_{av} - \dot{Q}_f - \dot{Q}_{wb}}{\dot{m}_w * h_{fg}} \quad (3.38)$$

The final amount of water in the bagasse will be  $\dot{m}_w * (1 - X)$ . So the final moisture content of the bagasse on a mass basis will be:

$$\%H_2O = \frac{\dot{m}_w * (1 - X)}{\dot{m}_b + \dot{m}_w * (1 - X)} * 100 \quad (3.39)$$

Where:  $\%H_2O$  = Moisture content of the bagasse [%]

$\dot{m}_b$  = Mass rate of flow of dry bagasse entering the dryer [kg/sec]

To determine the wet bulb temperature the following equations, taken from ASHRAE (1993) were used:

$$T_{wb} = \frac{2501 * (W - W_{swb}) + 1.805 * W * t + t}{4.186 * W - 2.381 * W_{swb} + 1} \quad (3.40)$$

$$W = 0.62198 * \frac{P_w}{P - P_w} \quad (3.41)$$

$$W_{swb} = 0.62198 * \frac{P_{wsb}}{P - P_{wsb}} \quad (3.42)$$

$$\ln(P_{wsb}) = \frac{C1}{T_{wbK}} + C2 + C3 * T_{wbK} + C4 * T_{wbK}^2 + C5 * T_{wbK}^3 + C6 * \ln(T_{wbK}) \quad (3.43)$$

Where:  $T_{wb}$  = Gas Wet Bulb Temperature [°C]

$T_{wbK}$  = Absolute Gas Wet Bulb Temperature [K]

$W$  = Humidity Ratio

$W_{swb}$  = Humidity Ratio of Saturated Air at the Wet Bulb Temperature

$t$  = Gas Dry Bulb Temperature [°C]

$P_w$  = Partial Pressure of Water Vapor [kPa]

$P_{wsb}$  = Saturation Pressure of Water Vapor in the absence of air at the Wet Bulb Temperature [kPa]

$P$  = Gas Pressure [kPa]

$C1 = -5.8002206 \text{ E } 03$

$C2 = -5.5162560 \text{ E } 00$

$$C3 = -4.8640239 \text{ E } -02$$

$$C4 = 4.1764768 \text{ E } -05$$

$$C5 = -1.4452093 \text{ E } -08$$

$$C6 = 6.5459673$$

This model is appropriate as long as the drying process stays in the constant-rate drying period. From basic theory of solid drying the drying process is done in two phases. First, the constant-rate drying period during which the mass of water removed from the body, in equal intervals of time, remains constant. Second, the falling-rate drying period, during which the mass of water removed, in the same equal intervals of time, systematically decreases. During the constant-rate period the solid remains at the psychrometric wet-bulb temperature. Beginning with the falling-rate period, the temperature of the solid continuously increases. If the drying is continued long enough, the temperature of the solid reaches the temperature of the drying air (Pabis et al, 1998). Despite the fact that no report was found specifying the temperature of the bagasse and drying air leaving dryers, e-mail communications with two researchers in the field report that the temperature of the bagasse leaving the dryer is the wet bulb temperature. Based on this, it was assumed that at least until a moisture content of 20 %, the drying process for bagasse is in the constant rate drying period. During this period the proposed model for drying should be accurate.

In order to compare the performance of the dryer with the other heat recovery components of the plant, economizer and air pre-heater, its irreversibility was calculated with the following equation:

$$I_d = \Delta A_{bag} + \Delta A_{water} + X_d * \Delta A_g + \dot{Q}_{loss} \quad (3.44)$$

Where:  $I_d$  = Irreversibility generation rate in the dryer [kW]

$\Delta A_{bag}$  = Rate of change of the availability flow of the dry bagasse  
(no water) [kW]

$\Delta A_{water}$  = Rate of change of the availability flow of the water in the  
bagasse [kW]

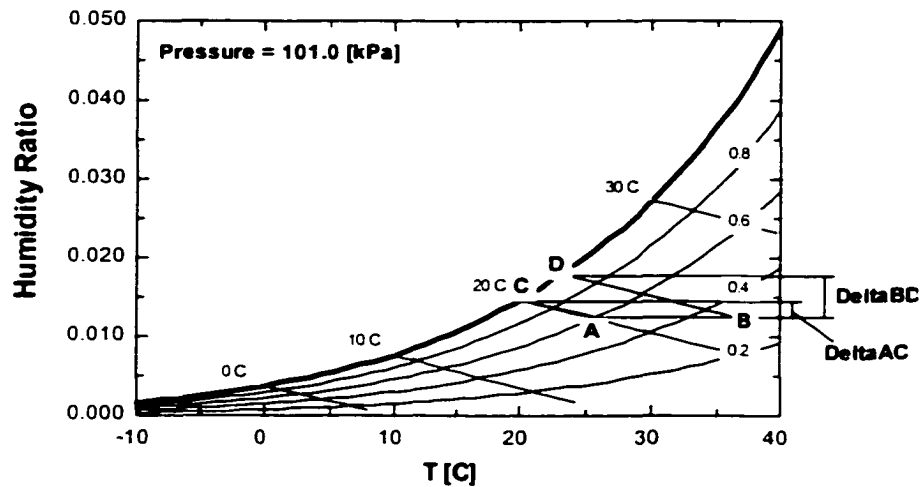
$\Delta A_g$  = Rate of change of the availability flow of the flue gas [kW]

$X_d$  = Fraction of the flue gas diverted to the dryer

$Q_{loss}$  = Rate of heat loss to the surroundings [kW]

The temperature of the gas used to dry the bagasse is very important. The higher this temperature is, the larger is the amount of moisture that can be evaporated from the bagasse. Psychrometric principles can illustrate this fact. Looking at Figure 3.9 we can see that points A and B represent flue gas at different temperatures, but with the same moisture content. The maximum moisture that can be evaporated into the air is limited by its saturation point. Points C and D represent these saturated points for the two different initial temperatures. It is easy to see that the hotter air (point B) will remove more moisture (DeltaBD) than the cooler one, point A, represented by DeltaAC. It is important to keep this in mind so that when drying is evaluated this effect is not over looked. On the simulations this was reflected in the capacity of the flue gas to transfer energy, since a limitation on the lowest possible temperature of the flue gas leaving the dryer was set.

For the dryer, as well as the economizer and the air pre-heater, the temperature of the flue gas leaving these components was set to be 25 °C above the dew point of the gas.



**Figure 3.9: Psychrometric Chart**

### ***3.2.6 – Corrosion in Heat Recovery Equipment***

During the simulations the dew point temperature of the gas was calculated. This was done in order to prevent designing a system that will have condensation. Avoiding condensation consequently limited the amount of heat recovery possible. To determine the dew point temperature of the gas an equation suggested by the ASHRAE Handbook of Fundamentals (1993) was used (Equation (3.45) below). Another approximation that can be used is to calculate this temperature using the Clausius-Clapeyron equation (Equation (3.46) below) with an average latent heat of vaporization of water for the

temperature range between 60 and 100 °C. However, the first option was found to be more accurate when compared with other sources giving data on the dew point temperature.

$$T_{dp} = C7 + C8 * \alpha + C9 * \alpha^2 + C10 * \alpha^3 + C11 * (P_w)^{0.1984} \quad (3.45)$$

Where:  $T_{dp}$  = Dew Point Temperature of the Flue Gas [°C]  
 $P_w$  = Water Vapor Partial Pressure [kPa]  
 $\alpha = \ln(P_w)$   
 $C7 = 6.54$   
 $C8 = 14.526$   
 $C9 = 0.7389$   
 $C10 = 0.09486$   
 $C11 = 0.4569$

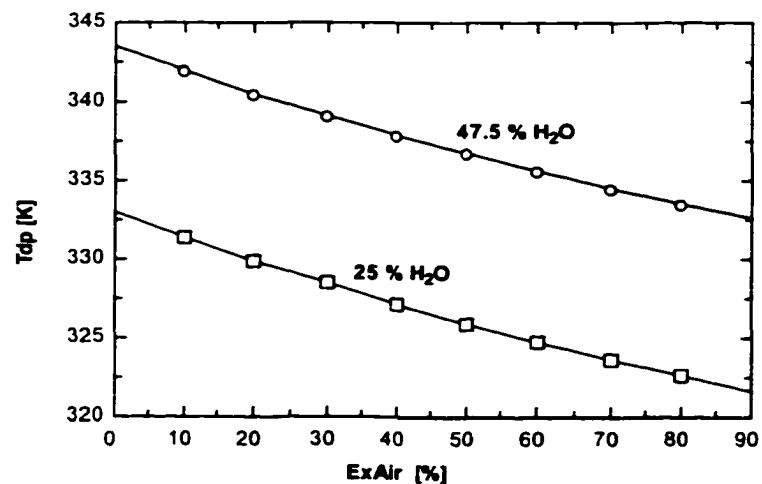
$$\frac{P}{P_{ref}} = \exp \left[ - \frac{hfg_{H2O}}{Ru / MW_{H2O}} * \left( \frac{1}{T_{dp}} - \frac{1}{T_{ref}} \right) \right] \quad (3.46)$$

$$\frac{P}{P_{ref}} = X_{H2O}$$

Where:  $X_{H2O}$  = Molar Fraction of Water in Flue Gas  
 $hfg_{H2O}$  = Latent Heat of Vaporization of Water  
 (Assume 2,307.75 KJ/Kg)  
 $Ru$  = Universal Gas Constant (8.315 KJ/(kmol\*K))  
 $MW_{H2O}$  = Molecular Weight of Water (18.016 Kg/kmol)  
 $P_{ref}, T_{ref}$  = Pressure and Temperature used as references  
 (101.325 kPa, 373.15 K)  
 $T_{dp}$  = Dew Point Temperature of Water in the Flue Gas [K]

Figure 3.10 shows some results from calculating the dew point temperature of the water in the flue gas, for two different bagasse moisture contents as a function of excess air. As expected it can be seen that increasing the excess air and reducing the moisture content in the bagasse both reduce the dew point temperature of the flue gas.

Further calculations were done to evaluate the effects that bagasse drying has on the bagasse combustion process. These effects are: an increase in the calorific value of the bagasse, a decrease in the excess air requirement, an increase of the temperature inside the boiler, an increase of the combustion efficiency, a decrease of the flue gas flow, an increase of the thermal efficiency of the cycle and steam generated per kg of bagasse, a decrease in stack losses, and a decrease in fuel consumption. Details about these calculations are presented in section 3.9 of this chapter.



**Figure 3.10: Dew Point Temperature of Water in Flue Gas vs Excess Air Rate and Moisture Content on Bagasse**

### ***3.2.7 – Turbine Modeling***

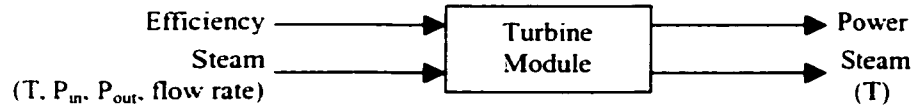
The turbine is simulated in a similar fashion to the boiler. The power generated by the turbine is calculated using inlet steam condition, exit pressure and a user-supplied efficiency.

#### ***Inputs***

- Inlet Steam Conditions: The pressure, temperature and flow rate of the steam entering the turbine.
- Turbine Efficiency: the first law efficiency of the turbine as a characteristic value from the literature review.
- Pressure of the steam leaving the turbine.

#### ***Outputs***

- Power Generated
- Temperature of the Steam leaving the turbine.



**Figure 3.11: Turbine Module**

### ***Equations***

The efficiency, defined as the capacity of the turbine/generator set to convert the heat energy into electricity (equation 3.47) varies depending on the steam conditions and the turbine design. This efficiency is the product of all the efficiencies related to the turbine and the generator. Mechanical efficiency of the turbine, efficiency of the reduction gearing, efficiency of the generator, and the thermodynamic efficiency of the Turbine. From manufacturers data Ablitt (1993) assumes that an average efficiency for steam turbines is 70 %. Analyzing the data provided by Hugot (1986) it suggested that the efficiency of the turbine/generator sets vary from 60 to 80 %. Averaging these values coincides with Ablitt's conclusion. An analysis done on electric generation from bagasse in Thailand (RONCO, 1986) also assumed an overall efficiency of 70% for the turbine.

$$\eta = \frac{\dot{E}}{\dot{m}(h_{in} - h_{out})} \quad (3.47)$$

Where:  $\eta$  = Total efficiency of the turbine/generator set

$\dot{m}$  = Steam Flow Rate

$\dot{E}$  = Electrical power generated

$h$  = Enthalpy of the steam – *in*: entering, *out*: leaving the turbine

The first law efficiency of the turbine can be defined as the actual work out of the turbine over ideal work of a reversible-adiabatic (isentropic) process between same pressures. The actual work is the real change of enthalpy of the steam passing through the Turbine. The ideal work is equal to the isentropic change of enthalpy of the steam going through the turbine. This can be as high as 85% (Equation (3.48) below).

$$\eta_{1st} = \frac{\dot{m}(h_{in} - h_{out})}{\dot{m}(h_{in} - h_{outs})} \quad (3.48)$$

Where:  $\eta_{1st}$  = First Law efficiency of the turbine

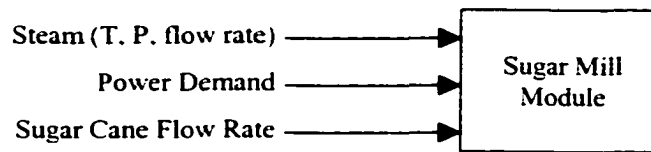
$\dot{m}$  = Steam flow rate

$h$  = Enthalpy of the steam – *in*: entering, *out*: leaving the turbine

$h_{outs}$  = Isentropic enthalpy of the steam leaving the turbine

### 3.2.8 – Mill Characteristics

Several parameters characterize the sugar mill and affect the co-generation cycle. In particular these are the steam consumption, at the different pressures, and the power consumption of the sugar factory.



**Figure 3.12: Sugar Mill Module**

To carry out the simulations, several parameters to describe the sugar mill needs are necessary. To determine average values of these parameters an extensive bibliographical review was done. As expected the data found varied considerably from one reference to another, but after putting them all together and under the same unit system, some typical ranges were determined and values were selected to represent a “typical sugar mill”. Table 3.6 shows the values assumed and a brief description of them follows.

- **Milling Capacity** A really wide range of milling capacities was found in the literature. Those varied from mills as small as 3 tonnes of cane per hour to as large as 1,521 tonnes of cane per hour. The most common range seems to be between 109 and 217 tonnes of cane per hour. A characteristic mill capacity of 217 tonnes of cane was

selected, considering it typical and large enough to allow a significant amount of power for export.

**Table 3.6: Parameters of Typical Sugar Mill**

<b>Milling Capacity</b>	217 TC/hr
<b>High Steam Pressure</b>	6 MPa
<b>HP Steam Temperature</b>	460 C
<b>Milling Steam Pressure</b>	2 MPa
<b>Process Steam Pressure</b>	0.2 MPa
<b>Condenser Steam Press.</b>	0.01 MPa
<b>Electric Requirements</b>	15 kWh/TC
<b>Steam For Milling</b>	15 kWh/TC
<b>Process Steam</b>	420 kg/TC

- *High Steam Pressure* The data reported for steam pressure for power generation varied from 0.6 MPa to 10.5 MPa. Both limits are exceptions. Older, non co-generating mills typically generate at 2.1 MPa, while newer or re-furnished facilities, with co-generation generate at around 6 MPa.
- *Steam Temperature* Corresponding to the maximum steam pressures, the temperature varied from 151 to 513 °C. A temperature of 460 °C is commonly chosen with the 6 MPa pressure. At this point it is worth mentioning that agencies providing financing assistance are setting lower limits to the pressure and temperature of the steam generated for the projects they finance. For example, in India, USAID/India (through the Industrial Development Bank of India (IDBI)), implemented financial assistance

grants that required a steam minimum pressure of 60 Kg/cm<sup>2</sup> (5.89 MPa) and a minimum temperature of 430 °C, for the projects they finance (CCI, 1998).

- *Steam pressure for milling* This pressure did not vary as drastically and 2 MPa is considered typical.
- *Steam Pressure for Process* It varies from 0.1 to 0.3 MPa. A pressure of 0.2 MPa was found to be very common.
- *Steam Pressure at the Condenser* Very few references were found about condensers in this industry. A characteristic value of 0.01 MPa was used in the simulations, when sub-atmospheric condensing was treated as one of the options.
- *Electric Requirements* The co-generation facility must provide power for the sugar mill. The values found in the literature varied from 11 kWh per tonne of canne processed to 40 kWh per tonne of cane. The lower end represents a very energy efficient mill using steam for milling, and the higher end a totally electric mill. For the simulations it was assumed that steam is used for milling and that the mill was moderately energy efficient. With this on mind an electric power consumption of 15 kWh per tonne of mill was assumed.

- *Steam for Milling* The steam demand for milling was assumed to be 15 kWh per tonne of cane. This was a value found to be common on mills that have implemented some energy efficiency.
- *Process Steam Demand* Several references indicate that the low end is close to 400 kg of steam per tonne of cane, while a common situation is about 500 kg of steam per tonne of cane. It is expected that a mill that is going to get in the co-generation business will reduce steam losses and improve the efficiency of its use, with this in mind a process steam demand of 420 kg per tonne of cane was selected.

### ***3.3 – Validation of the Models***

In order to verify and validate the results obtained with some of the models described in the previous sections, several analyses were done using EES<sup>1</sup> (Engineering Equation Solver). The results showed agreement with results reported by other authors and contributed to the understanding of processes that occur during the combustion of bagasse. These analyses were also useful to understand the effects of some variables (excess air, temperature of excess air, bagasse moisture content) on the processes that are difficult to appreciate when simulations of the complete cycle are performed.

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<sup>1</sup> A copy of the EES algorithm can be requested from the Mechanical Engineering Department at Colorado State University

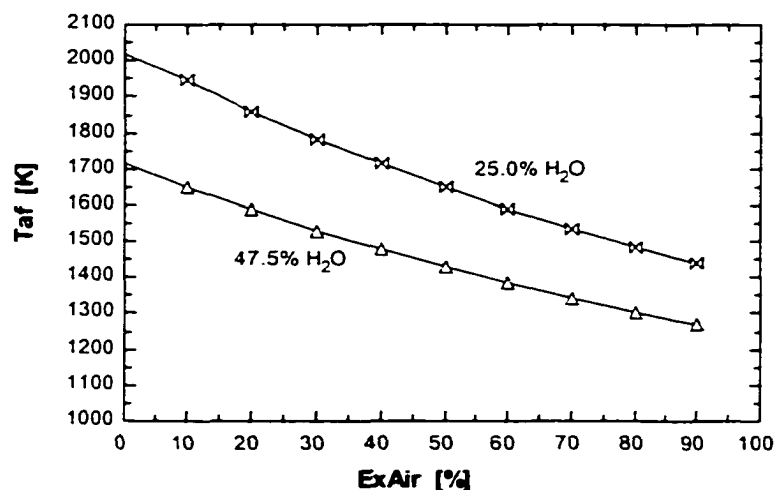
### ***3.3.1 – Bagasse Combustion - Results and Analysis***

#### ***3.3.1.1 – Introduction***

In this section detailed results for the bagasse combustion process are presented. The results show that high moisture content and excess air have a negative effect on the combustion process, decreasing the adiabatic flame temperature. In contrast, increasing the temperature of the excess air has a positive effect, increasing the adiabatic flame temperature.

#### ***3.3.1.2 – Results on Bagasse Combustion***

Using the equations described in the Section 3.3.1, calculations were carried out mainly to show the effect the amount of excess air, temperature of the excess air, and moisture content on the adiabatic flame temperature and the composition of combustion products. The critical results are summarized in Figures 3.13 to 3.18. Figure 3.13 shows the extent to which increasing the excess air causes a decrease of the adiabatic flame temperature. Two curves are shown, for bagasse moisture contents of 25 % and 47.5 %. The curves show a general nonlinear decrease of the adiabatic flame temperature with increasing excess air, with the higher moisture curve being shifted substantially downward from the low moisture curve.



**Figure 3.13: Adiabatic Flame Temperature vs Excess Air**

Figure 3.14 shows that, as expected, the more excess air used the lower the molar fraction of the other species present in the products (Note the difference in scale for  $x_{CO_2}$  compared with  $x_{CO}$  and  $x_{H_2}$ ). The same occurs with the other species not shown in the figure ( $H_2O$ ,  $OH$ ,  $NO$ , etc.). The curves for higher moisture content are shifted downward from those at 25%  $H_2O$ . This makes sense since more air and water are present in the products, meaning that the fractions of water and air are bigger. Also, the reduction in flame temperature with increased moisture shifts the equilibrium composition away from  $CO$  and  $H_2$ . Notice the fast decrease of the fractions of  $CO$  and  $H_2$  as more excess air is provided for combustion.

In Figure 3.15 the effect of preheating the air is shown. The results indicate a nearly linear increase in adiabatic flame temperature with air pre-heat. This is due to the extra energy that is provided by the air to the reaction. This results in a positive effect on efficiency since it allows a higher temperature inside the boiler which leads to better heat

transfer. As seen in Figure 3.16, since the temperature rise is small this preheating does not have a significant effect on the composition of the products.

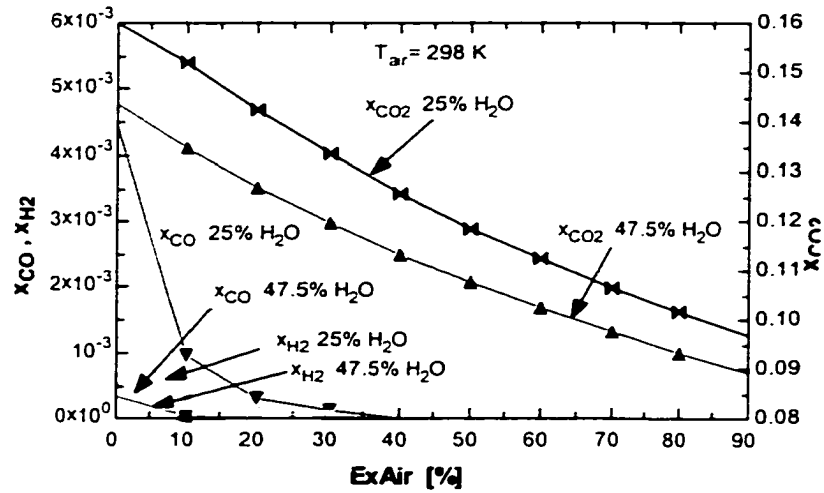


Figure 3.14: Molar Fractions vs Excess Air

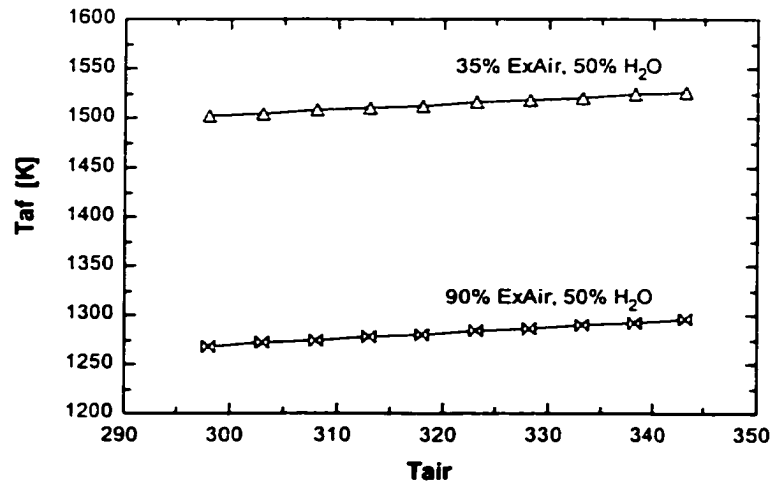
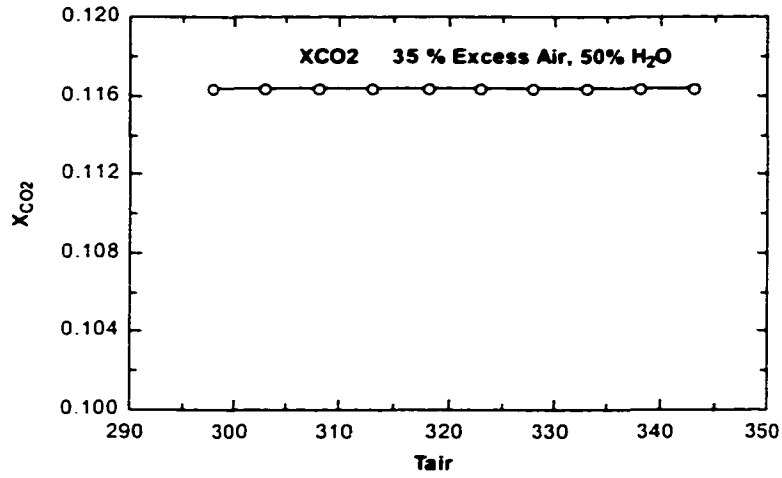
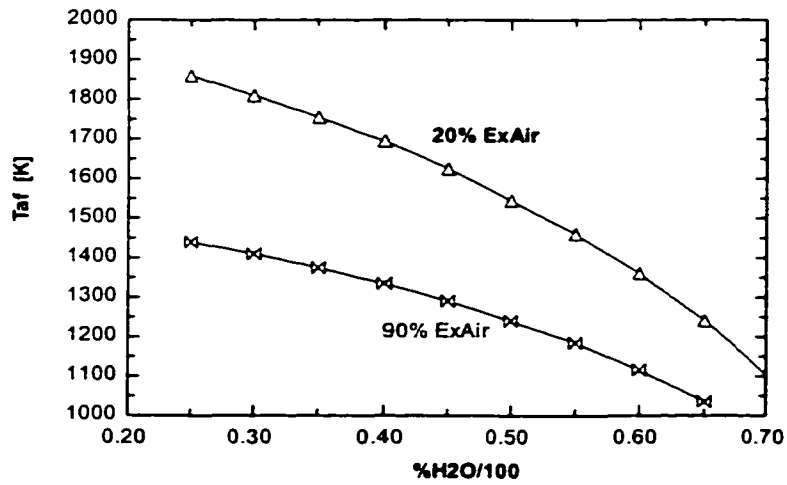


Figure 3.15: Adiabatic Flame Temperature vs Excess Air Temperature



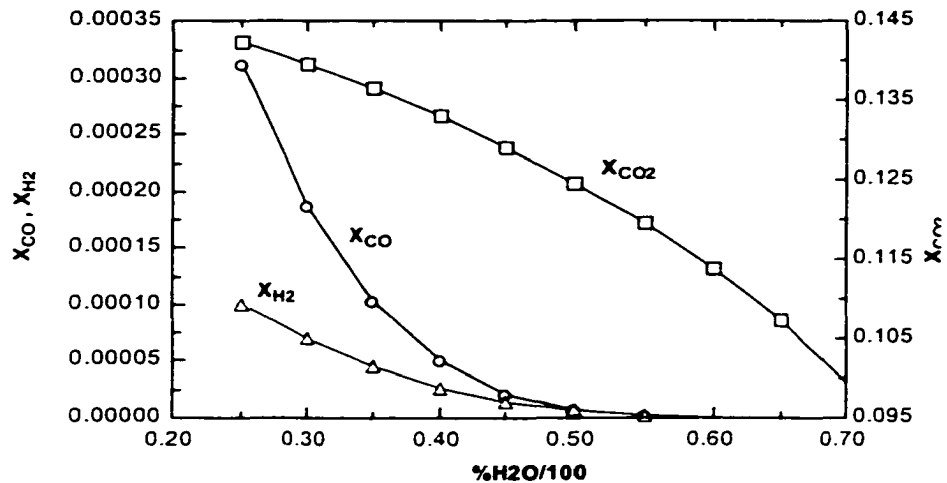
**Figure 3.16: Molar Fractions vs Excess Air Temperature**

Figure 3.17 shows, similar to Figure 3.13, that more moisture and excess air reduce the adiabatic flame temperature.



**Figure 3.17: Adiabatic Flame Temperature vs Moisture Content**

Finally, Figure 3.18 also shows the reduction in the molar fraction of some products as we add more moisture to the fuel.



**Figure 3.18: Molar Fractions vs Moisture Content**

In order to validate these results a software available in the literature, HPFLAME (Turns, 1996), was used to verify the predicted results. The agreement was very good, with differences below 1 % most of the time.

### ***3.3.1.3 – Conclusions***

The results show that to maximize the Adiabatic Flame Temperature it is desirable to do the following:

- Use the least excess air possible, but always enough to ensure a good mixing of the bagasse and the air
- Preheat the air before the combustion
- Reduce the moisture of the bagasse before the combustion in order to have a higher heating value

These results agree with other authors findings and demonstrate how important the design and operation of the furnace is for optimum utilization of the bagasse.

### ***3.3.2 – Consequences of Bagasse Drying***

The drying of bagasse before firing has long been thought to be desirable. Specific benefits are described below.

- *Increase in Heating Value* The reduction of moisture in the bagasse increases its heating value. As seen in Chapter 2, several authors have suggested equations to quantify this. The one suggested by Hugot (1986) is used in this analysis.

$$LHV = (4250 - 12 * i - 48.5 * W) * 4.1868 \quad [\text{kJ/kg}] \quad (3.49)$$

Where:  $LHV$  = Lower Heating Value of bagasse [kJ/kg]  
 $W$  = Moisture content of bagasse [%]  
 $i$  = Non-fibre impurities content of bagasse (2~3%) [%]

Table 3.7 shows the change in the LHV of the bagasse using as reference a moisture content,  $W$ , of 50 % and an impurity content,  $i$ , of 2 %. When the moisture content is reduced, the weight of the sample is reduced too. Thus the sample will be less than 1 kg of bagasse at a moisture content lower than 50 %. The first column,  $W$ , represents

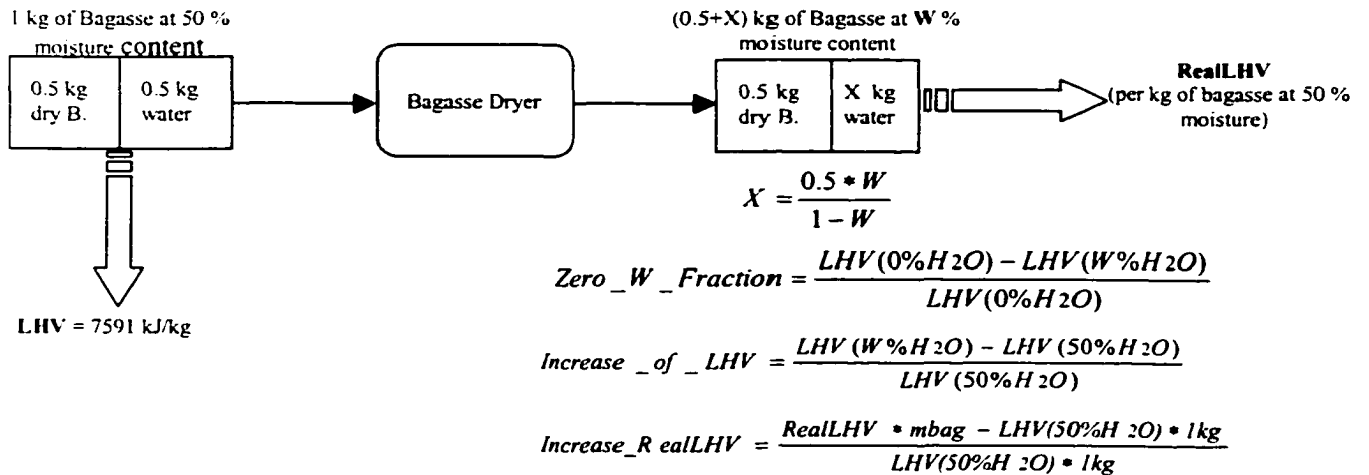
the moisture content of the sample. The second column, *LHV*, is the lower heating value of a 1 kg sample of bagasse at the moisture content to its left. The third column, *Zero W Fraction*, compares the lower heating value at the particular moisture content with that of a sample with zero moisture. In other words this is the percentage of the maximum possible heating value. This increase of the heating value, is due to the availability of heat that otherwise was needed to evaporate the water from the bagasse. The fourth column, increase of LHV, shows how much the lower heating value increases comparing a sample of 1 kg of bagasse at 50 % moisture with another 1 kg sample of bagasse at the moisture content in column one (on the same row). However, due to the loss of moisture, we know that the total mass decreases. Thus, because of drying the total mass of a sample with moisture *W* (column one) decreases to the value represented on the *mbag* column. The next column shows the percentage decrease of the mass compare with the initial 1 kg of bagasse at 50 % moisture content. Taking this into consideration the heating value of the sample referred to its original weight (before drying) of one kg can be calculated. This is designated the “*RealLHV*” and is given in column seven. It represents the real heating value obtained by drying and should be used instead of column two in assessing the relative merits of using a bagasse dryer.

- *Decrease in excess air requirement for boiler operations* A similar analysis is done to estimate the amount of air required for the combustion (see Table 3.8). Here the stoichiometric amount of air for a 1 kg of bagasse sample at the moisture content *W* is calculated and is listed in column two, *as*. However since the initial 1 kg sample is

being dried out to the moisture content  $W$ , the mass of combustible material in it remains the same and the stoichimetric air remains constant, *Realas* (column three). Considering that the burning of wet bagasse is carried out with excess air which varies between 60 and 50 %, while with dried bagasse it may fall to 20 % (Correia Maranhao, 1994), the total amount of air required for the combustion was calculated, *Total air*, and its relative decrease is shown in the last column to the right.

**Table 3.7: Increase in Heating Value of bagasse**

$W$ [%/100]:	LHV [kJ/kg]:	Zero $W$ Fraction [%]:	Increase of LHV [%]:	$mbag$ [kg]:	Decrease of $mbag$ [%]:	RealLHV [kJ]:	Increase RealLHV [%]
0.50	7591.00	42.66	---	1.00	---	7591.00	---
0.45	8601.00	48.34	13.31	0.91	9.09	7819.00	3.00
0.40	9611.00	54.01	26.61	0.83	16.67	8009.00	5.51
0.35	10621.00	59.69	39.92	0.77	23.08	8170.00	7.63
0.30	11632.00	65.37	53.23	0.71	28.57	8308.00	9.45
0.25	12642.00	71.05	66.54	0.67	33.33	8428.00	11.03
0.20	13652.00	76.72	79.84	0.63	37.50	8533.00	12.41



**Table 3.8: Change in the air requirement**

<b>W [%]:</b>	<b>as [kgair/kgfuel]:</b>	<b>Realas [kgair/kgfuel]:</b>	<b>ExAir [%]:</b>	<b>Total air [kgair/kgfuel]:</b>	<b>Decrease of Total air [%]:</b>
0.50	0.0183	0.0183	60.00	0.0292	-----
0.45	0.0201	0.0183	53.33	0.0280	4.17
0.40	0.0219	0.0183	46.67	0.0268	8.33
0.35	0.0237	0.0183	40.00	0.0256	12.50
0.30	0.0256	0.0183	33.33	0.0243	16.67
0.25	0.0274	0.0183	26.67	0.0231	20.83
0.20	0.0292	0.0183	20.00	0.0219	25.00

- Increase of the temperature inside the furnace* Reduction in the moisture content of the bagasse results in higher combustion temperature, which acts to improve the heat transfer rates at the boiler's heating surfaces. Table 3.9 shows how the temperature increases when the moisture content is reduced. The second column from the right, shows the change of the adiabatic flame temperature with respect to the change in the moisture content, while keeping a fixed excess air rate of 20 %. The column next to the right shows the relative increase.

**Table 3.9: Adiabatic flame temperature**

<b>W [%]:</b>	<b>ExAir [%]:</b>	<b>Taf [K]:</b>	<b>Increase of Taf [%]:</b>	<b>Fix ExAir Taf [K]:</b>	<b>Increase Taf Fix ExAir [%]:</b>
0.50	60.00	1352.00	0.00	1547	0.00
0.45	53.33	1442.00	6.66	1624	4.98
0.40	46.67	1531.00	13.24	1693	9.44
0.35	40.00	1621.00	19.90	1755	13.45
0.30	33.33	1713.00	26.70	1810	17.00
0.25	26.67	1808.00	33.73	1861	20.30
0.20	20.00	1906.00	40.98	1906	23.21

By looking at the change in the adiabatic flame temperature (*Taf*, Table 3.9) we can see how it increases dramatically as the moisture content and excess air are reduced.

This is a desirable condition since it improves the combustion efficiency and heat transfer inside the furnace. However, unless the boiler is designed to stand this high temperature and to handle the melting ashes, the maximum flame temperature is limited (to approximately 1873 K) so that slag does not form on the furnace walls.

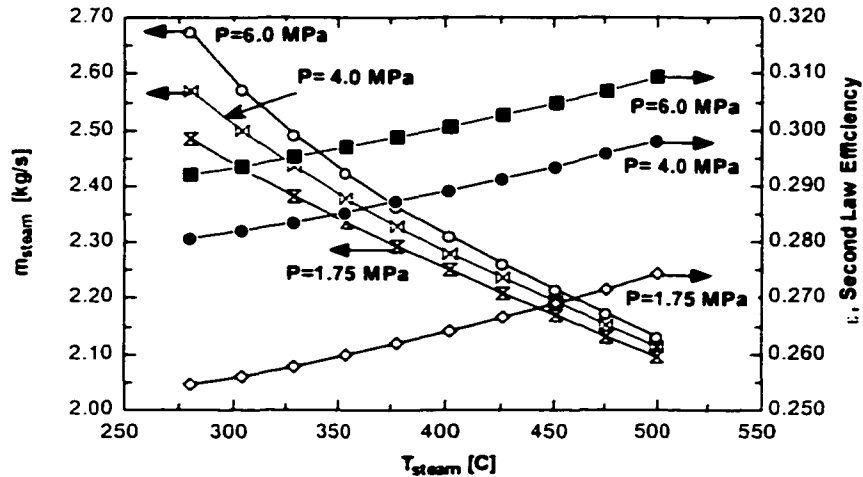
- *Increase in combustion efficiency* Higher temperature inside the furnace improves combustion of the bagasse. Ranaiah and Chikhalikar (1986) found that for a moisture content of 50 % the combustion efficiency varied from 93 to 95 %. With a moisture content between 40 and 30 %, this efficiency increases up to 98.5 %. Despite this, Kinoshita (1991) suggests that the effect on combustion efficiency has not been studied extensively and recommends further tests in order to establish reliable values.
- *Decrease in the flue gases flow* By reducing the amount of air required for the drier bagasse, the amount of flue gas is reduced as well. This is translated into a reduction of the power required for the fan to handle this air (Saechu and Soewarno, 1986; Correia Maranhao, 1994).
- *Increase of the thermal efficiency of the cycle and steam generated per kg of bagasse*  
As a consequence of the previously reported effects of bagasse drying, the thermal efficiency and the amount of steam generated per kg of bagasse burnt are increased. This is easily seen since less “waste” heat is being thrown away through the stack, and the fan power is reduced

- *Decrease in fuel consumption* As Arrascaeta and Fridman (1987) reported this may be reduced from 10 to 20 %. Also by looking at table 3.7 we can see that we will have more energy available per kg of bagasse. Consequently for the same amount of energy less bagasse is required as it is dried.

Simulations were carried out to compare the results from the algorithm developed in this research with the results from the drying simulations carried out by another author (Kilicaslan et al, 1999). Final moisture content of the bagasse was compared between the two calculations. Results were very similar, depending on what was considered. When neither the heating of the fibre nor the increase of the moisture content in the flue gases are considered the difference was about 2.5 %. When both were considered the difference was about 12 %. The size of the last difference is due to the fact that Kilicaslan neglected the energy required to heat up the fibre in his calculations.

### ***3.3.3 – Boilers - Results and Analysis***

Some brief calculations were done to determine the amount of steam generation that can be expected. Calculations were carried out using one kg/s of bagasse with 50% moisture and 2% ash content. For this moisture and ash content the bagasse has a LHV of 7,540 kJ/kg. The following figures show the rate of steam generated and the second law efficiency,  $\epsilon$ , under this situation, at different pressures and temperatures.



**Figure 3.19: Steam Generation and 2<sup>nd</sup> Law Efficiency vs Steam Temperature**

As can be seen in Figures 3.19 and 3.20 the mass flow rate of steam generated increases either when the pressure is increased or the temperature is reduced. Even though at times a low flow rate may be desirable to reduce pressure drop in the piping, certain flow rates are going to be needed by the factory for process steam. The point is that increasing the pressure and temperature may reduce the flow rate too much and jeopardize the steam supply to the factory. Comparatively the temperature has a more dramatic impact on the steam flow rate generated. The flow rate reduction can, of course, be overcome by using a larger boiler (and therefore more fuel).

On the figures we can also see how the second law efficiency,  $\epsilon$ , is increased when the temperature and the pressure are increased. This confirms the well-known principle that a higher temperature and a higher pressure permits a better utilization of the energy available in the fuel.

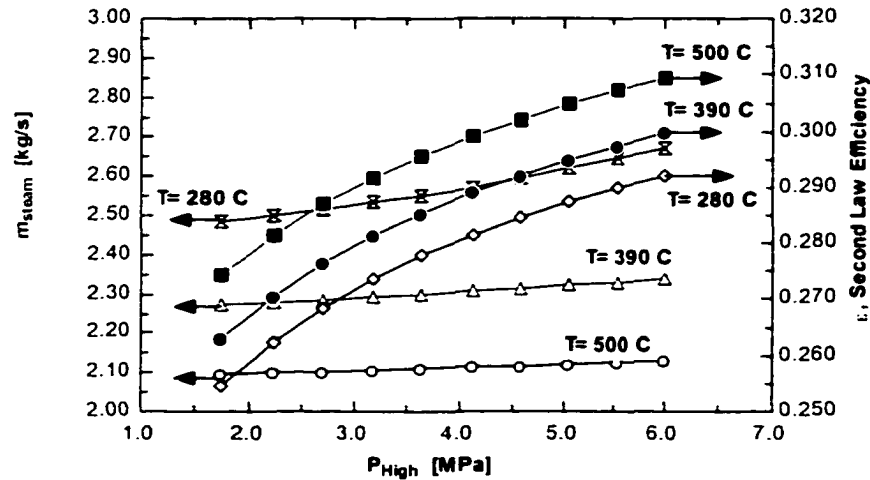


Figure 3.20: Steam Generation Rate and 2<sup>nd</sup> Law Efficiency vs Steam Pressure

### 3.4 – Co-generation Cycles to be Analyzed

In order to constrain the scope of possible simulations and focus on analyzing the recovery of the heat available in the flue gas, seven cycles were analyzed. The cycles chosen were

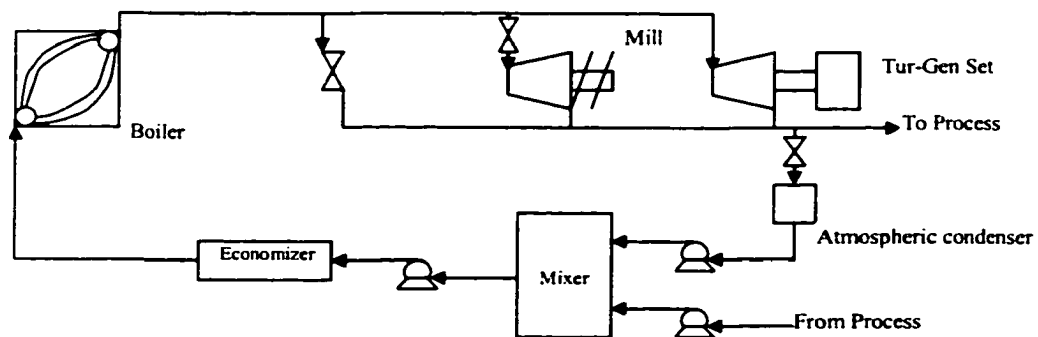
- Base Cycle
- Single Extraction Cycle
- Parallel Cycle
- Single Extraction Condensing Cycle
- Double Extraction Condensing Cycle
- Regenerative Cycle
- Reheat Cycle

All of these are described next using cycle diagrams. This selection of cycles will show the effect of back-pressure versus condensing turbines, single versus double extraction and the use of high pressure instead of medium pressure steam for milling.

### ***Base Cycle***

As a reference point a Base Cycle (Figure 3.21) was analyzed. This base cycle simulated milling in parallel with power generation at low pressure (2 Mpa), which is the common situation in the industry. This cycle was analyzed to get a feeling of where most sugar factories stand in terms of co-generation.

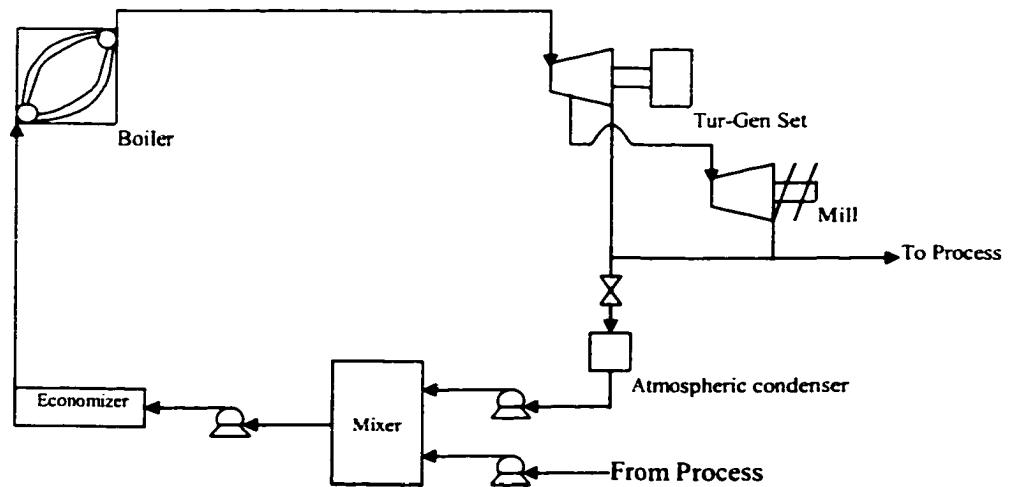
The steam leaves the turbines (mill and generator set) at the pressure required by process. The steam not needed for process is sent to an atmospheric condenser, and then combine with the condensed water from process at the mixer from where it is sent back to the boiler.



**Figure 3.21: Base Cycle**

### *Single Extraction Cycle*

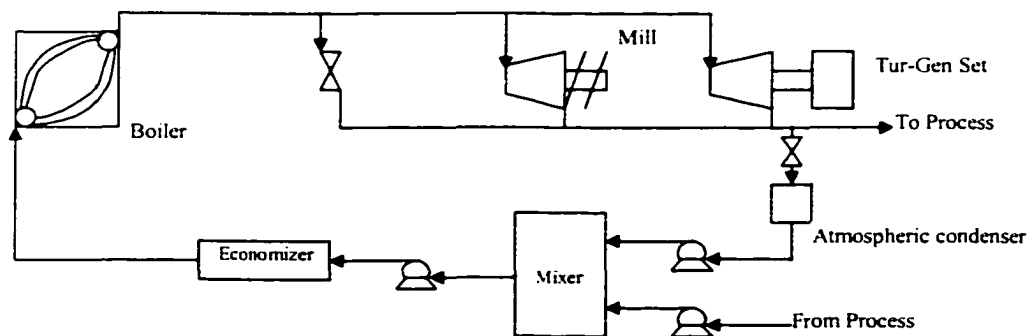
Figure 3.22 shows the Single Extraction Cycle that is also commonly used in the industry. Here one steam extraction in the turbine provides the steam necessary for milling. The rest of the steam is then expanded to the process pressure in the power-generating turbine-generator set. This cycle is found in factories that have conducted some improvements in their co-generation system mainly to become energy self-sufficient. The cycle uses an atmospheric condenser to condense the steam not needed for process. Later this condensed water plus the condensed water from process meet at the mixer, from which they are pumped to the boiler.



**Figure 3.22: Single Extraction Cycle**

## ***Parallel Cycle***

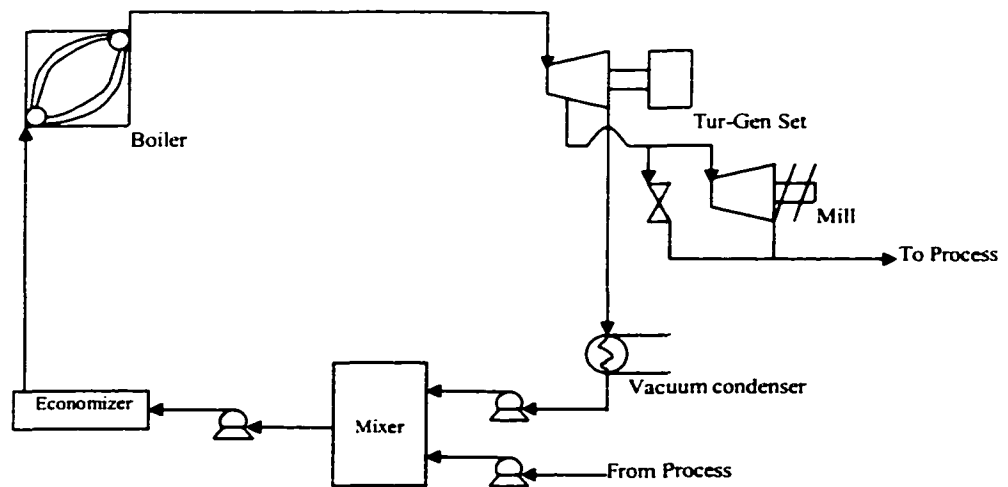
The so called Parallel Cycle (Figure 3.23) represents a factory where the milling is done in parallel with the power generation, but at higher pressure than the Base Cycle. Since this cycle was evaluated at 4.5 and 6 MPa, milling was simulated at these higher pressures. This design is expected to increase the efficiency of the steam usage, but no reference to this cycle was found in the literature. The reason is probably that it requires mill turbines to operate at much higher pressure than usual. To do this in a mill's upgrade is almost out of the question, but for a new mill it may be an option worth considering.



**Figure 3.23: Parallel Cycle**

### *Single Extraction Condensing Cycle*

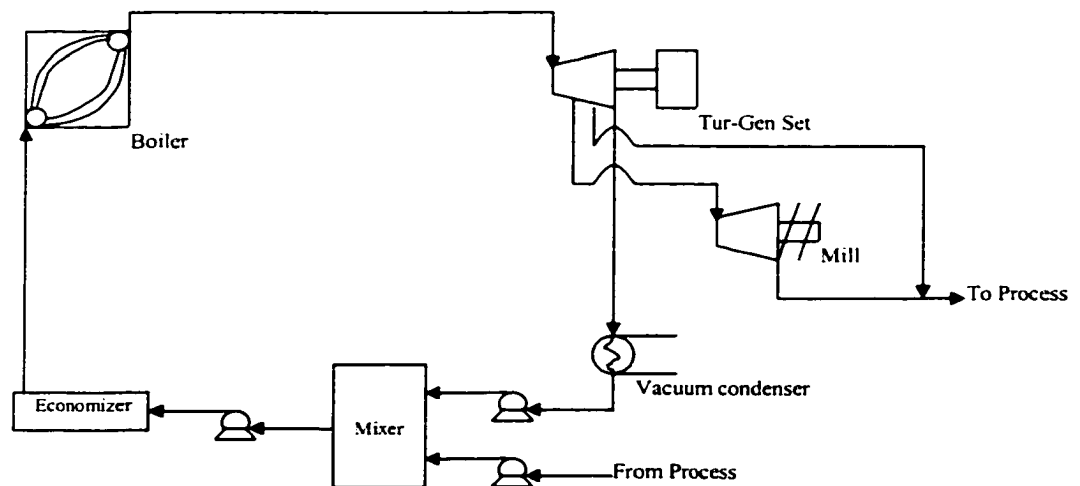
The Single Extraction Condensing Cycle (Figure 3.24) uses one extraction to get all the steam necessary for both milling and process. The rest of the steam is expanded in the power turbine to the condenser pressure. This cycle was analyzed to compare its performance with that of the single extraction cycle with an atmospheric condenser. This permitted a comparison between single extraction condensing and non-condensing cycles. A noticeable inefficiency of this cycle can be identified. Since only one extraction is used, the steam for milling and supplementary steam for process have to be extracted together. This means that the supplementary process steam needs to be expanded before meeting the steam leaving the mill. In this expansion process energy is conserved but substantial available energy is lost.



**Figure 3.24: Single Extraction Condensing Cycle**

### ***Double Extraction Condensing Cycle***

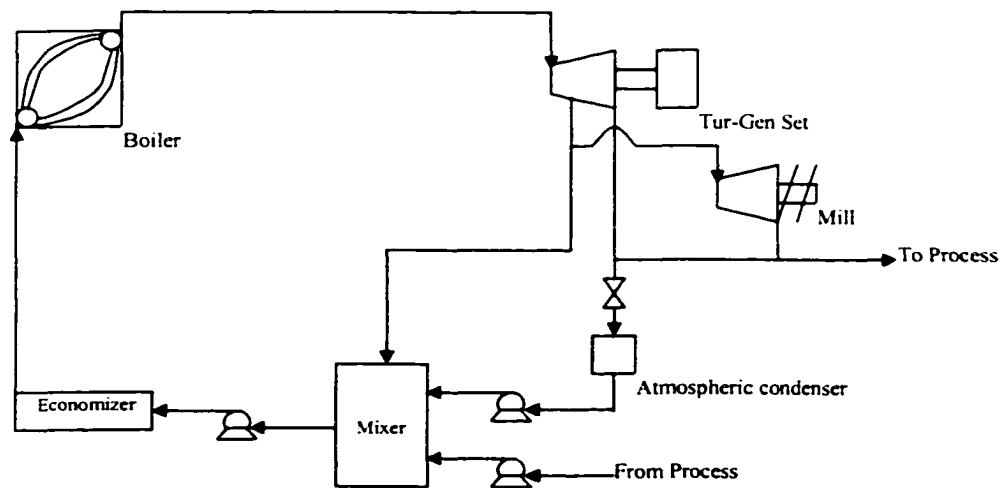
Figure 3.25 represents the Double Extraction Condensing Cycle. In the first extraction, the steam necessary for milling is obtained. The second extraction provides the supplementary steam necessary for process and the rest of the steam is expanded to below atmospheric pressure. After the condenser the steam is pumped to the mixer where it meets with the condensed water coming back from process. Technically this is believed to be the most efficient cycle used in this industry. Performance results for this cycle were compared with the other less sophisticated cycles.



**Figure 3.25: Double Extraction Condensing Cycle**

## *Regenerative Cycle*

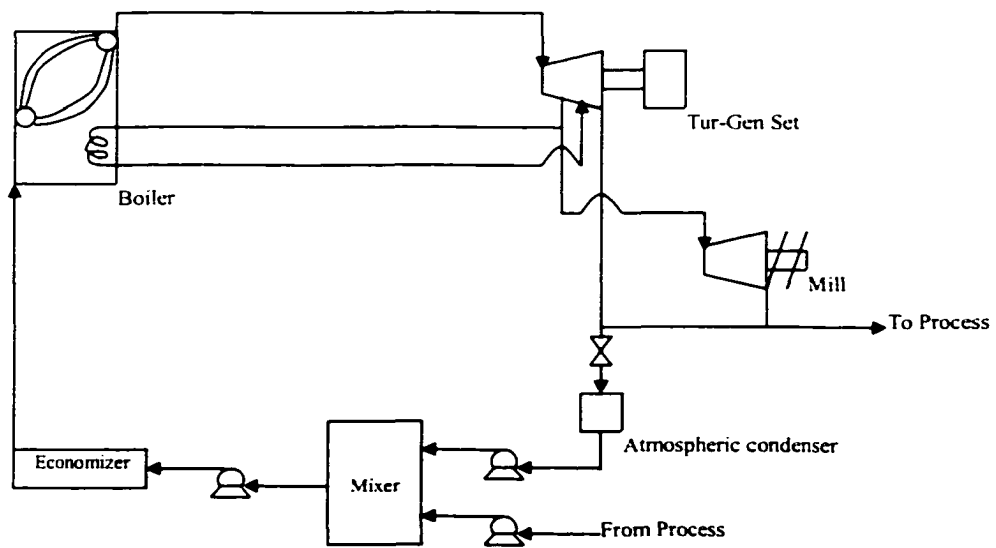
The Regenerative Cycle analyzed uses a single extraction turbine and an atmospheric condenser. On the extraction along with steam for milling, some extra steam is extracted to be used in the mixer, which acts as a feed water heater. This raises the temperature of the water before it enters the boiler, improving the efficiency of the cycle. Though this cycle is not common in the sugar industry its potential for increased efficiency made it one of the choices for the simulation.



**Figure 3.26: Regenerative Cycle**

## *Reheat Cycle*

The Reheat Cycle (Figure 3.27) also uses a single extraction turbine. In this extraction all the steam is extracted at the milling pressure and sent back to the boiler to be reheated, then it is send back to the turbine to continue its expansion to atmospheric pressure. This cycle is also uncommon in the industry, yet offers the potential of efficiency improvements.



**Figure 3.27: Reheat Rankine Cycle**

A possible reason why neither regeneration nor reheat cycles are found in the sugar industry, besides the cost, is that they can be of greatest benefit when high pressure steam and a condensing turbine are used. So far this situation has not been common. For example, the greatest advantage of the reheat Rankine cycle is that with it the moisture

content in the low-pressure stage of the turbine is reduced, preventing blade erosion. However, if steam is not expanded to low pressure, moisture is not a problem.

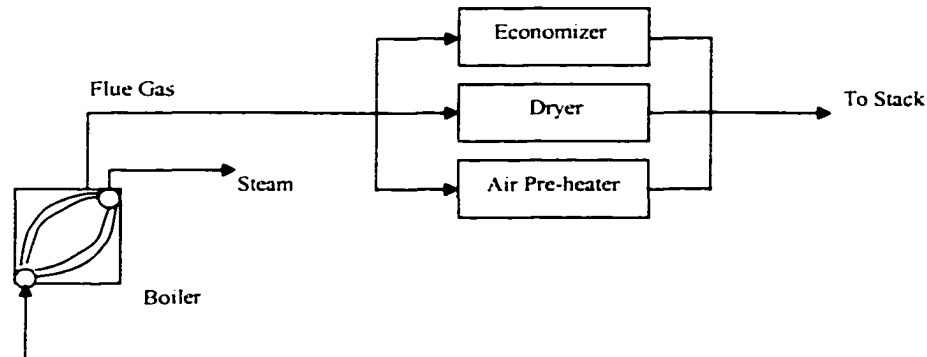
### ***3.4.1 – Simulation of Co-generation Cycles***

From the literature review operating characteristics of the equipment used for steam and power generation and milling have been determined. Given the quantity and state of the bagasse fed to the boiler, the temperatures of the feed-water and combustion air, and the temperature and pressure of the steam to be produced, the amount of steam produced from any of the cycles just described can be determined.

The temperature of the feed-water, the temperature of the combustion air and the state of the incoming bagasse are determined by the fraction of the flue gas that was diverted to the economizer, the air pre-heater, and the dryer respectively. A block diagram of the possible distribution of flue gas to the heat recovery components is shown in Figure 3.28. The flue gas had to be exhausted at a temperature higher than the dew point temperature to avoid condensation, this determines how much total energy can be recovered, since the flue gas exit temperature was set to be 25° higher than its dew point.

As discussed earlier, average numbers from a typical sugar mill were used to determine how much steam is needed for process, how much energy is needed for milling, and how much electricity is needed to run the factory. Subtracting these from the

corresponding sources in a given cycle then determine how much power is left for exporting.



**Figure 3.28: Flue Gas Distribution**

### **3.4.2 – Scenarios**

In order to do an analysis that offered answers to the initial questions of this research the scenarios evaluated had to be constrained by using variables of interest. The scenarios were set by the following variables:

- *Pressure-Temperature* To assess the effect of the steam generated pressure three pressures-temperature pairs in the boiler were considered. Low: 2 MPa (593.15 K), medium: 4.5 MPa (733.15 K) and high: 6 MPa (733.15 K). These Pressures-Temperatures (P-T) pairs were selected as being representative of the three common situations of the industry. The lowest P-T set represents the least advanced factory,

and the most common situation. This occurs when the factory is not in the power generation business and only generates for self-sufficiency. The mid level P-T pair represents a medium developed power generation facilities where some power export is done, probably to other industries nearby. The highest P-T pair represents a modern co-generation facility that makes power generation an important part of its business. This last scenario is most commonly found when the sugar mill has a power generation agreement with the local utility. Higher pressures seem to be not economically feasible under existing payment agreements between the mills and the utilities. Higher pressures require more expensive materials and increase the cost of power generation. This is especially important for mills generating power on a seasonal basis only.

- *Mill size* The simulations were carried out for a medium size mill, 217 Tonnes of cane per hour (5000 tonnes per day, 23 hours day). If the same component performances are assumed the results can be proportionally scaled down or up, to smaller or bigger mills.
- *Use of the flue gas* The flue gas generated was divided to supply heat to one, two, or all three heat recovery components. Different fractions that added to 100 % of flue gas utilization were examined. These fractions were varied in steps of 10 % each to add a total of 100 % heat recovery. For example, possible combinations are 100 % to economizer, 0 % to air pre-heater, and 0 % to dryer; 90 % to economizer, 10 % to air pre-heater, and 0 % to dryer; 80 % to economizer, 10 % to air pre-heater, and 10

% to dryer. In the simulations the combinations were varied until the energy was distributed 0 % to economizer, 0 % to air pre-heater, and 100 % to dryer. The options of no heat recovery (0 % to economizer, 0 % to air pre-heater, and 0 % to dryer), and an equal one third of heat recovery on each equipment (1/3 to economizer, 1/3 to air pre-heater, and 1/3 to dryer) were also evaluated. Table 3.10 summarizes the matrix of choices for the flue gas distribution.

**Table 3.10: Matrix of Choices for Flue Gas Distribution**

#	XE	XAP	XD	#	XE	XAP	XD	#	XE	XAP	XD
1	1	0	0	24	0.3	0.5	0.2	47	0.1	0.1	0.8
2	0.9	0.1	0	25	0.2	0.6	0.2	48	0.9	0	0.1
3	0.8	0.2	0	26	0.1	0.7	0.2	49	0.8	0	0.2
4	0.7	0.3	0	27	0.6	0.1	0.3	50	0.7	0	0.3
5	0.6	0.4	0	28	0.5	0.2	0.3	51	0.6	0	0.4
6	0.5	0.5	0	29	0.4	0.3	0.3	52	0.5	0	0.5
7	0.4	0.6	0	30	0.3	0.4	0.3	53	0.4	0	0.6
8	0.3	0.7	0	31	0.2	0.5	0.3	54	0.3	0	0.7
9	0.2	0.8	0	32	0.1	0.6	0.3	55	0.2	0	0.8
10	0.1	0.9	0	33	0.5	0.1	0.4	56	0.1	0	0.9
11	0	1	0	34	0.4	0.2	0.4	57	0	0	1
12	0.8	0.1	0.1	35	0.3	0.3	0.4	58	0	0.9	0.1
13	0.7	0.2	0.1	36	0.2	0.4	0.4	59	0	0.8	0.2
14	0.6	0.3	0.1	37	0.1	0.5	0.4	60	0	0.7	0.3
15	0.5	0.4	0.1	38	0.4	0.1	0.5	61	0	0.6	0.4
16	0.4	0.5	0.1	39	0.3	0.2	0.5	62	0	0.5	0.5
17	0.3	0.6	0.1	40	0.2	0.3	0.5	63	0	0.4	0.6
18	0.2	0.7	0.1	41	0.1	0.4	0.5	64	0	0.3	0.7
19	0.1	0.8	0.1	42	0.3	0.1	0.6	65	0	0.2	0.8
20	0.7	0.1	0.2	43	0.2	0.2	0.6	66	0	0.1	0.9
21	0.6	0.2	0.2	44	0.1	0.3	0.6	67	0	0	0
22	0.5	0.3	0.2	45	0.2	0.1	0.7	68	1/3	1/3	1/3
23	0.4	0.4	0.2	46	0.1	0.2	0.7				

### ***3.4.3 – Simulation Restrictions***

In order to guarantee a logical simulation and to keep different parameters within normal levels some restrictions were applied. Any scenario not meeting these restrictions was discarded. Specifically the constraints were as follows:

- *Adiabatic flame temperature* In order to avoid the melting of the ash over the heat transfer surfaces inside the boiler the adiabatic flame temperature was kept under 1600 °C (1873.15 K). This is accomplished by discarding scenarios where combustion conditions allowed this to happen.
- *Temperature of the fresh air leaving the air pre-heater* The temperature of the air after it passes the air pre-heater was not allowed to be higher than the temperature of the flue gas entering the air pre-heater minus 25 °C. This is to avoid the over-sizing of the heat exchanger that may happen if we expect both temperatures to be almost the same.
- *Temperature of the water leaving the economizer* This had to be at least 20 °C lower than the saturation temperature at the working pressure, to avoid steam generation inside the economizer. And, for the same reason explained above, at least 25 °C lower than the temperature of the flue gas entering the economizer.

- *Temperature of the flue gas leaving the economizer* The flue gas leaving the economizer had to be at a higher temperature than the water entering the economizer. Otherwise will be a violation of the laws of thermodynamics.
- *Temperature of the flue gas leaving the air pre-heater, dryer, and economizer* This had to be 25 °C higher than the dew point temperature of the flue gas to avoid condensation of acids in the surfaces of the heat recovery equipment and stack.
- *Moisture content of bagasse after the dryer* This was not allowed to be less than 20 % to avoid the possibility of auto ignition.

### ***3.5 – Economic Analysis Fundamentals***

This section describes the economic analysis tools that were included in the software developed. A complete financial and economic analysis requires a great deal of information about the local economy, which is not available. This includes investors expectations, local price for the power generated, etc. The economic tools that were included in this research were oriented to calculate only the cost of the power generated based on the capital cost of the equipment. This can be accomplished by calculating the levelized cost of the energy, which uses the annual energy production and the different annual costs, including the capital cost and salvage cost. Some systems were considered to exemplify the methodology and can be found in Appendix A.

### 3.5.1 – Levelized Cost of the Energy

The cost of the energy could be calculated in several ways. A simple one is explained following. This methodology calculates the annual worth of all the costs of the plant.

Calculating the annual worth of both all the costs and energy production is a simple and easy way to calculate the levelized cost of the energy.

$$COE = \frac{CC * CRF + O \& MC + FC + SC * CRF2}{AnnualEnergyProduction} \quad (3.50)$$

Where:  $CRF$  = Capital Recovery Cost Factor

$$CRF = \frac{i * (1 + i)^n}{(1 + i)^n - 1}$$

$$CRF2 = \frac{i}{(1 + i)^n - 1}$$

$i$  = Annual interest rate

$n$  = Project life [years]

$CC$  = Capital Cost [\$]

$O\&MC$  = Annual Operations and Maintenance Cost [\$/year]

$FC$  = Annual Fuel Cost [\$/year]

$SC$  = Salvage Cost [\$]

This methodology is commonly used in the utility and other power generation industries, so it is believed that it will be appropriate for co-generation analysis.

The Annual Energy Production, or Annual Energy Output, or Total Power, is calculated by multiplying the generation capacity, times the number of days of generation, times the number of hours per day of generation, for both the crushing and non-crushing season. Equation 3.51 was used for this calculation:

$$AEO = CS\_R * CS\_hrs * CS\_days + NCS\_R * NCS\_hrs * MCS\_days \quad (3.51)$$

Where:  $AEO$  = Annual Energy Output or Total Power [kWh/year]

$CS\_R$  = Generation Capacity, rating, crushing season [kW]

$CS\_hrs$  = Hours/day of power generation, crushing season [hrs/day]

$CS\_days$  = Days of power generation, crushing season [days]

$NCS\_R$  = Generation Capacity, rating, no-crushing season [kW]

$NCS\_hrs$  = Hours/day of power generation, no-crushing season [hrs/day]

$NCS\_days$  = Days of power generation, no-crushing season [days]

### 3.5.2 – Payback Period and Internal Rate of Return Analysis

Simple and discounted payback periods, and internal rate of return calculations were suggested to compare the different options evaluated. Even though these calculations are done in a very simplified way, assuming constant cash flows, they are still useful as comparison parameters. Equations (3.51) to (3.53) were used in the analysis.

$$\text{Simple Payback Period : } \sum_{t=0}^N At = 0 \quad (3.51)$$

$$\text{Discounted Payback Period : } \sum_{t=0}^N At * (1 + i)^{-t} = 0 \quad (3.52)$$

$$\text{Internal Rate of Return : } \sum_{t=0}^n At * (1 + I)^{-t} = 0 \quad (3.53)$$

Where:  $N$  = Payback Period [years]

$n$  = Project life or study period [years]

$At$  = Annual cash flow, year 0 must include capital cost [\$/year]

$i$  = Annual interest rate [%/100]

$I$  = Internal Rate of Return [%/100]

As mentioned above some examples of these calculations can be found in Appendix A.

## *Chapter 4*

# **Simulations Results**

This chapter begins with a description of the final version of the software produced during this dissertation, BIOFIRE<sup>1</sup>, as an introduction to the quantitative results of this project. Included are the results obtained during the simulations and an analysis of them. The results: total power generated, plant efficiency, and irreversibility of the heat recovery components, help to clarify some questions that researchers have had for some time about the benefits of bagasse drying versus the use of other heat recovery equipment. They also indicate the impact of the different cycles on power generation.

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<sup>1</sup> A copy of the Algorithm in DELFHI can be requested from the Mechanical Engineering Department at Colorado State University

## ***4.1 – BIOFIRE – The software***

BIOFIRE is the final software created during this research. It is divided in 3 different windows that are accessible through the tabs on the upper left side of the screen (See Figure 4.1). They divide BIOFIRE into the Power Generation, Economics, and Flue Gas Analysis pages.

### ***4.1.1 – BIOFIRE – Power Generation***

Figure 4.1 shows the Power Generation window of BIOFIRE. As can be seen it is divided in two sections, the Inputs, and the Outputs (called Results in the window). The Inputs section contains different fields for data input and options selection. The data input requires specific information about:

- Bagasse Characteristics
- Moisture Content and Temperature Limits
- Equipment Performance, and,
- Mill Characteristics

The options selection require information about

- The Pressure-Temperature pair
- Cycles to Evaluate, and,
- Variables to Maximaze

The Input fields are described in more detail in section 3.2 of Chapter 3, and a theoretical background is given in chapter 2. On Figure 4.1 can be seen the default data. These data represents the average bagasse composition and typical medium size mill. These average/typical inputs are:

- Bagasse Composition (Dry basis): Carbon = 44%, Hydrogen = 6%, Oxygen = 48%
- Moisture Content = 50%
- Limit Moisture Content after the dryer = 20%
- Maximum Temperature inside the Boiler = 1873.15 K
- Boiler Efficiency = 70%
- Burn Bagasse efficiency = 97.5%
- Radiation Efficiency = 95%
- Economizer Efficiency = 95%
- Air Pre-Heater Efficiency = 95%
- Dryer Efficiency = 95%
- Turbine Efficiency = 75%
- Generator Efficiency = 95%
- Pumps Efficiency = 90%
- Mill – Turbine Efficiency = 75%
- Mill Capacity = 217 Tonnes of cane per hour
- Steam for Milling = 15 kWh per tonne of crushed cane
- Steam for Process = 420 kg of steam per tonne of crushed cane
- Power Requirement = 15 kWh per tonne of crushed cane

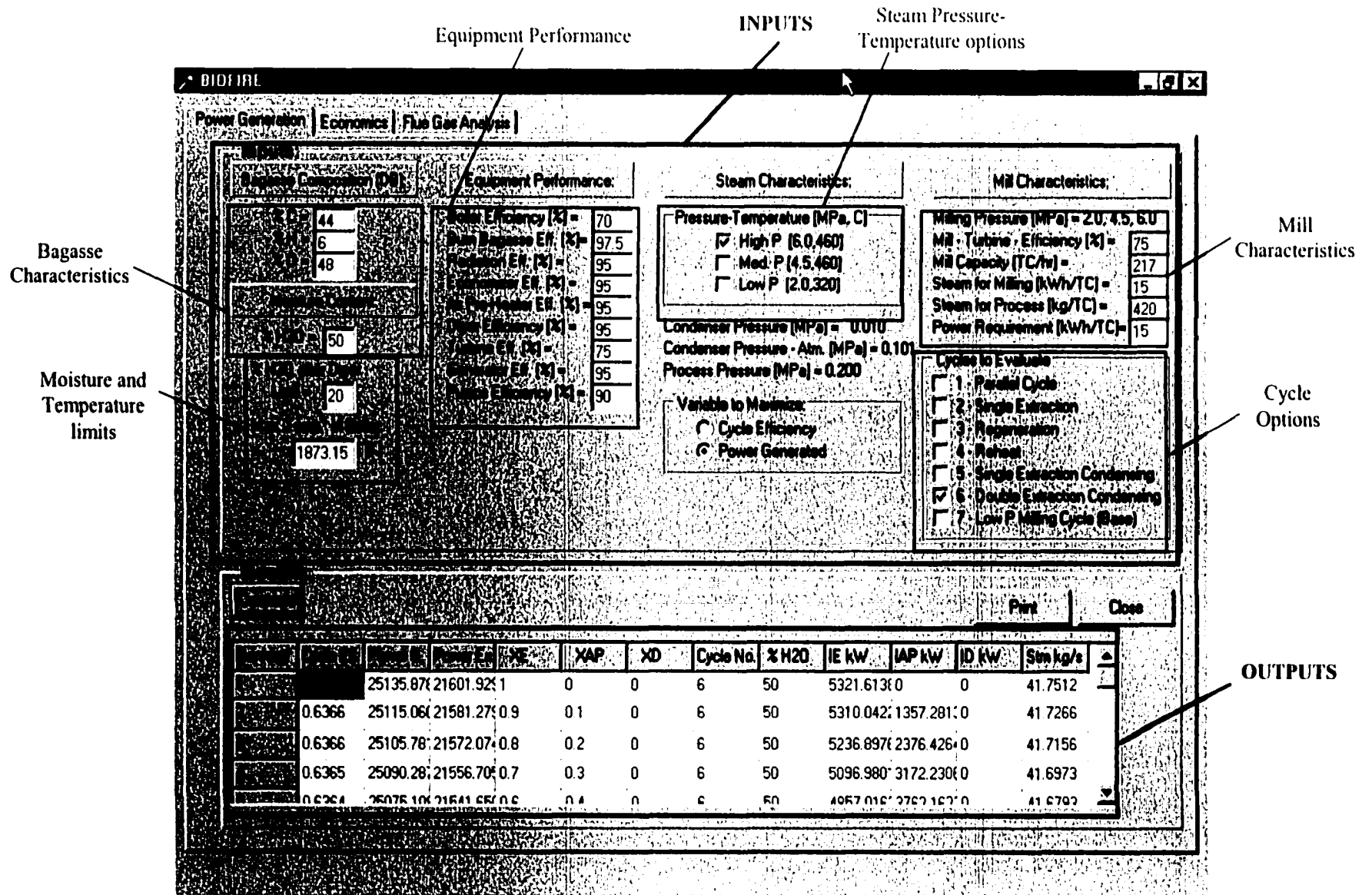


Figure 4.1: BIOFIRE

The Results section of this window show the different fields or variables evaluated during the simulations. These variables are described later in the Simulations Results section of this chapter. Figure 4.1 shows the first 4 best results from simulating a Double Extraction Condensing cycle at high pressure (6.0 MPa).

BIOFIRE has been enabled with a print button and creates a 1 page report containing the Input data and the first 48 best options (see insert on following page). The best options are organized by ranking the results from the most efficient cycle, or the cycle that generates the most power (depending of what was selected in the Inputs), to the least efficient. This ranking is done by evaluating each case in which the flue gas distribution was varied. Those options that yielded a result that exceeded any of the simulation restrictions (see Section 3.4.3) are discarded and zeros are shown in the print out. The flue gas energy distribution was varied in steps of 10 % for each heat recovery option. The optimization was not done with smaller steps because results indicated that the variation between consecutive options for 10 % steps was sufficiently small.

#### ***4.1.2 – BIOFIRE – Economics***

In order to enable the user to do a quick project analysis some very basic tools are provided in this page. It is important to notice that a complete economic analysis will include a considerable greater number of variables more than the ones included in this application, but these will allow some simple analyses.

BIOFIRE - Biomass Combustion Systems

Prepared by Jean V. Seguro as part of the dissertation work at Colorado State University

Bagasse Composition (Dry Basis)    %C= 44    %H= 5    %O= 48    Initial Moisture Content (%) = 50  
 Moisture Lower Limit= 20    Boiler Efficiency= 70    Burnt Bagasse Efficiency=97.5    Radiation Efficiency=95  
 Economizer Efficiency=95    Air Pre-Heater Efficiency=95    Dryer Efficiency=95    Turbine Efficiency=75  
 Generator Efficiency=95    Pump Efficiency=90    Mill Efficiency=75    Mill Capacity=217  
 Steam for Milling=15    Steam for Process=420    Power Requirement=15  
 PHigh= 6.0 MPa    THigh= 733.15 K

Ranking	Cycle Eff.	Power G.	Power Ex.	XE	XAP	XD	Cycle No.	% H2O	IE kW	IAP kW	IO kW	Strm kg/s
1	0.6366	25135.8789	21601.9297	1	0	0	5	50	5321.6138	0	0	41.7512
2	0.6366	25115.0605	21581.2793	0.9	0.1	0	5	50	5310.0422	1357.2913	0	41.7255
3	0.6366	25105.7812	21572.0742	0.8	0.2	0	5	50	5298.3975	2375.4254	0	41.7056
4	0.6365	25090.2871	21556.7051	0.7	0.3	0	5	50	5096.9801	3172.2306	0	41.6973
5	0.6364	25075.1094	21541.6504	0.6	0.4	0	5	50	4857.0162	3752.1622	0	41.6793
6	0.6362	25058.1016	21524.7793	0.5	0.5	0	5	50	4519.1251	4172.0471	0	41.6592
7	0.6361	25038.7734	21505.6074	0.4	0.6	0	5	50	4049.425	4420.3574	0	41.6364
8	0.636	25022.6562	21489.6191	0.3	0.7	0	5	50	3408.6068	4499.7694	0	41.6173
9	0.636	25018.918	21485.9121	0.5	0.1	0.4	6	30.113	5330.7205	1694.0837	7653.0478	41.6129
10	0.6358	25003.5469	21470.6641	0.6	0	0.4	5	31.1564	5708.209	0	7256.8771	41.5947
11	0.6358	25000.8711	21468.0098	0.2	0.8	0	5	50	2562.2888	4435.791	0	41.5915
12	0.6356	24975.2109	21442.5556	0.4	0.2	0.4	5	29.384	4758.0533	2927.7623	8030.1622	41.5612
13	0.6354	24945.2129	21412.8008	0.3333	0.3333	0.3333	5	33.5687	4127.0613	3953.7533	6717.366	41.5257
14	0.635	24882.5719	21350.7637	0.5	0.1	0.3	5	38.2567	5334.2266	1574.3757	5192.6384	41.4517
15	0.6349	24875.4746	21344.6152	0.5	0.2	0.3	5	37.5704	4965.1048	2742.1425	5447.2084	41.4444
16	0.6349	24873.791	21341.9531	0.7	0	0.3	5	38.7989	5562.7268	0	4955.2584	41.4412
17	0.6349	24871.2285	21339.4121	0.4	0.3	0.3	5	36.9574	4458.8018	3630.539	5727.2469	41.4382
18	0.6349	24870.4414	21338.6309	0.2	0.5	0.3	5	35.3987	2841.2371	4709.3596	6383.4834	41.4373
19	0.6349	24869.041	21337.2422	0.3	0.4	0.3	5	36.1817	3759.2559	4275.9356	6039.4756	41.4356
20	0.6347	24841.1738	21309.5996	0.7	0.1	0.2	5	43.6091	5322.7362	1480.5793	3164.0886	41.4027
21	0.6344	24801.127	21259.875	0.4	0.4	0.2	5	42.5615	4245.3205	4052.3706	3640.1702	41.3553
22	0.6343	24790.7441	21259.5762	0.5	0.3	0.2	5	43.2411	4732.5532	3433.017	3475.0547	41.343
23	0.6343	24786.2129	21255.082	0.1	0.7	0.2	5	41.1322	1531.9279	4808.6835	4267.7485	41.3377
24	0.6337	24711.502	21180.9727	0.6	0.2	0.2	5	43.864	5022.8655	2571.085	3278.8798	41.2493
25	0.6336	24691.248	21160.8809	0.3	0.5	0.2	5	42.811	3541.1536	4469.7912	3800.5772	41.2253
26	0.6336	24691.0293	21160.6641	0.2	0.6	0.2	5	42.3019	2668.0645	4712.6183	4004.1631	41.2251
27	0.6335	24686.0488	21155.7246	0.8	0	0.2	5	44.9098	5374.269	0	2995.6219	41.2192
28	0.6335	24677.5332	21147.2773	0	0.8	0.2	5	41.2551	0	4706.8203	4475.3481	41.2091
29	0.6322	24506.8633	20977.9824	0.9	0	0.1	5	49.1527	5112.3851	0	1313.9505	41.0673
30	0.6321	24490.7324	20961.9824	0.8	0.1	0.1	5	49.0496	5063.5797	1375.4384	1375.3777	40.9882
31	0.6321	24486.3359	20957.6211	0.7	0.2	0.1	5	48.9106	4920.4931	2408.4325	1437.3222	40.983
32	0.632	24477.1855	20948.543	0.6	0.3	0.1	5	48.7923	4707.9436	3213.7585	1506.3542	40.9722
33	0.632	24469.8496	20941.2676	0.5	0.4	0.1	5	48.6482	4390.3318	3817.7718	1580.9309	40.9635
34	0.6319	24459.334	20930.8359	0.4	0.5	0.1	5	48.5099	3950.3859	4236.6652	1664.7503	40.9511
35	0.6318	24449.6387	20921.2198	0.3	0.6	0.1	5	48.3548	3333.0442	4483.714	1750.876	40.9396
36	0.6317	24438.3613	20910.0332	0.2	0.7	0.1	5	48.1855	2507.0424	4577.1888	1844.2392	40.9263
37	0.6317	24432.252	20903.9727	0.1	0.8	0.1	5	47.9817	1425.8348	4501.4184	1953.1255	40.919
38	0.5781	17020.3066	13551.7432	0	0	0	6	50	0	0	0	32.153
39	0	0	0	0	0	0	0	0	0	0	0	0
40	0	0	0	0	0	0	0	0	0	0	0	0
41	0	0	0	0	0	0	0	0	0	0	0	0
42	0	0	0	0	0	0	0	0	0	0	0	0
43	0	0	0	0	0	0	0	0	0	0	0	0
44	0	0	0	0	0	0	0	0	0	0	0	0
45	0	0	0	0	0	0	0	0	0	0	0	0
46	0	0	0	0	0	0	0	0	0	0	0	0
47	0	0	0	0	0	0	0	0	0	0	0	0
48	0	0	0	0	0	0	0	0	0	0	0	0
49	0	0	0	0	0	0	0	0	0	0	0	0
50	0	0	0	0	0	0	0	0	0	0	0	0

Figure 4.2 shows the “Economics” window of BIOFIRE. Five small calculators are included in this page, identified as:

- Annual Energy Output
- Cost of Energy
- Simple Payback Period
- Discounted Payback Period, and,
- Internal Rate of Return

The *Annual Energy Output* calculator enables the user to estimate the *Total Power Generated* per year, considering crushing and non-crushing seasons. The *Cost of Energy* calculator estimates the cost per kWh generated as defined in section 3.6.1 of chapter 3. This can give an idea of the cost per kWh generated that can be used to compare with the rates that are being paid per kWh generated. On the right side of the screen we see three simple project evaluation calculators. These are simple analysis tools, *Simple Payback*, *Discounted Payback*, and, *Internal Rate of Return*, that assume an initial capital cost and a constant cash flow through the life of the project (the equations used can be seen in section 3.6.2 of chapter 3). A more complete analysis would consider local parameters, like inflation, not constant cash flows, salvage costs, etc.

After the user has obtained the technical performance of the scenarios to evaluate, the cost of equipment have to be ordered and used as input in these calculators.

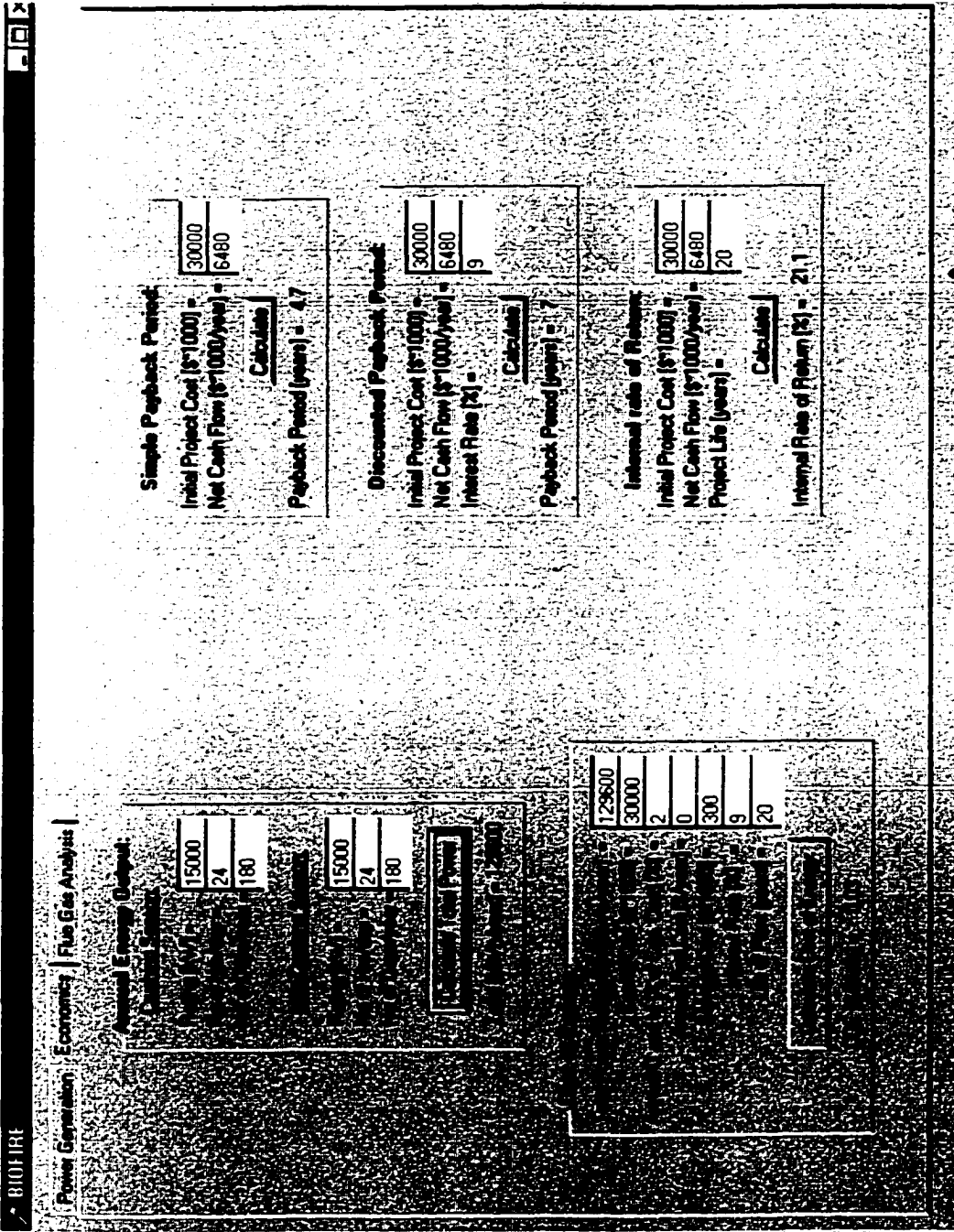


Figure 4.2: BIOFIRE - Economics

### ***4.1.3 – BIOFIRE – Flue Gas Analysis***

This window (See Figure 4.3) enables the user to estimate the parts per millions (molar and mass based) of CO, CO<sub>2</sub>, H<sub>2</sub>O, H<sub>2</sub>, and N<sub>2</sub>, which are the species considered to be present on the flue gas. Similar to the Power Generation page, it is divided in two sections, the inputs and results. The inputs are bagasse composition, moisture content, air temperature, furnace performance, and, bagasse (fuel) flow rate. The results given are the parts per million (ppm) of each specie on a molar and mass base, the flue gas flow rate, and the adiabatic flame temperature of the combustion gas.

### ***4.2 – Usefulness of BIOFIRE***

BIOFIRE allows the user to calculate the performance of a co-generation plant that employs bagasse as fuel. Furthermore, it considers all the possible options of heat recovery to maximize plant performance. This enables the user to determine which heat recovery system to choose without having to make multiple calculations and evaluations. Consider that for each click on the “Calculate” button BIOFIRE can evaluate as many as 1156 different scenarios. The output will show the power generation and efficiency of each cycle for each heat recovery combination. By inspecting the outputs the user can determine which will be the best, the second best, and the worst options for heat recovery. In addition to this, the irreversibility generated in the heat recovery components

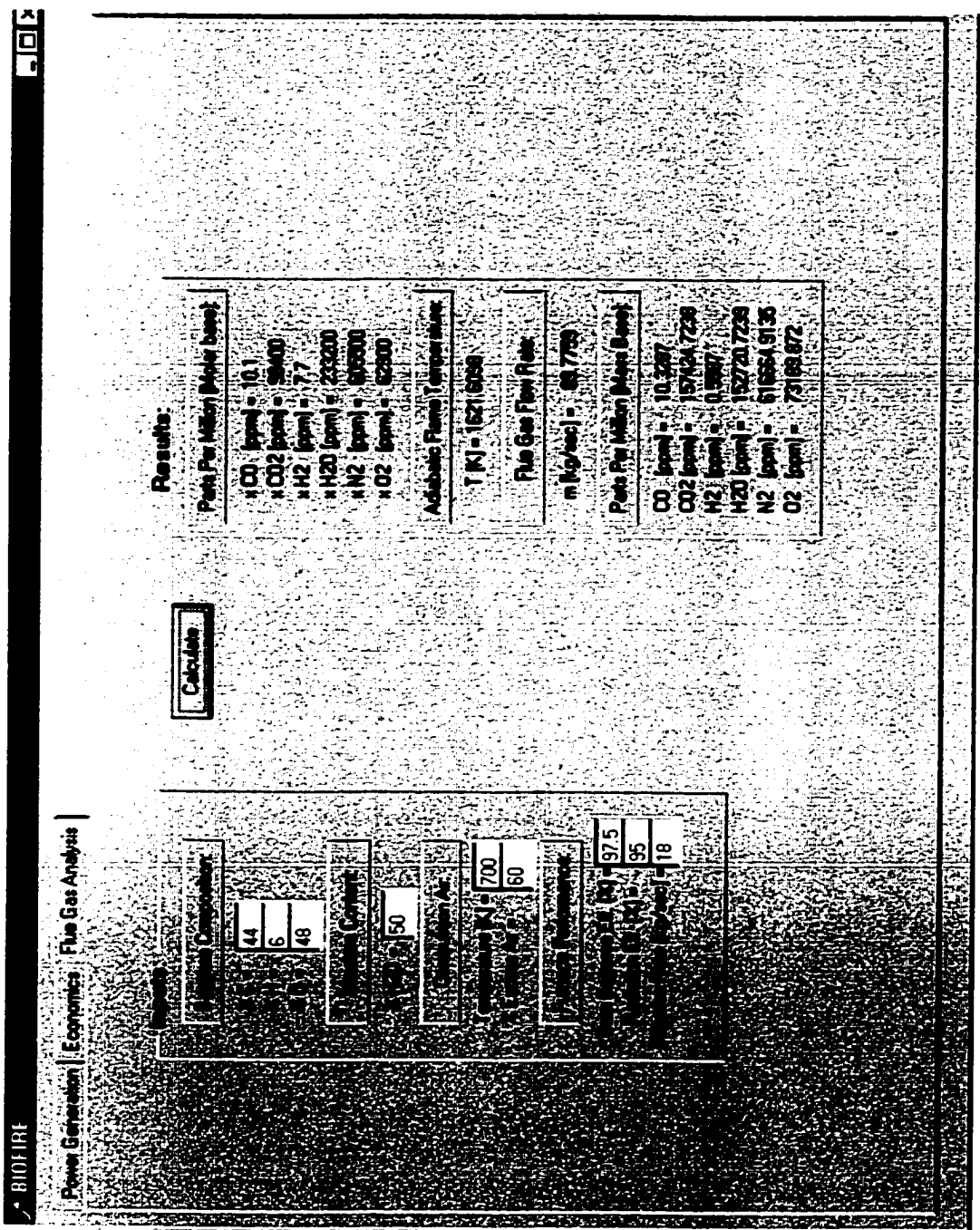


Figure 4.3: Biofire -- Flue Gas Analysis

can be examined to determine where the work is lost, and to compare the heat recovery options on this basis.

Even though it was not possible to create an algorithm that included the cost of the components evaluated, the tools needed to make cost estimates are included in BIOFIRE. After the user does the technical analysis and determine the cycles/scenarios to evaluate s/he can order the particular equipment costs and do the economic analysis using the tools provided with BIOFIRE. Appendix A shows an example analysis that where these tools were used.

### ***4.3 – Cycle Simulations (Technical Overview)***

The following pages contain tables and figures summarizing the results of simulating the different cycles under different scenarios. In this section no economic analysis is done, only the technical aspects are considered. Seven different cycles at three different pressures with 68 different combinations for the distribution of the flue gases were evaluated. This totals 1156 different cycles.

#### ***4.3.1 – Simulations Results***

The following table and figures display the total power generated, and the irreversibility for each heat recovery component for 3 flues gas distribution scenarios. No

flue gas usage ( $x= 0$ ), 1/3 of the flue gas diverted to each component ( $x= 1/3$ ), and the flue gas distribution that yields the maximum power generation for each cycle ( $x= \text{Max}$ ). Please notice that these values are for a 217 tonnes per hour Mill, which was assumed as typical in Section 3.3.8. The variables evaluated during the simulations are:

$$\eta_{\text{Cycle}} = \frac{\text{PowerG} + \text{PowerMill} + Q_{\text{process}}}{\text{mbag}(50\%H_2O) * \text{LHV}(50\%H_2O)} \quad (4.1)$$

**$\eta_{\text{Cycle}}$**  : Overall plant efficiency, defined as energy for process plus energy for milling plus power generated over energy content on the bagasse as delivered.

**$\text{Power G.}$** : Total power generated [kW]

**$\text{Power Ex.}$** : Exportable power [kW]

**$\text{XE}$** : Fraction of the flue gases send to the economizer

**$\text{XAP}$** : Fraction of the flue gases send to the air pre-heater

**$\text{XD}$** : Fraction of the flue gases send to the dryer

**$\% H_2O$** : Percentage of water in the bagasse (by weight) [%]

**$\text{IE}$** : Irreversibility generated in the economizer [kW]

**$\text{IAP}$** : Irreversibility generated in the air pre-heater [kW]

**$\text{ID}$** : Irreversibility generated in the dryer [kW]

**$\text{Stm}$** : Flow mass of steam [kg/sec]

These variables were chosen because they show how the different heat recovery combinations affect the performance of the plant. Not all of them are shown in the

following table and figures, but an example of the complete results can be seen on the print out sample presented in Section 4.1.1.

Table 4.1 summarizes the scenarios evaluated: 3 Pressure-Temperature pairs, 68 flue gas distribution combinations, and 7 different cycles. A more complete description of these scenarios can be found in Section 3.5.2 of Chapter 3.

**Table 4.1: Scenarios Evaluated**

<b>Pressure-Temperature Pair</b>	<b>Cycles:</b>	<b>Heat Recovery Combinations:</b>
2.0 MPa/320 C	- Parallel - Single Extraction Condensing (acts as a simple condensing cycle) - Double Extraction Condensing (acts as a single extraction cycle)	68
4.5 MPa/460 C	- Parallel - Single Extraction - Regenerative Cycle - Reheat Cycle - Single Extraction Condensing - Double Extraction Condensing - Base Cycle	68
6.0 MPa/460 C	- Parallel - Single Extraction - Regenerative Cycle - Reheat Cycle - Single Extraction Condensing - Double Extraction Condensing - Base Cycle	68

On the figures presented next the cycles were identified by number and not by name. To help identify them please see Table 4.2.

**Table 4.2: Cycle's Names**

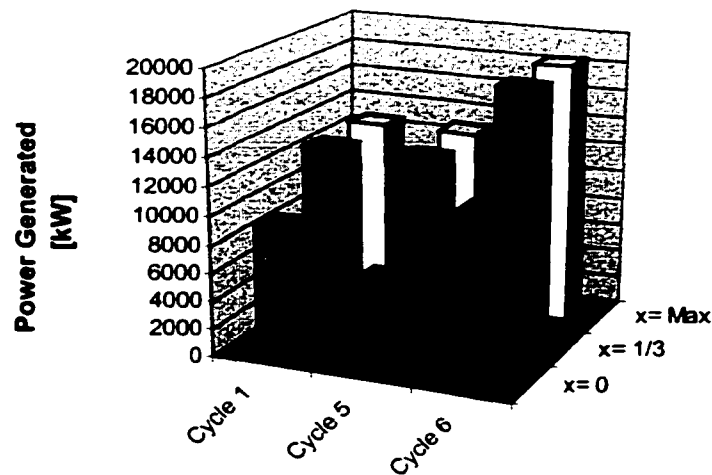
<b>Cycle Number:</b>	<b>Cycle's Name:</b>
1	Parallel Cycle
2	Single Extraction Cycle
3	Regenerative Cycle
4	Reheat Cycle
5	Single Extraction Condensing Cycle
6	Double Extraction Condensing Cycle
7	Base Cycle

In Table 4.3 some results from the simulations done are presented. Please remember that the flue gas distribution is represented by the fraction of flue gas going to the economizer, XE, to the air pre-heater, XAP, and to the dryer, XD. At the lowest pressure evaluated (2.0 MPa) only the Parallel Cycle, Single Extraction Condensing Cycle, and Double Extraction Condensing Cycle (Cycles 1, 5, and 6) were evaluated. Since the pressure of steam generation in this case is the same as the pressure used for milling (2.0 MPa) the Base Cycle (Cycle 7) and the Single Extraction Cycle (Cycle 2) are the same as the Parallel Cycle (Cycle 1). Similarly at 2.0 MPa the Regeneration and Reheat cycles were not evaluated since the extraction pressure used was the milling pressure too (also 2.0 MPa). Also important is to notice that on the 2.0 MPa scenario the Single Extraction Condensing Cycle uses a condensing turbine with no extraction since the steam necessary for milling and process is diverted before it enters the turbine. Similarly, the Double Extraction Condensing Cycle uses, in fact, a single extraction condensing turbine. Here the steam for milling is diverted before it enters the turbine and the steam for process is extracted from the turbine at 200 kPa. The values reported are for a sugar cane feeding rate of 217 Tonnes per hour.

**Table 4.3: Summary of Cycle's Performance for a 217 Tonnes/hr Mill**

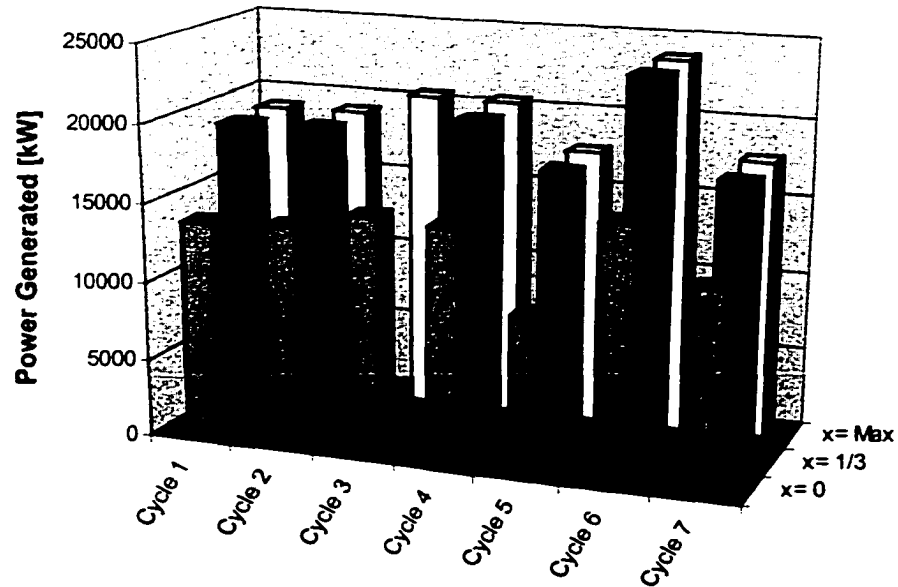
Pressure [Mpa]:	Cycle:	XE:	XAP:	XD:	Cycle Eff.	Power G. [kW]:	IE [kW]:	IAP [kW]:	ID [kW]:
2	1	0.7	0.3	0	0.5447	12871	4535	3171	0
		0.333333	0.333333	0.333333	0.5442	12802	3711	3954	6717
		0	0	0	0.5171	9051	0	0	0
	5	0.8	0.2	0	0.545	12905	5622	2376	0
		0.333333	0.333333	0.333333	0.544	12772	4287	3954	6717
		0	0	0	0.4966	6223	0	0	0
	6	0.8	0.2	0	0.5848	18405	5621	2376	0
		0.333333	0.333333	0.333333	0.5838	18273	4287	3954	6717
		0	0	0	0.5364	11723	0	0	0
4.5	1	1	0	0	0.6055	19067	4388	0	0
		0.333333	0.333333	0.333333	0.6047	18948	3610	3954	6717
		0	0	0	0.567	13745	0	0	0
	2	1	0	0	0.6036	19198	4389	0	0
		0.333333	0.333333	0.333333	0.6027	19079	3610	3954	6717
		0	0	0	0.565	13875	0	0	0
	3	0.2	0.8	0	0.6135	20575	781.64	4437	0
		0	0	0	0.5731	14989	0	0	0
	4	0.9	0.1	0	0.6125	20438	4270	1357	0
		0.333333	0.333333	0.333333	0.6117	20317	3573	3954	6717
		0	0	0	0.5714	14749	0	0	0
	5	1	0	0	0.5925	17666	5281	0	0
		0.333333	0.333333	0.333333	0.5911	17481	4111	3954	6717
		0	0	0	0.5355	9800	0	0	0
	6	1	0	0	0.6367	23774	5281.16	0	0
		0.333333	0.333333	0.333333	0.6354	23589	4111	3954	6717
		0	0	0	0.5797	15907	0	0	0
	7	1	0	0	0.593	17742	4389	0	0
		0.333333	0.333333	0.333333	0.5922	17623	3610	3954	6717
		0	0	0	0.5545	12421	0	0	0
	6	1	1	0	0	0.6064	20457	4416	0
0.333333			0.333333	0.333333	0.6055	20331	3622	3954	6717
0			0	0	0.5655	14802	0	0	0
2		1	0	0	0.6041	20621	4416	0	0
		0.333333	0.333333	0.333333	0.6032	20495	3622	3954	6717
		0	0	0	0.5632	14966	0	0	0
3		0.4	0.6	0	0.6172	22427	1131	4419	0
		0.333333	0.333333	0.333333	0.6167	22356	1499	3954	6717
		0	0	0	0.5735	16388	0	0	0
4		1	0	0	0.6141	21999	4202	0	0
		0.333333	0.333333	0.333333	0.6131	21864	3573	3954	6717
		0	0	0	0.5701	15926	0	0	0
5		1	0	0	0.596	19495	5322	0	0
		0.333333	0.333333	0.333333	0.5946	19304	4127	3954	6717
		0	0	0	0.5372	11379	0	0	0
6		1	0	0	0.6368	25136	5322	0	0
		0.333333	0.333333	0.333333	0.6354	24945	4127	3954	6717
		0	0	0	0.5781	17020	0	0	0
7		1	0	0	0.5898	18639	4416	0	0
		0.333333	0.333333	0.333333	0.5889	18513	3622	3954	6717
		0	0	0	0.5488	12984	0	0	0

Figure 4.4 shows the power generated, as defined in Equation 3.46, by the cycles evaluated at 2.0 MPa at the 3 representative flue gas distribution options.  $x=0$  represents the case when no heat is recovered from the flue gases, therefore no economizer, air pre-heater, or dryer are used.  $x=1/3$  represents the case when one third of the flue gases are diverted to each heat recovery component. And,  $x=Max$  represents the fractions at which the cycle yields the maximum power. It can be appreciated how the use of the heat recovery equipment has a great effect on the power generated. Also noticeable is the fact the maximum power generation does not occur when the heat is recovered on equal amounts on each heat recovery component, but it occurs at a flue gas distribution that favors the heat recovery in the economizer.



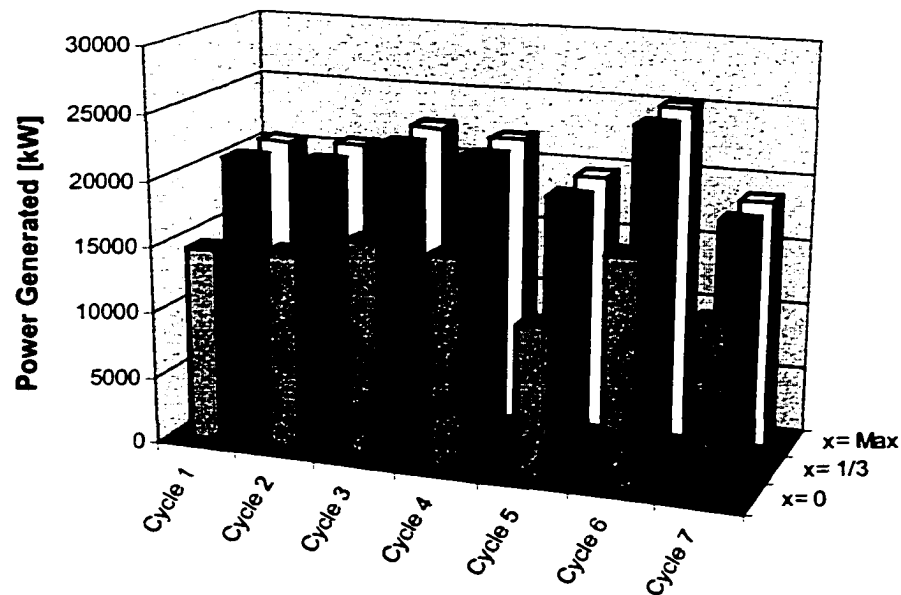
**Figure 4.4: Cycles Performance [2.0 MPa]**

Figure 4.5 is similar to Figure 4.4. It shows the performance of the cycles evaluated at 4.5 MPa, at this pressure all seven cycles can be evaluated and compared. Again we can appreciate the substantial increase in power generation achieved when heat recovery is used, and how there is a better option for flue gas distribution than equal amounts for each component. These options again favor the use of the economizer for heat recovery.



**Figure 4.5: Cycles Performance [4.5 Mpa]**

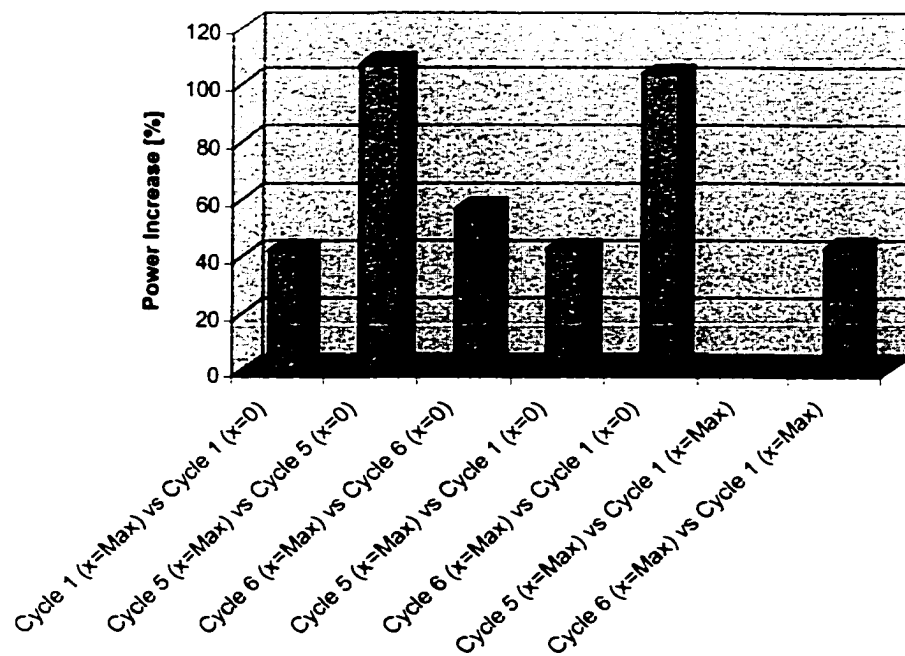
Similar to Figures 4.4 and 4.5, Figure 4.6 shows the same trends rewarding the heat recovery for the cycles evaluated at high pressure (6.0 MPa). A noticeable increase in power generation thanks to the heat recovery equipment, and a preference for the use of the economizer as the heat recovery component that causes the highest increase in power generation can be identified.



**Figure 4.6: Cycles Performance [6.0 MPa]**

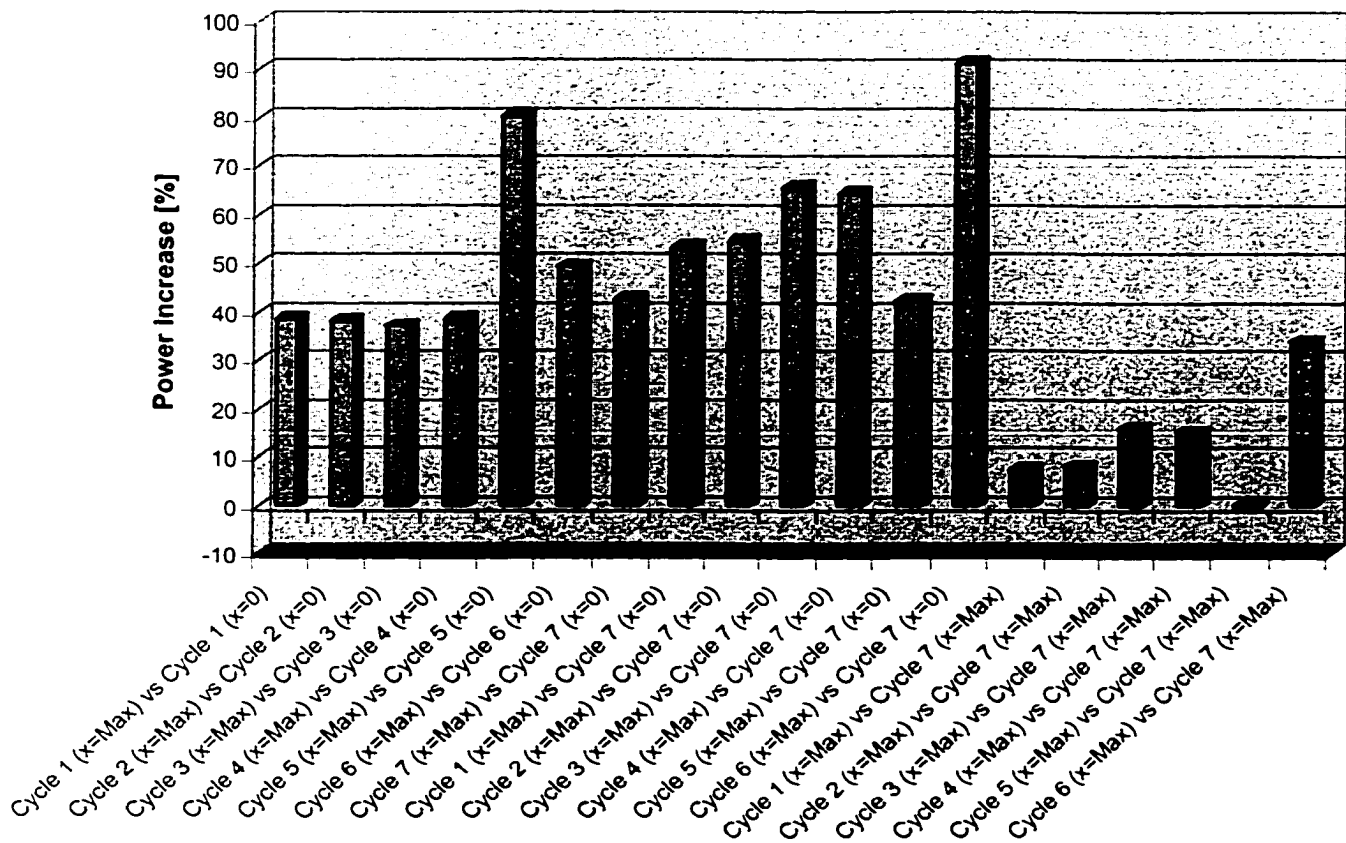
Figures 4.7 through 4.9 represent the relative power increase of the different cycles when comparing each one with maximum heat recovery ( $x=Max$ ) versus no heat recovery ( $x=0$ ). Cycles are also compared to the Base Cycle with maximum heat recovery and without heat recovery. Figure 4.7 shows the relative power increase of the different cycles evaluated at 2.0 MPa. Each cycle with maximum heat recovery was compared

with itself without heat recovery and with the simplest cycle evaluated at this pressure. For example in Figure 4.7 we can see how the maximum heat recovery option increased the power generation about 35% for the simplest cycle evaluated (Cycle 1) and about 65% for the extracting condensing cycle (Cycle 6). When comparing Cycle 6 (maximum heat recovery) with Cycle 1 (no heat recovery) we see an amazing 90% increase in power generation, using the same fuel flow rate and the same steam pressure.



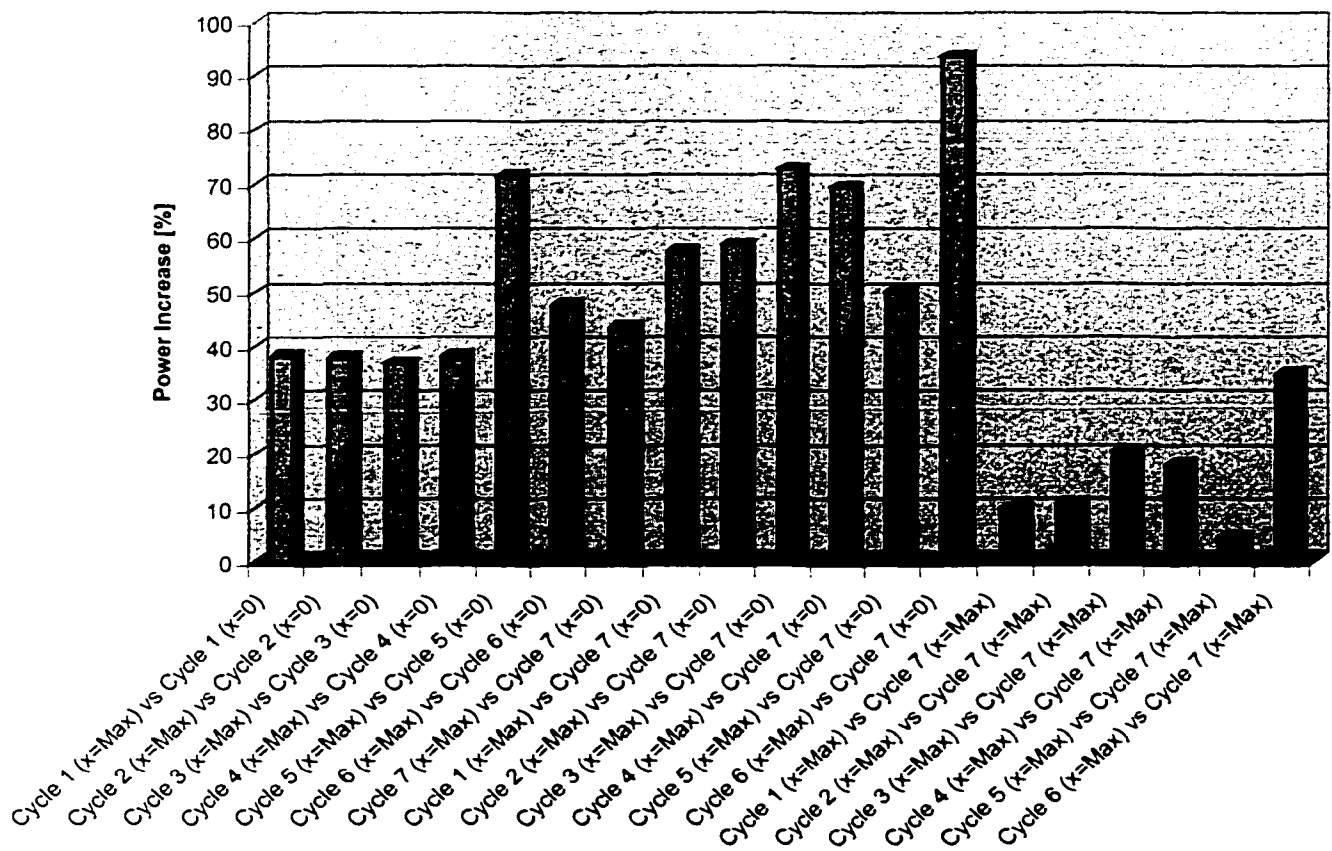
**Figure 4.7: Relative Power Increase [2 MPa]**

Figure 4.8 is similar to Figure 4.7 but this shows the comparison between the cycles evaluated at 4.5 MPa. Again we see the dramatic increase in the power generation when maximum heat recovery is used. On all cycles this increase was over 50%. The most noticeable increase is seen when the double condensing cycle (maximum heat recovery) is compared with the base cycle (no heat recovery). The increase in power generation is more than 110%. The increase when comparing the same two cycles both with maximum heat recovery is more modest but equally impressive, about 60%.



**Figure 4.8: Relative Power Increase [4.5 MPa]**

Figure 4.9 shows trends similar to Figure 4.8. Here the cycles evaluated at 6.0 MPa are compared and the results also encourage the use of double extraction cycles and maximum heat recovery in order to maximize the power generation. Equally impressive is the 120% increase in the power generation when cycle 6, with maximum heat recovery, is compared with cycle 7, with no heat recovery.



**Figure 4.9: Relative Power Increase [6.0 MPa]**

### **4.3.2 – Analysis**

Examination of Table 4.3 and Figures 4.4 to 4.9 leads to the following conclusions:

- Among the three heat recovery components analyzed the dryer is the most irreversible. For the non-condensing cycles, when 1/3 of the flue gases are sent to each of the heat recovery components, the one with the least irreversibility is the economizer, followed by the air pre-heater (~ 10 % more than the economizer) and the dryer (~ 60 % more than the economizer). As for the condensing cycles, the economizer has a little more irreversibility than the air pre-heater, the dryer being again the worst. As we know the irreversibility is a representation of the availability lost during a process. Therefore these results lead us to believe that the economizer is the component where the least work is lost. Even though for the condensing cycles the economizer has a little more irreversibility than the air pre-heater this is more than compensated for by the fact that the heat recovered in the economizer goes directly to the working fluid, while in the air pre-heater it does not. Therefore, it can be said that for equal energy flows, the economizer generates the least irreversibility and correspondingly the least lost work. From this point of view, at least, it is the most effective heat recovery choice.
- In agreement with the results for irreversibility just given, the best performance in terms of highest power generation (remember that the bagasse feed rate is fixed), is

achieved by using most, or all (if possible) of the flue gases in the economizer, and the rest in the air pre-heater. The heat recovery in the economizer is limited by the boiling temperature of the working fluid. Thus, to prevent boiling inside the economizer the heat transfer should be such that the water temperature remains below its boiling point at all times. In the regeneration cycles, where no economizer is used or little flue gas can be sent to it, because of the regenerative heat already added, the best option is to use whatever is possible in the economizer, and the rest in the air pre-heater. If there still some remaining flue gas to be used it should be used in the dryer.

- Contrary to what is commonly thought, the use of the dryer does not increase the power generation more than the use of the other heat recovery equipment (for example Wang et al, 1990; Correia Maranhao, 1986). In addition to this, if the power requirement of the dryer is considered, the energy gain for the cycle must be even lower when the dryer is used than when the economizer is used. The energy required to operate the dryer, which includes transporting belts, fans, etc, is expected to be considerably higher than the extra energy required to pump the water through the economizer.
- Sufficient heat is available in the flue gas so that the bagasse dryer could reduce the moisture content in the bagasse below 20 %, if a large fraction of the flue gas is sent to the dryer. This is something that needs to be prevented to avoid the risk of auto ignition that may occur as a result of spontaneous exothermic reaction (Upadhiaya, 1991), and to prevent the temperature inside the boiler rising to undesirable levels.

- Even though low excess air rates are desirable, higher excess air rates could be used when the dryer and air pre-heater are used in order to keep the adiabatic flame temperature below the ash fusion temperature. In the simulations the excess air rate was not adjusted to decrease the adiabatic flame temperature, it was calculated as a function of the bagasse moisture content based on standard practice described in Section 3.2 of Chapter 3.
- The single extraction cycle was not evaluated for the low-pressure case because the milling pressure was the same as the generated steam pressure. Steam extraction increased the power generation by 8 % and 11 % for the medium and high-pressure cases when compared with the base cases at the same pressures. When the high-pressure single extraction cycle (maximum heat recovery from the flue gas) is compared with the base case at the low pressure (no heat recovery) the increase in power generated was 128 %.
- The effect of the regeneration cycle is more pronounced at higher pressures. This is expected since a rule of thumb says that the optimum extraction pressure for the regeneration is at about half the pressure difference between steam generation and turbine exhaust. In the cases studied the steam was extracted at 2 MPa, so for the low-pressure case the regeneration does no good. While for the high-pressure (maximum heat recovery) case the regeneration increased the power generation 147 % when compared with the low-pressure (no heat recovery) base case. When compared with the high-pressure single extraction cycle (both maximum heat recovery) the

regeneration increased the power generation by 9 %. When compared with the base case (maximum heat recovery) it increased the power generation by 20 %.

- Reheat (maximum heat recovery) also has a positive effect, increasing the power generation 7 % for both medium and high pressure cases when compared with the single extraction (maximum heat recovery) cycles at the same pressures. When comparing the high-pressure reheat cycle with the low-pressure base cycle (no heat recovery) the power generation increased 143 %.
- The single extraction condensing cycle showed a poor performance in all the cases. The reason for this is that all the steam necessary for both milling and process is extracted at 2 MPa to allow the rest to expand in the condenser. By doing this, a lot of steam has to be expanded from 2 MPa to 200 kPa (the process pressure) without producing any work, or equivalently wasting the available energy. This fraction is about 40 % of the steam generated for the maximum heat recovery options and around 50 % of the steam generated for the no heat recovery options. For the 217 tonnes of cane per hour plant evaluated the availability loss was around 7 MW! The use of heat recovery increased the power generation by 100, 80, and 70 % for the low, medium, and high pressure single extraction condensing cycles. When these cycles with maximum heat recovery were compared with the low pressure parallel (base) cycle the increase was of 42, 95, and 115 % respectively.

- The double extraction cycles showed the best performance of all. This cycle makes the best thermodynamic use of the steam, since the steam necessary for milling and for process are extracted at the required pressures, allowing the extra steam to expand in the turbine to the condenser pressure. This cycle increased the power generation by 28 % in both the medium and high-pressure cases when compared with the base cases at the same pressures. The high-pressure double extraction condensing cycle produced 95 % more power than the low-pressure base case, both with maximum heat recovery. When the maximum heat recovery case was compared with no heat recovery an increase in the power generation by 57, 49, and 47 % for the low, medium, and high pressure for double extraction condensing cycles was seen. When these cycles with maximum heat recovery were compared with the low pressure parallel (base) cycle the increase of power generated was of 103, 162, and 177 % respectively.
- The calculations done to determine the effectiveness of the boiler, economizer, and air pre-heater showed high values for the economizer and air pre-heater. This may have a negative effect on the economics of these components since it may suggest relatively big heat exchangers. The range of the effectiveness for the boiler was between 0.7 and 0.8, for the economizer it was around 0.95, and for the air pre-heater it was between 0.82 and 0.95. These values can be reduced if we use a bigger gap between the temperatures of the fluids leaving these heat exchangers. Remember that they the gap used was only 25°.

### ***4.3.3 – Results for Year Round Power Generation***

Two scenarios for year round power generation were evaluated. The first considered using the bagasse generated during the milling season as a year round source of fuel for power generation. The second added the additional biomass from cane trash to the potential fuel. Two cycles for power generation were included in this calculations: a double extraction condensing cycle and a double extraction condensing cycle with reheat. These scenarios were simulated by modifying the algorithm developed in EES, since BIOFIRE does not consider them. Both of these were identified as high performance cycles in the preceding sections. Consequently, their use in a plant with the goal of continuous power generation and maximization of fuel utilization was considered reasonable. Maximum cycle pressures and temperatures were assumed to be 6 MPa and 460 °C, respectively.

#### ***Year Round Power Generation – Bagasse Only***

Consider a scenario in which it is assumed that the harvest season lasts 6 months but the available bagasse is distributed over an 11 ½ month period for year round power generation. For this, the bagasse would have to be baled and stored until it stabilizes at around 25 % moisture content. If the harvest season lasts 6 months and sustains the operation of the mill at 217 tonnes of cane per hour for 23 hours a day during 25 days a month, there would be a total of 224,595 tonnes of bagasse available for the year. This

translates into a year-round average feed rate of 6.3 kg/s of bagasse at 25% moisture content.

### *Cycle Calculations*

In this scenario the double extraction cycle was evaluated with no heat recovery, and with maximum heat recovery at the economizer. Maximum heat recovery at the economizer was chosen considering the result of the previous sections that show this as the best option for heat recovery. The main results can be summarized as follows:

No heat recovery:	Not enough steam was produced to sustain the factory during the harvest season
Maximum heat recovery:	<i>Crushing season:</i> Barely enough steam produced to satisfy factory needs Power Generation = 11.2 MW  <i>No-Crushing season:</i> Power Generation = 20.1 MW

In order to keep a close to uniform power generation year round the maximum heat recovery option for the cycle was also evaluated using different feeding rates for the crushing and no crushing seasons with the following results:

Crushing season:	7.5 kg of bagasse per second at 25 % H <sub>2</sub> O content Power Generation = 15 MW
No-Crushing season:	4.5 kg of bagasse per second at 25 % H <sub>2</sub> O content Power Generation = 14.4 MW

### ***Year Round Power Generation – Using Cane Trash plus bagasse***

To evaluate this scenario the following assumptions were made:

- Cane trash represents 35% of the plant and it is about 30% dry matter (see section 2.1.4)
- Cane trash can dry to 25% moisture content while baled
- 30% of the cane trash is left in the field (see Section 2.1.4 and Section 2.3.5.3.4.2)
- Composition and heating value of the cane trash are the same as those of bagasse

Using these assumptions one kilogram of a sugar cane plant yields 0.07 kg of dry matter as cane trash and 0.1 kg of dry matter as bagasse. Therefore, the use of cane trash adds 70% more biomass to the power plant. With this in mind and assuming the same conditions as the previous scenario, an average bagasse feeding rate of 10.7 kg/s of bagasse at 25% moisture content can be sustained year round.

#### ***Cycle Calculations***

Employing again the double extraction condensing cycle the following results were obtained:

No heat recovery:	During the crushing season Power Generation = 16.6 MW
Maximum heat recovery:	Crushing season: Power Generation= 25.2 MW  No-Crushing season: Power Generation = 34.1 MW

Again, to achieve uniform power generation year round, variable bagasse feeding rates between the crushing and no crushing season were evaluated:

Crushing season: 11.5 kg of bagasse per second at 25 % H<sub>2</sub>O content  
Power Generation = 27.8 MW

No-Crushing season: 8.5 kg of bagasse per second at 25 % H<sub>2</sub>O content  
Power Generation = 27.2 MW

### ***Effect of Reheat on Performance***

In this simulation year round power generation using both bagasse and cane trash was again considered, but a double extraction cycle with reheat was assumed. Other assumptions remained the same, and resulted in the following power predictions:

No heat recovery: Power Generation = 20.4 MW

Maximum heat recovery: Power Generation = 29 MW

For a variable feed rate designed to give a nearly constant power output year round, the results were:

Crushing season: 11 kg of bagasse per second at 25 % H<sub>2</sub>O content  
Power Generation = 30.1 MW

No-Crushing season: 9 kg of bagasse per second at 25 % H<sub>2</sub>O content  
Power Generation = 30.4 MW

Two observations can be made:

- The year round scenarios show that by using cane trash the mill can generate power all year round, and by using different feeding rates during each season the mill can generate an almost constant amount of power through the year.
- The year round scenarios also show the requirement of smaller equipment when compared with the scenarios for which all the bagasse is used during the crushing season. For example, if all the bagasse produced at the 217 tonnes of cane per hour facility is used during the crushing season, a bagasse boiler feeding rate of 18.1 kg of bagasse per second will result. This compares with 11.5 kg of bagasse per second rate when using cane trash but generating all year round. Smaller size equipment represents smaller capital investment, and longer generation periods represent shorter payback periods. Of course, the cost of baling and storing the bagasse must be considered when a complete analysis is done.

## *Chapter 5*

# **Conclusions and Recommendations**

In this Chapter conclusions and recommendations for further research are presented. The conclusions represent a summary of the analysis of the results achieved and disclose the answers to the fundamental questions that were posed at the beginning of this research. In the recommendations section, some modifications and extensions that could help to improve the scope of this dissertation are identified. Finally, research necessary to provide more realistic inputs to the simulation and extend its applications are suggested.

### ***5.1 – Conclusions***

The following questions were posed in the Introduction: Is there an advantage as to where to recover the most heat? Should we use all these components at the same time? Should we recover more in a particular component, and reduce the size or eliminate another? Is there a particular configuration that gives the best performance? Should the design be done for each specific scenario, customized? Are there rules of thumb for best

designs that can be followed? Answers to these questions obtained through application of the simulation developed in this work and examination of results predicted using it are given below.

- **The economizer is the heat recovery component of choice.** From a technical point of view, the analysis done shows that in all cases most power was generated when most heat was recovered in the economizer. The exception is for the regenerative cycle when the water enters the economizer at a higher temperature than in the other cycles. In this case energy addition in the economizer was limited by boiling. Similarly; when the steam pressure is low the boiling point can also be reached in the economizer. In these two cases the best option is to use enough flue gas in the economizer to raise the water temperature to slightly below the boiling point. The rest of the heat then should be used in the air pre-heater.
- **The air pre-heater is the second best option for heat recovery.** BIOFIRE showed that the air pre-heater was the second best choice for heat recovery during the simulations. More research needs to be done to determine the effects of air pre-heating on the combustion of bagasse. It is expected that pre-heating will improve combustion efficiency and thereby the performance of the cycle, but whether this will be sufficient to make pre-heating a more attractive heat recovery option will require details of the combustion process not currently available.

- **The effectiveness of the economizer and air pre-heater are high.** This may suggest big heat exchangers that may not be the best economical option. However, without economic data for these components the magnitude of this effect is not easy to determine.
- **The bagasse dryer is the most irreversible of the three heat recovery components evaluated.** In addition to this, the bagasse dryer requires more mechanical complexity and power to operate than the other heat recovery components. Having many moving parts it will also require more maintenance and possibly more down time than the other components. In consequence, the dryer is not an advisable heat recovery component unless it is absolutely required to provide a combustible feed.
- **The single extraction condensing cycle is the least advisable for power generation.** In this cycle the steam necessary for process is taken out of the turbine at the milling pressure and then expanded without producing any work. This affects greatly the power generation performance of the cycle.
- **The double extraction condensing cycle generates the most power.** This is expected since this cycle is the one that makes the best use of the steam. The improvement in efficiency can be up to 16% greater than that of the simple base cycle.

- **The methodology used for the calculations is a fast way to evaluate the performance of different co-generation cycles.** BIOFIRE utilizes this methodology, and it demonstrates how fast the different cycles can be evaluated. BIOFIRE is a valuable tool for comparing design options and can reduce evaluation of several cycles and options to a few clicks of the mouse.
- **Milling at higher pressure allows more power generation.** Milling at the same pressure as that of the steam generated, allowed the generation of about 10% more power than the standard practice of milling at a lower pressure (around 2 MPa).
- **Economic data, cost of equipment, is very hard to obtain.** It was a difficult task to get manufacturers to provide cost estimates for the components analyzed, especially if they did not believe that a sale was possible. This limited the initial goal of doing an economic analysis to all the scenarios evaluated and only the tools to do this were included in the software developed.
- **Year round scenarios should be considered for any co-generation project.** Since distributing the fuel throughout the year will decrease the bagasse flow rate, the capacity generation will be smaller, but the coefficient of performance will be considerably higher. This will greatly and positively impact the economics of the project.

- **Baling and storing bagasse showed to be a worthy option.** The year round scenarios showed how baling and storing bagasse takes advantage of a dryer fuel and the energy required to evaporate the moisture can be used for power generation. The economic impact of this option remains to be determined.

## ***5.2 – Recommendations for further research***

- **Examine the possibility of using variable excess air to reduce the adiabatic flame temperature.** Including in the algorithm variable excess air rate that will keep the adiabatic flame temperature below the desired limit is an interesting option to consider. During this research the excess air rate was taken to be directly proportional to the moisture content of the bagasse. Therefore, under some scenarios when the bagasse moisture content was low, the adiabatic flame temperature was higher than the limit for ash melting. During the simulations these scenarios were discarded, but it would be interesting to see what happens to overall performance when increasing the excess air flow rate is used to decrease the adiabatic flame temperature.
- The methodology followed specified the temperature of the flue gas leaving the economizer and the air pre-heater. Under some scenarios when the temperature of the flue gas leaving the heat exchanger is very low and the temperature of the cooler stream entering the heat exchanger is very high, this might cause the effectiveness of

the heat exchanger to be greater than 1. This is not possible and would violate the second law of thermodynamics. Though this did not occur in any of the scenarios evaluated in this work, a better approach might be to use the effectiveness as a parameter of design to determine the temperature of the flue gas leaving the heat exchanger.

- **Develop partnerships with equipment manufacturers.** In order to obtain a more representative economic analysis a partnership with a boiler/turbine manufacturer could be used to obtain accurate equipment costs. This would allow an accurate economic assessment of the different scenarios.
- **Determine the relationship between moisture content of bagasse and required excess air.** Very little information was found about the relationship of these two variables. It is known that at higher moisture contents more excess air is needed to ensure mixing of fuel in the boiler and achieve optimum combustion. However, a relationship that yields the optimum excess air for a given moisture level was not found in the literature. In the simulation a relationship based on some data reported by one author was used.
- **Determine the relationship between moisture content of bagasse and combustion efficiency.** It has been found that at higher moisture contents, higher carbon remains in the ash. No quantitative relationship between these two was found, therefore, a constant value was used during the simulations.

- **Determine the relationship between combustion air temperature and combustion efficiency.** It is generally assumed that with hotter combustion air the combustion efficiency is better. Here again no data reporting the relationship of these two was found and a constant value was used during the simulations.
- **Determine of the value for the specific heat of the bagasse.** No bibliography on this subject was found during the research. A constant value of 1.76 kJ/(kg K) was used, this is a typical value assumed for soft wood.
- **Develop a method for baling together cane trash and fresh bagasse.** This may have the advantage of providing a lower average moisture in the bale allowing the bagasse to dry faster and providing an uniform mixture of the fuel (bagasse and cane trash) for later use.
- **Investigate pumped storage of bagasse energy.** In some locations hydropower is the main energy source. In these locations the harvesting season often coincides with the dry season and low hydro resources. The use of energy generated from bagasse to pump water back to the reservoir could be a viable option for the use of power generated during low demand periods.

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## 7 – *Glossary*

- **Air pre-heater:** Heat exchanger where the heating fluid is the flue gas coming from the furnace and the heated fluid is the air going to be use for combustion in the furnace.
- **Ash:** Inorganic residue remaining after incineration.
- **Bagasse:** The fibrous residue of the sugar cane after it has been crushed on a train of mill to extract the juice from it.
- **Bagasse Dryer:** Dryer using flue gas from the furnace to reduce the moisture content of the bagasse before it enters the furnace to be burned.
- **Cane Trash:** Tops and leaves of the sugar cane plant usually left at the field after harvesting of the cane.
- **Co-generation:** Simultaneous generation of heat and power in the same power plant.

- **Economizer:** Also known as water pre-heater. Heat exchanger where the heating fluid is the flue gas coming from the furnace and the heated fluid is the water going to the furnace.
- **Flue Gas:** Gaseous products of combustion leaving the combustion chamber of the furnace.

## ***8 – Appendix A***

# ***Economic Analysis Example***

Since it was a difficult task to collect economic data (costs) for the equipment desired this section summarizes the data obtained and together with some assumptions about cost of equipment give an example of the economic analysis that can be done with BIOFIRE.

Boiler economic data was impossible to find, therefore an estimating number given by a power generation consultant was used. This number, \$300/kWe, represents the dollars per kW of equipment installed in a typical power plant and need to be translated into thermal energy figures. By assuming that a typical plant using biomass has an thermal efficiency of 20% it could be said that the cost figure for thermal energy is \$60/kWth. This number was used to estimate the cost of the boilers selected in this section.

Turbine's cost was much easier to find. In fact 4 different manufacturers provided such data, Table A-1 summarizes it. Data from manufacturers 1, 2, and 3, was used to

determine an average cost per kW. The data from manufacturer 4 was not included in this average because it seems considerable lower than the other 3.

**Table A-1: Steam Turbines Manufacturer's Data**

<b>Turbine</b>	<b>Manufacturer 1</b>	<b>Manufacturer 2</b>	<b>Manufacturer 3</b>	<b>Manufacturer 4</b>	<b>Average 1-2-3</b>
2 MPa Back Pressure	10,172 kW \$2,950,000 \$290/kW	10,050 kW \$2,704,000 \$269/kW	10,200 kW \$3,040,000 \$298/kW	10,450 kW \$2,400,000 \$230/kW	\$286/kW
2 MPa Back Pressure	14,430 kW \$3,150,000 \$218/kW	14,175 kW \$2,847,000 \$200/kW	14,700 kW \$3,153,000 \$214/kW	14,700 kW \$2,450,000 \$167/kW	\$211/kW
2 MPa Condensing	20,425 kW \$4,100,000 \$200/kW	18,700 kW \$4,674,000 \$250/kW	19,900 kW \$5,520,000 \$277/kW	20,600 kW \$3,050,000 \$148/kW	\$242/kW
6 MPa Back Pressure	21,000 kW \$3,350,000 \$155/kW	20,200 kW \$3,347,000 \$166/kW	21,000 kW \$3,350,000 \$160/kW	21,000 kW \$2,700,000 \$129/kW	\$160/kW
6 MPa Back Pressure	14,400 kW \$3,125,000 \$217/kW	13,500 kW \$2,748,000 \$204/kW	14,000 kW \$3,000,000 \$214/kW	13,800 kW \$2,550,000 \$185/kW	\$212/kW
6 MPa Condensing	27,800 kW \$4,375,000 \$157/kW	26,150 kW \$5,146,000 \$197/kW	26,500 kW \$6,200,000 \$234/kW	28,000 kW \$3,250,000 \$116/kW	\$196/kW

Eight systems were evaluated to compare low pressure versus high pressure, no heat recovery versus heat recovery, and, back pressure versus condensing turbines. Two other double extraction systems were evaluated to compare year round operation with and without the use of cane trash. Table A-2 summarizes the systems evaluated.

The economic analysis done sized the boiler proportional to the amount of energy that the water/steam gained in the boiler. As it was said before the cost assumed was

\$60/kWh. For the turbines, after determining the size of the turbine (rating) using BIOFIRE, the \$/kWh of the most similar turbine from Table A-1 was used to estimate the price of it. Using the Annual Energy Output calculator from BIOFIRE the total power generated was calculated. With this and the estimated capital cost for the boilers and turbines a partial cost of energy was calculated. The results of these calculations are summarized in Table A-3. It is important to keep in mind that these calculations were done only as an example and many of the capital costs (other equipment, construction, engineering, etc) are not included.

**Table A-2: Summary of Systems Analyzed**

<b>System #</b>	<b>Pressure</b>	<b>Cycle #</b>	<b>Heat Recovery</b>	<b>Comments</b>
1	2 MPa	1	No	Low pressure - Back pressure turbine No heat recovery - Crushing season only
2	2 MPa	1	Maximum	Low pressure - Back pressure turbine Max. heat recovery - Crushing season only
3	2 MPa	6	Maximum	Low pressure - One extraction condensing turbine Max. heat recovery - Crushing season only
4	6 MPa	7	No	High pressure - Back pressure turbine No heat recovery - Crushing season only
5	6 MPa	7	Maximum	High pressure - Back pressure turbine Max. heat recovery - Crushing season only
6	6 MPa	6	Maximum	High pressure - Two extractions condensing turbine Max. heat recovery - Crushing season only
7	6 MPa	6	Maximum	High pressure - Two extractions condensing turbine Max. heat recovery - Year round operation No cane trash
8	6 MPa	6	Maximum	High pressure - Two extractions condensing turbine Max. heat recovery - Year round operation With cane trash

**Table A-3: Cost of Energy Results**

<b>System #</b>	<b>Boiler Rating [MWth]</b>	<b>Turbine Rating [Mwe]</b>	<b>AEO [MWh]</b>	<b>Boiler Cost [\$*1000]</b>	<b>Turbine Cost [\$*1000]</b>	<b>Total Cost [\$*1000]</b>	<b>COE [\$/kWh]</b>
1	96	9	38,880	5,760	2,574	8,334	0.0278
2	126	13	56,160	7,560	2,743	10,300	0.0238
3	126	18	77,760	7,560	4,356	11,910	0.0198
4	96	13	56,160	5,760	2,756	8,516	0.0196
5	126	19	82,080	7,560	3,040	10,600	0.0167
6	126	25	108,000	7,560	4,900	12,460	0.0149
7	88	15 Year round	129,600	5,280	2,940	8,220	0.0082
8	136	28 Year round	241,920	8,160	5,488	13,640	0.0073

Also as an example, the payback periods and internal rates of return were calculated using BIOFIRE. These calculations were done under the following assumptions:

- The total capital cost of the plant is twice the cost of the boiler and turbine
- The cash flow was calculated as the annual energy output times a power rate of \$0.04 per kWh. This gives a value of \$0.04/kWh to the power used and the mill and to the power sold to the utility. This is not necessary true since the mill may be paying a higher rate for the power it buys than the one that it would get for the power that it may sell. In addition to this the utility may pay under different rates depending the time of the day, the day of the week, and the month of the year.
- The interest rate assumed was 9%
- The project life assumed was 20 years

Table A-4 summarizes these results.

**Table A-4: Payback Periods and Internal Rates of Return**

<b>System #</b>	<b>Capital Cost [\$*1000]</b>	<b>Cash Flow [\$*1000]</b>	<b>Simple Payback [Years]</b>	<b>Discounted Payback [Years]</b>	<b>IRR [%]</b>
1	16,668	1,555.20	10.8	39	6.6
2	20,600	2,246.40	9.2	21	8.7
3	23,820	3,110.40	7.7	14	11.4
4	17,032	2,246.40	7.6	14	11.6
5	21,200	3,283.20	6.5	11	14.3
6	24,920	4,320.00	5.8	9	16.4
7	16,440	5,134.00	3.2	4	31.4
8	27,280	9,676.80	2.9	4	35.4

Tables A-3 and A-4 show how the addition of heat recovery, the use of higher pressure, and the use of condensing turbines, all are positive upgrades to the plant that reduce the cost of each kWh generated, and can improve the payback periods and rate of return of the project.

Here we need to remember that these upgrades are feasible only if there is a market for the power to be sold. It would also be fair to give a value to the steam generated, and include it in the annual cash flow, or take the fraction of the capital investment that represent this steam from the analysis so that the analysis is done only considering the power to be sold.