AN ABSTRACT OF THE THESIS OF

John E. Simonds for the degree of Master of Science in Mechanical Engineering presented on July 18, 1990.

Title: Computer Model for Estimating Cogeneration System Performance and Economic Feasibility

Redacted for Privacy

Abstract Approved: _____

Dr. Dwight J. Bushnell

Cogeneration is the simultaneous generation and use of both electricity (or mechanical power) and thermal energy from a single fuel source. It is a technology that has been used for years by industries, commercial institutions, and municipalities that need both heat and electrical power. An analytical approach has been developed to determine cogeneration feasibility. The analysis is designed to estimate economic feasibility based on the performance characteristics of different types of equipment, including wood-fired boilers. It can be used to compare several options for cogeneration systems. The economic analysis is based on simple payback. A microcomputer spreadsheet application program performs the technical analysis.

The program offers many features that are designed to improve and simplify prediction of cogeneration system performance:

- Available energy calculation for wood fuel based on species and moisture content. Also uses natural gas and fuel oils.
- Hourly steam and electrical load profiles for up to 6 typical operating days per year.

- Three steam turbine types:
  Condensing,
  Backpressure,
  Autoextraction.

- Full and part load performance modelling.

- Three operating modes:
  Thermal load following,
  Electrical load following,
  Constant load operation.

- Steam enthalpy calculation for any operating conditions.

- User-defined utility rate structure.

- Independent value for excess electricity sold.

Our approach is to compare the annual cost of providing thermal and electrical power separately and independently (i.e., electric utility power and a steam boiler) to that of producing the required power simultaneously (cogeneration). This requires an understanding of the power requirements, the utility rate structures, fuel costs, operation and maintenance costs, and available technology for power production.

The program calculates fuel requirements, electrical energy and power production, costs, equipment size and cost, total cost savings, and simple payback for several cogeneration alternatives.

COMPUTER MODEL FOR ESTIMATING COGENERATION SYSTEM PERFORMANCE AND ECONOMIC FEASIBILITY

by

John E. Simonds

A THESIS
submitted to
Oregon State University

in partial fulfillment of
the requirements for the
degree of
Master of Science

Completed July 18, 1990
Commencement June, 1991

APPROVED:

Redacted for Privacy

Associate Professor of Mechanical Engineering
in charge of major

Redacted for Privacy

Head of department of Mechanical Engineering

Redacted for Privacy

Dean of Graduate School

Date thesis is presented July 18, 1990

ACKNOWLEDGEMENT

My graduate studies would not have been possible without the love, understanding, and support of my partner Jennifer and my family. I am also deeply indebted to my mother, my father and my grandfather for their inspiration, support, and guidance.

I would like to thank my major professors, Dr. Dwight Bushnell and Dr. Greg Wheeler for the opportunity to work in the field of energy conservation and for their support and counsel throughout the production of this work.

I would also like to thank Marcus Wilcox and Larry Luskay, cohorts in conservation, for their help and companionship during my graduate career.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

COMPUTER MODEL FOR ESTIMATING COGENERATION SYSTEM PERFORMANCE AND ECONOMIC FEASIBILITY

INTRODUCTION

The wood products industry generates considerable waste. The utility of this waste wood varies with quality, location, and local market conditions. One method of wood waste disposal is incineration. Since many wood products manufacturers require both electricity and heat, a wood-fired, steam-turbine cogeneration system may be feasible. As the cost of electricity increases, wood-fired cogeneration will become a more attractive alternative.

Cogeneration applications can be classified into two categories; topping cycles, and bottoming cycles. In a topping cycle, electricity or mechanical power is produced first and the waste heat is used to meet a thermal load. In a bottoming cycle, the energy leaving a process as heat is converted into electricity. This tends to be a low efficiency conversion, applicable only to certain processes.

There are three main systems available for cogeneration applications: 1) engine systems, 2) gas turbine systems, and 3) boiler/steam turbine systems. A brief description of each type of system follows:

Engine systems. A diesel, natural gas, or gasoline engine is the prime mover. The shaft power from the engine is used directly or turns a generator to produce electricity. Waste heat is recovered from the water-cooled cylinders, the engine lubricating oil, and the exhaust gases as either hot water, hot air, or steam. These systems use a topping cycle.

Gas turbine systems. Hot combustion products are expanded through a turbine. The turbine drives a generator to produce electricity. Heat is recovered from the exhaust gases leaving the turbine. The exhaust gases are commonly used in one of three ways; 1) directly, as in a drying process, 2) to produce process steam in a heat recovery boiler, or 3) to produce hot water in a heat recovery hot water boiler. These systems use a topping cycle.

Boiler/steam turbine systems. Natural gas, oil, and solid or waste fuels are commonly burned alone or in combination in a boiler. The boiler produces steam in a boiler which is expanded in a turbine to drive the electrical generator. The thermal load is supplied by the steam flow. These systems can use either a topping or bottoming cycle.

Each of these systems can operate in a variety of configurations, depending on the application. For example, the engine systems are designed to operate at high or low speed with capacities ranging from 75 kilowatts to 30 megawatts.

Gas turbine systems are designed to operate in a variety of thermodynamic cycles including a simple cycle gas turbine with a heat recovery steam generator (HRSG) and a combined-cycle with a gas turbine, HRSG, and steam turbine. Several types of turbines can be used in the boiler/steam turbine system, depending on the load requirements and the mode of operation. The latter is the system of interest for this study.

The analysis has been divided into seven sections:

1.) Fuels
2.) Loading Profiles
3.) Utility Rate Schedules
4.) Equipment
5.) Operation
6.) System Performance
7.) Cost Estimation

The model will analyze a specific system selected by the user or run all options automatically, creating a summary table of system performance.

1. FUELS

Thermal energy in our analysis is provided by a steam boiler fired with natural gas, oil, or waste wood. Three characteristics of the fuel and its combustion are considered:

- Availability and Cost
- Fuel Mix (Wood Fuel only)
- Boiler Efficiency

1.1 Availability and Cost

The existing annual cost of providing thermal and electrical energy are needed. This requires monthly fuel use for a minimum period of one year. The average cost of boiler fuel ($/10^{6}Btu) is used to estimate monthly fuel costs. The existing cost for electrical energy is based on the electric utility rate schedule which will be discussed later. Table 1 shows sample fuel data used by the model.

Table 1. Sample availability and cost data for hogged fuel.

	PLANT PRODUCED		PURCHASED	
Month	Fuel (MMBtu)	Value ($)	Fuel (MMBtu)	Cost ($)
January	900	$990	1,500	$2,250
February	950	$1,045	1,600	$2,400
March	975	$1,073	1,570	$2,355
April	960	$1,056	1,580	$2,370
May	1,010	$1,111	1,560	$2,340
June	1,000	$1,100	1,550	$2,325
July	1,020	$1,122	1,530	$2,295
August	990	$1,089	1,510	$2,265
September	985	$1,084	1,450	$2,175
October	990	$1,089	1,500	$2,250
November	1,000	$1,100	1,540	$2,310
December	1,010	$1,111	1,590	$2,385

If natural gas or oil is used to fire the boiler, consumption and cost from the monthly bills can be used. If wood waste is produced by the facility, monthly records or estimates of quantity and value are required. If wood fuel is purchased, the value is the purchase price. If there is a cost to dispose of wood waste, then the value could be negative. If excess waste wood can be sold, the value would be the sales price at the mill. This information is used to estimate the expense of additional wood fuel required by the cogeneration system.

1.2 Fuel Mix

The type of wood fuel must be specified for wood-fired boilers. The heating value of wood fuel depends on species and moisture content. The program includes a database of 26 wood species common to the northwest. The database includes the higher heating value (HHV) and green moisture content (wet basis). The higher heating value is the amount of heat recovered when the products of complete combustion of a unit quantity of a fuel are cooled to the initial temperature of the air and fuel, including the latent heat of the water vapor in the products of combustion [1].

Wet basis moisture content (WBM%) is defined as the weight of water in a sample of wood divided by the sum of the ovendry weight of the wood and the weight of the water. Dry basis moisture content (DBM%) is defined as the weight of the water in a sample of wood divided by only the ovendry weight of the wood. Either wet or dry moisture content can be determined from the other using the following relationship:

$$WBM\% = DBM\%/(100\%+DBM\%) \quad (1)$$

Figure 1 is a plot of this relationship.

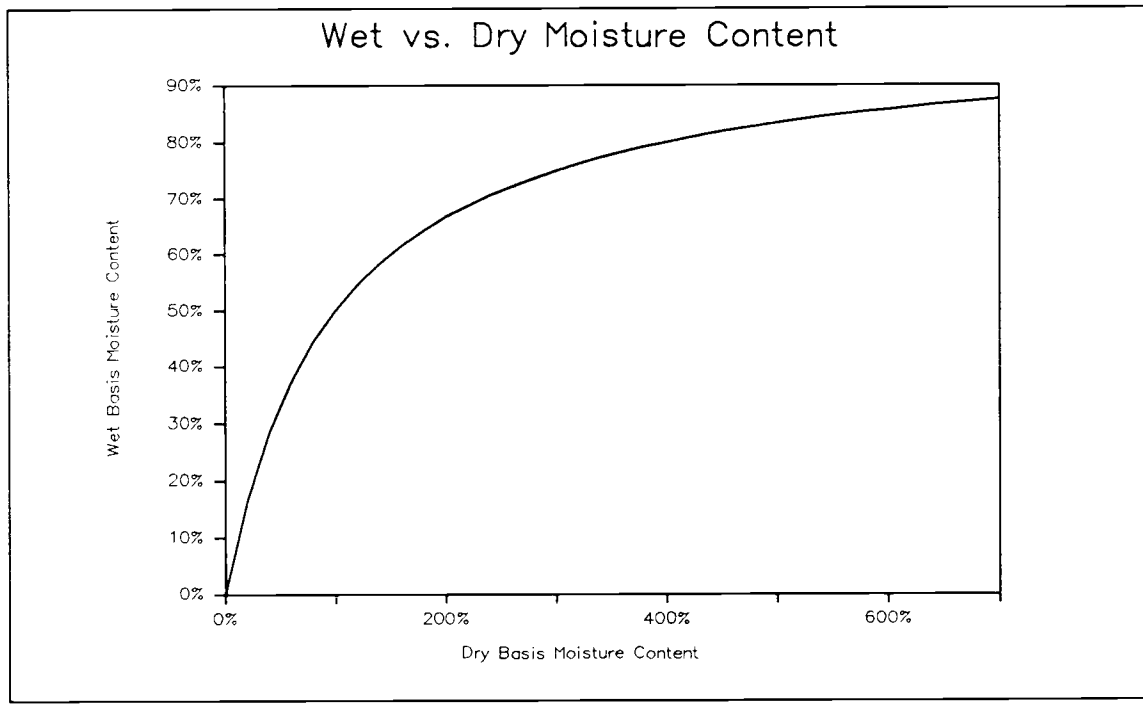

Figure 1. Relationship of wet and dry moisture content.

Wood fuel can be proportioned by species to more accurately model available heat. These values are used to calculate available heat, efficiency, and fuel requirements.

Moisture content of wood is an important factor in determining the available heat. Table 2 shows how available heat decreases with moisture content.

Table 2. Available energy in typical wood fuel as a function of moisture content.

Wet Basis Moisture Content	Available Energy (Btu/lb)	Dry Basis Moisture Content
0 %	9,000	0 %
20 %	7,200	25 %
40 %	5,400	67 %
60 %	3,600	150 %

Default values for moisture content of green wood are used by the model. The user can override these values, if desired.

Each fuel is specified in terms of its content of carbon (%C), hydrogen (%H), oxygen (%O), nitrogen (%N), sulfur (%S), and ash (%Ash). For wood fuels, the wet basis moisture content (%M) is also specified. The model uses fuel elemental composition in determining available heat in the fuel and also in the calculation of boiler efficiency. Table 3 shows typical elemental composition for several fuels. All percentages are by weight. The wood fuel database is listed completely in Appendix C.

Table 3. Typical fuel properties (percentages by weight).

Fuel Type	%C	%H	%O	%N	%S	%Ash	HHV (Btu/lb)
Softwood	52.9	6.3	39.7	0.1	Trace	1.0	9,000
Hardwood	50.8	6.4	41.8	0.4	Trace	0.6	9,000
Natural Gas	77.0	23.0	0.0	0.0	0.0	0.0	23,500
#2 Oil *	86.4	12.7	0.2	---	0.7	Trace	19,567
#6 Oil *	85.7	10.5	0.92	---	2.8	0.08	18,266

* %O value is oxygen and nitrogen combined.

If several wood species are used, the components of the composite fuel are used in the efficiency calculation. The higher heating value and moisture content are also proportioned.

1.3 Boiler Efficiency

An existing boiler can be used if it can produce the quality and quantity of steam required for power production. For either a new or existing boiler, an estimate of efficiency is needed to determine fuel requirements. The

model calculates boiler efficiency using the American Society of Mechanical Engineers (ASME) Heat Loss Method [2].

This method calculates the following energy losses as a percent of the available energy in the fuel:

- Dry Gas
- Moisture in the Fuel
- Moisture Formation from Hydrogen in the Fuel
- Formation of Carbon Monoxide
- Radiation

Energy losses are subtracted from 100% to obtain an efficiency. These calculations use the "as-fired" heating value (available energy) defined by the following equation:

$$AHV = HHV(100\%-WBM\%)/100\% \qquad (2)$$

This quantity has units of Btu/lb and represents the amount of energy available in the fuel as it enters the boiler.

Nomenclature used in the equations for calculating boiler efficiency are listed below:

$\%CO_2$	=	Percent carbon dioxide by volume: %
$\%O_2$	=	Percent oxygen by volume: %
$\%N_2$	=	Percent nitrogen by volume: %
	=	$100\% - \%CO_2 - \%O_2 - \%CO$
CO	=	Carbon monoxide: ppm
%CO	=	Carbon monoxide by volume: % (ppm x 10^{-4})
h_g	=	Enthalpy of water vapor at stack gas temperature: Btu/lb
h_f	=	Enthalpy of liquid water at fuel temperature: Btu/lb
Ts	=	Stack gas temperature: °F
Tc	=	Combustion intake air temperature: °F

Each of the heat loss calculations are described below.

Dry Gas Loss (HL_{DG}). The percentage of heat lost to dry flue gases in the stack is calculated using the following equation:

$$HL_{DG} = P_f Cp(Ts-Tc)100/AHV \quad (3)$$

where

P_f = pounds of dry stack gases leaving the boiler per pound of as-fired fuel
= $\{[44(\%CO_2) + 32(\%O_2) + 28(\%CO + \%N_2)] / 12(\%CO_2 + \%CO)\} [(\%C / 100) + (12(\%S) / (32\ (100))]$

Cp = Specific heat of dry gases: 0.24 Btu/lb

The coefficients in the above equation (44, 32, 28) represent the atomic weights of the molecules they precede. The coefficient in the denominator (12) represents the atomic weight of carbon in CO_2 and CO.

Moisture in Fuel Loss (HL_M). The percentage of heat lost to moisture present in the fuel is calculated by

$$HL_M = WBM\%(h_g-h_f)/AHV \quad (4)$$

Moisture Formation Loss (HL_{MF}). The percentage of heat lost to moisture formed during combustion from hydrogen (H) in the fuel is calculated by:

$$HL_{MF} = 9\ \%H(h_g-h_f)/AHV \quad (5)$$

where the coefficient (9) is the ratio of the atomic weights of water (H_2O) to hydrogen (H_2).

Carbon Monoxide Loss (HL_{CO}). The percentage of heat lost to the formation of carbon monoxide (CO) during combustion is calculated as follows:

$$HL_{CO} = [C_b\ 10{,}160\ \%CO/(\%CO+\%CO_2)]/AHV \qquad (6)$$

where

C_b	=	%C - (W_{dp} H_{dp}/14,500)
W_{dp}	=	pounds of total dry refuse per pound of "as-fired" fuel
H_{dp}	=	Heating value for total dry refuse from laboratory determination: (Btu/lb dry refuse)
14,500	=	Heat value of carbon: (Btu/lb)
10,160	=	Heating value of CO: (Btu/lb)

The quantity C_b represents the amount of carbon burned per pound of as-fired fuel. We neglect dry refuse and assume C_b is equal to %C (lb carbon/lb fuel).

Radiation Loss (HL_R). The percentage of heat lost to radiation and convection from the boiler jacket can be estimated using the American Boiler Manufacturer's Association (ABMA) standard radiation loss chart. To use this chart, the boiler's rated and actual heat output (10^6Btu/hr), as well as jacket design, must be known. A radiation heat loss of 4% is used as the default value.

$$HL_R = 4\% \qquad (7)$$

Boiler Efficiency (η_B). The overall boiler efficiency is calculated by subtracting all of the losses (in percent) from 100%:

$$\eta_B = 100\% - HL_{DG} - HL_M - HL_{MF} - HL_{CO} - HL_R \qquad (8)$$

A stack gas analysis can be used to determine existing boiler operating conditions. Table 4 shows the default efficiencies used by the model according to fuel type.

Table 4. Default boiler efficiencies.

Fuel Type	Efficiency
Wood	65 %
Natural Gas	75 %
Oil	80 %

The model will calculate boiler efficiency on a monthly basis for wood-fired systems, allowing for varying moisture content of the fuel throughout the year.

2. LOAD PROFILES

Both thermal and electrical load requirements can be represented by hourly load profiles. Both load profiles, occurrence of typical days, and steam conditions in the facility are needed to perform the cogeneration analysis:

- Steam Conditions
- Day Types
- Hourly Profiles

2.1 Steam Conditions

The thermal load is specified by steam pressure, temperature, and flowrate at certain points in the system. Steam pressure and temperature must be specified at the following points:

- Thermal Load
- Boiler Inlet (feedwater)
- Boiler Outlet

Pressure and temperature determine the total available energy in the steam. This quantity is called enthalpy. Enthalpy of steam has been calculated for a wide range of pressures and temperatures and published in 'steam tables' [3]. The model calculates steam enthalpy at these locations in the system using equations which determine enthalpy as a function of pressure and temperature [4, 5].

2.2 Day Types

Ideally, thermal and electrical power requirements and costs would be known for each day of the year. Cogeneration performance could then be calculated for each day. However, this would be quite costly in terms of time and effort.

One approach is to model the year as a small number of "typical days." For example, a sawmill operation might easily be described by a typical two-shift day, and a typical weekend and holiday day. For each of these two day types, the thermal and electrical power requirements would be monitored or estimated. If the number of similar days of each type were specified for a year, annual plant energy requirements could be estimated.

The model was designed to accept up to six day types. For each month the number of occurrences of each day type is specified. Cogeneration system performance is then determined for each day type. The model uses the monthly occurrences of day types as multipliers to calculate monthly fuel use and power costs. Monthly totals are summed to determine annual performance.

2.3 Hourly Profiles

For each day type steam use (in pounds per hour) and electrical power (in kilowatts) are specified for each hour. Figures 2 and 3 show typical hourly profiles of steam and electrical power use.

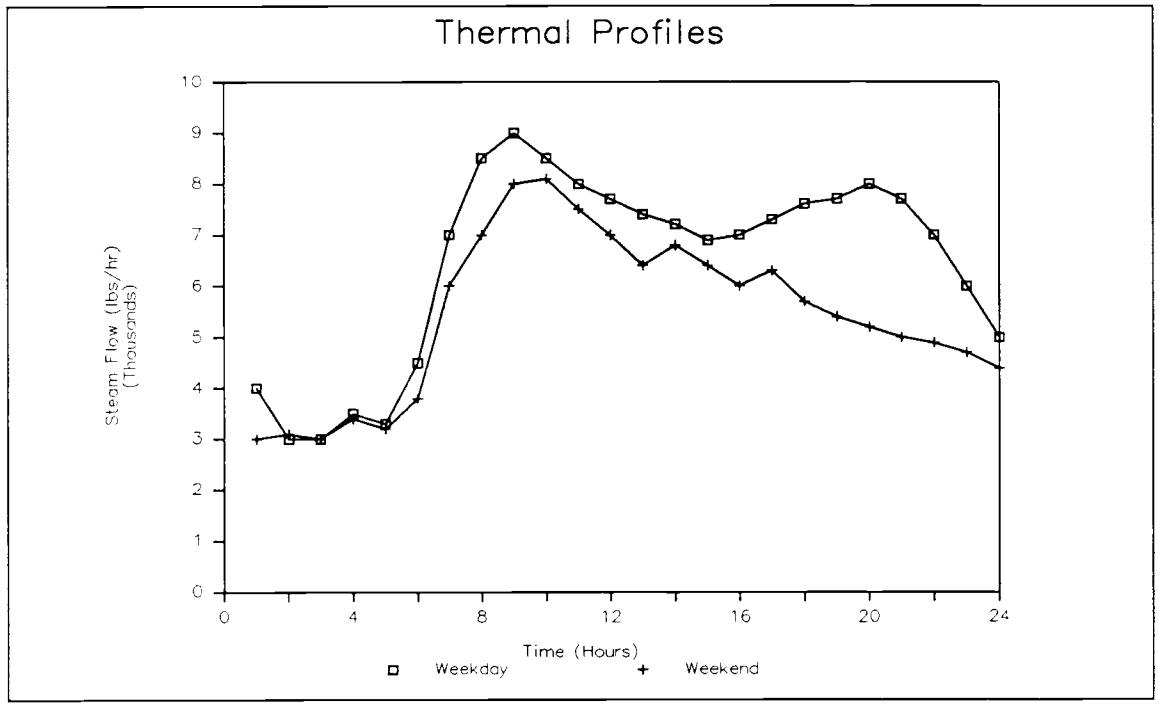

Figure 2. Typical hourly thermal profile.

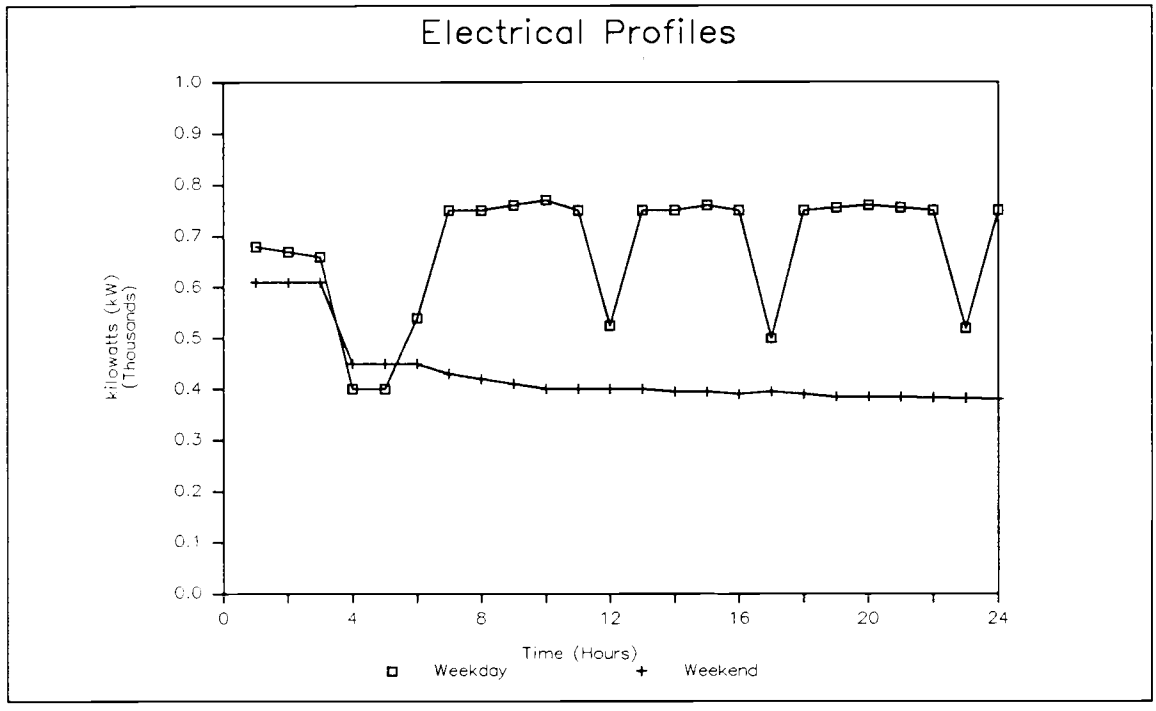

Figure 3. Typical hourly electrical profile.

The electrical load profile can usually be obtained with the help of the electric utility. Most utilities have power metering equipment which can be connected to the main power line to the plant. An alternative is to read the electric meter hourly for each typical day.

Hourly steam profiles are more difficult to obtain unless steam flow is metered in the plant. However, many equipment operators can estimate steam flow based on their experience and knowledge of the facility.

3. UTILITY RATE SCHEDULES

Cogeneration feasibility depends strongly on the cost of purchasing electrical power from the local utility. Because cogeneration replaces some or all of the power purchased from the utility, the value of the purchased power must be determined.

The utility's charge to the customer generally consists of two parts: one part fixed by the maximum power in kilowatts taken at any time during a certain definite period, and another part fixed by the total energy used during the same period [6]. In addition, a flat service charge is usually added on.

If the cogeneration system produces more electricity than the plant needs, the utility is required to purchase it. The value of this electrical energy is known as the "avoided cost." Utilities are required to file their avoided costs with a state agency.

Discussion of utility rate schedules is divided into four sections:

- Basic Charge
- Demand Charge
- Energy Charge
- Avoided Cost

3.1 Basic Charge

This monthly charge, sometimes called the service or customer charge, pays for fixed utility costs. It is independent of energy and includes operation, maintenance, and administrative costs for metering and billing. It is generally, but not always, a fixed cost based on transformer

size and whether the service provides single- or three-phase power [7]. In the simulation, the basic charge is treated as a constant cost.

3.2 Demand Charge

The utility meters power demand of its larger commercial and industrial customers. The charge is based on the highest (peak) demand metered over a specific time period (generally 15 or 30 minutes) during the billing period. The rates are often broken into blocks of demand where the first block is charged at one rate and the second (and third) blocks are charged at another. For example, the demand rates might be $0.50 per kilowatt for the first 50 kilowatts and $3.50 per kilowatt in excess of 50.

Seasonal variations also occur in the demand rates. The model will accept three block definitions for demand and also winter and summer variations. For the simulation, April through September are considered summer months and October through March are winter months.

Some utilities also charge for demand according to time-of-day to encourage customers to operate equipment during periods of low power demand on the utility. This type of load shifting helps to distribute the power demand and enables the utility to operate its generation equipment more consistently and efficiently. For example, an on-peak period might be defined as Monday through Friday, 6:00 A.M. to 10:00 P.M.. The rest of the week would be considered off-peak.

The on-peak demand charge will be higher than the off-peak demand charge. Typically, the off-peak charge is only applied to off-peak demand in excess of the on-peak demand.

Although there may be some additional cost savings in using time-of-day rates for the cogeneration analysis, time-of-day rate scheduling is not included in the model. The result will be a more conservative economic analysis.

3.3 Energy Charge

Energy consumption, like demand, is often split into blocks, with different rates for different blocks. For example, the energy rates might be $0.034/kWh for the first 10,000 kWh used in the billing period, $0.031/kWh for the next 17,000 kWh used, and $0.027/kWh for any additional kWh. Energy rates may also vary seasonally. The model will accept three block definitions for energy consumption and also winter and summer variations.

3.4 Avoided Cost

This is the price (in dollars per kilowatt-hour) the cogenerator receives for excess electricity sold to the utility.

4. EQUIPMENT

The two primary pieces of equipment for a cogeneration system are the boiler and the steam turbine-generator.

An existing boiler and/or steam turbine can be used in the cogeneration analysis. The rating of existing equipment determines if it can be used effectively for power production in the facility. The program allows the user to specify an existing boiler and/or turbine or new equipment.

4.1 Boiler

There are several boiler options for a cogeneration system. The options depend on whether an existing boiler is used or if a new boiler will be installed. The proposed boiler system will be sized to meet the facility's peak thermal load.

Existing boiler. The following modifications may be available for existing boilers:

A. Increase the operating pressure to the maximum rated pressure (if the boiler is not already at maximum pressure).

B. Add an additional boiler to increase power generation capacity.

New Boiler. A new boiler will be sized to meet the peak thermal load with additional capacity to meet some portion of the plant electrical load.

4.2 Turbine-Generator

Turbine steam pressure and temperature must be specified at the following locations:

- Turbine Inlet
- Turbine Outlet(s)

There are many types of turbines available for power generation. In general, however, the turbine can be one of four types:

Condensing. The turbine exhausts to a vacuum (typically between 2 and 4 inches of mercury absolute pressure) to increase electrical production. The cost for higher efficiency is that the low quality exhaust steam is usually wasted. Approximately 65% of the available energy in the steam is lost in the condenser at approximately 100°F. The facility's steam load is met with a parallel steam line. This option is not true cogeneration since the generation of both thermal and electrical power is not from a single fuel source.

Condensing with autoextraction. The boiler produces steam at a pressure higher than the facility requires. The pressure is reduced through the initial stages of the turbine. Process steam is extracted at the required pressure. Any additional steam continues to the turbine exhaust. This is a more efficient option than the straight condensing turbine but energy is still wasted at the turbine exhaust.

Backpressure (Non-condensing). All of the steam used in the facility passes through the turbine. However, the turbine is designed so that the exhaust is at the correct pressure to be used in the facility. This is the most

efficient system because steam energy not extracted in the turbine is used as process heat. The disadvantage is that turbine steam flow and therefore electrical generation depend on the steam load. An alternative is to keep steam flow constant by adding a condenser. This will help keep the electrical generation constant but heat will be wasted in the condenser.

Backpressure with autoextraction. This turbine operates similar to a condensing turbine with autoextraction but the exhaust from the turbine is still usable as process steam. This type of turbine is most appropriate for facilities that need steam at intermediate pressures.

Combinations. The use of multiple turbines of one or more types is not unusual in cogeneration systems. This allows more flexibility in trying to match the thermal and electrical load requirements with power production. The model, however, will simulate performance of only one turbine at a time.

Each of these turbine types can be simulated by the model. The model uses a database of turbine performance characteristics for sizes ranging from 500 kW to 7500 kW. If an extraction turbine is selected, it is assumed that the process steam flow will be extracted, not exhausted, from the turbine.

5. OPERATION

Cogeneration systems typically operate in one of three modes depending on the size and application of the system:

Constant Loading. The cogeneration system is operated at a constant level to meet a base loading condition, either electrical or thermal, or operated at maximum boiler or turbine capacity.

Electrical Load Following. The system is operated to follow the electrical demands of the plant. Excess steam is wasted or sold. Steam shortages are made up by an auxiliary boiler or parallel steam line.

Thermal Load Following. The system is operated to follow the thermal demands of the plant. Excess electrical power is sold. Additional electrical power required by the plant is purchased from the utility.

Extraction turbines are, by design, thermal load following. Variation can occur in how much power is produced over and above that produced by the extraction flow.

6. SYSTEM PERFORMANCE

Overall cogeneration system performance depends upon the behavior of the boiler and turbine under full and part load conditions. Although the most cost effective systems will typically operate at or near full load, system performance under partial loading must be considered when comparing systems.

6.1 Steam Rates

A steam turbine is typically evaluated using a characteristic known as the steam rate (SR). The steam rate is defined as the amount of steam (in pounds per hour) required by the turbine to produce a specified unit of power. The steam rate is usually expressed as pounds of steam per hour per kilowatt (lb/kW-hr) or pounds of steam per hour per horsepower (lb/hp-hr).

The theoretical steam rate (TSR) is the steam rate for a 100% efficient turbine. The actual steam rate (ASR) for a turbine is greater because of losses that occur during the conversion of available steam energy into mechanical work. Theoretical steam rates are calculated based upon the inlet and exhaust steam enthalpy assuming an isentropic expansion across the turbine. The enthalpy of the steam (Btu/lb) at these two points can be found in steam tables or on a Mollier diagram.

The difference in enthalpy between the inlet and exhaust can be converted into kilowatt-hours or horsepower-hours and expressed as the steam rate. Figure 4 is a plot of enthalpy (h) versus entropy (s) and represents the expansion of steam through a turbine. The lines P1 and P2 represent the turbine inlet and outlet steam pressures,

respectively. The ideal expansion is represented by the line 1-2. Line 1-3 represents the actual expansion through the turbine. Both the theoretical and actual steam rates can be determined by drawing an enthalpy-entropy diagram for the operating conditions of a turbine.

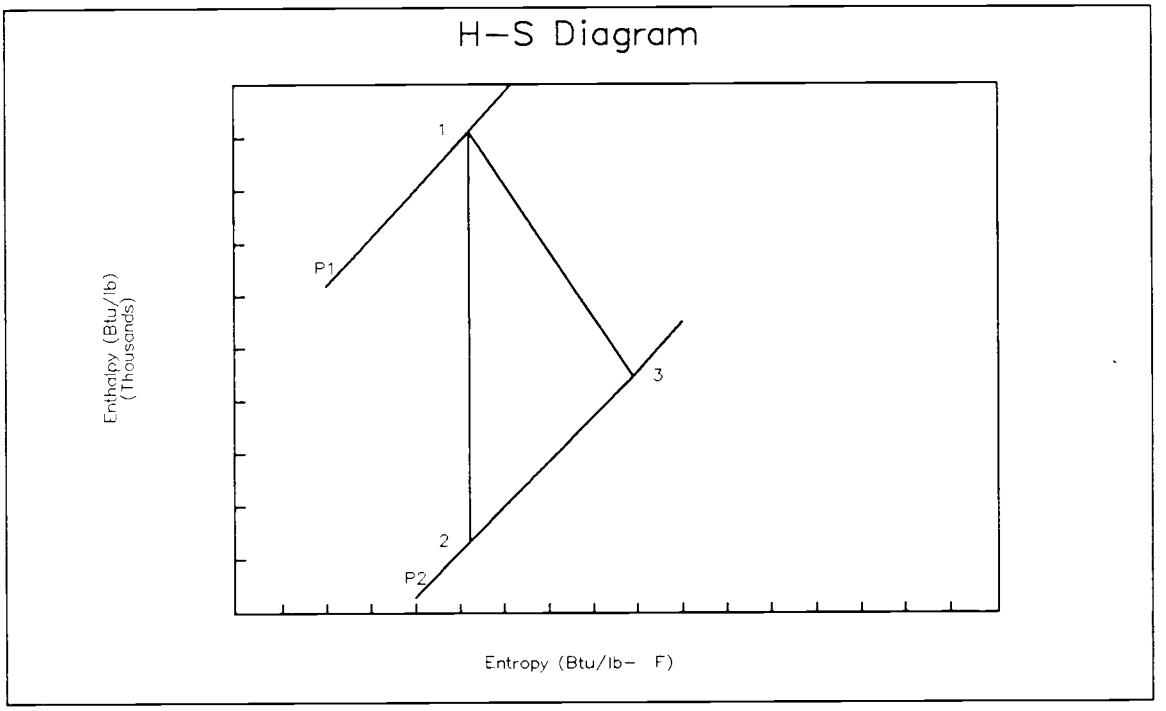

Figure 4. Typical expansion process across a steam turbine on an enthalpy-entropy diagram.

In equation form, the calculation is

$$\begin{aligned} TSR &= 3{,}413 \text{ Btu/kW-hr} \;/\; (h_1 - h_2) \quad \text{(lb/kW-hr)} \\ &= 2{,}545 \text{ Btu/hp-hr} \;/\; (h_1 - h_2) \quad \text{(lb/hp-hr)} \end{aligned} \tag{9}$$

and

$$\begin{aligned} ASR &= 3{,}413 \text{ Btu/kW-hr} \;/\; (h_1 - h_3) \quad \text{(lb/kW-hr)} \\ &= 2{,}545 \text{ Btu/hp-hr} \;/\; (h_1 - h_3) \quad \text{(lb/hp-hr)} \end{aligned} \tag{10}$$

where

h_1 = Steam enthalpy at inlet conditions

h_2 = Steam enthalpy at exhaust conditions based on an isentropic expansion
h_3 = Steam enthalpy at exhaust conditions based on actual expansion

The efficiency of a turbine (η_T) is the ratio of the theoretical steam rate to the actual steam rate.

$$\begin{aligned} \eta_T &= \text{Theoretical steam rate / Actual steam rate} \\ &= [3{,}413 / (h_1 - h_2)] / [3{,}413 / (h_1 - h_3)] \\ &= (h_1 - h_3) / (h_1 - h_2) \end{aligned} \tag{11}$$

6.2 Performance Charts

A typical plot of steam flow versus power output for condensing or noncondensing non-extraction turbines is shown in Figure 5. This curve is known as the "Willans Line." When throttle governing at constant speed is used, it is substantially straight from no load to full load for almost all turbines regardless of type [8]. Both steam flow and turbine output are expressed as a percent of full load flow and power, respectively.

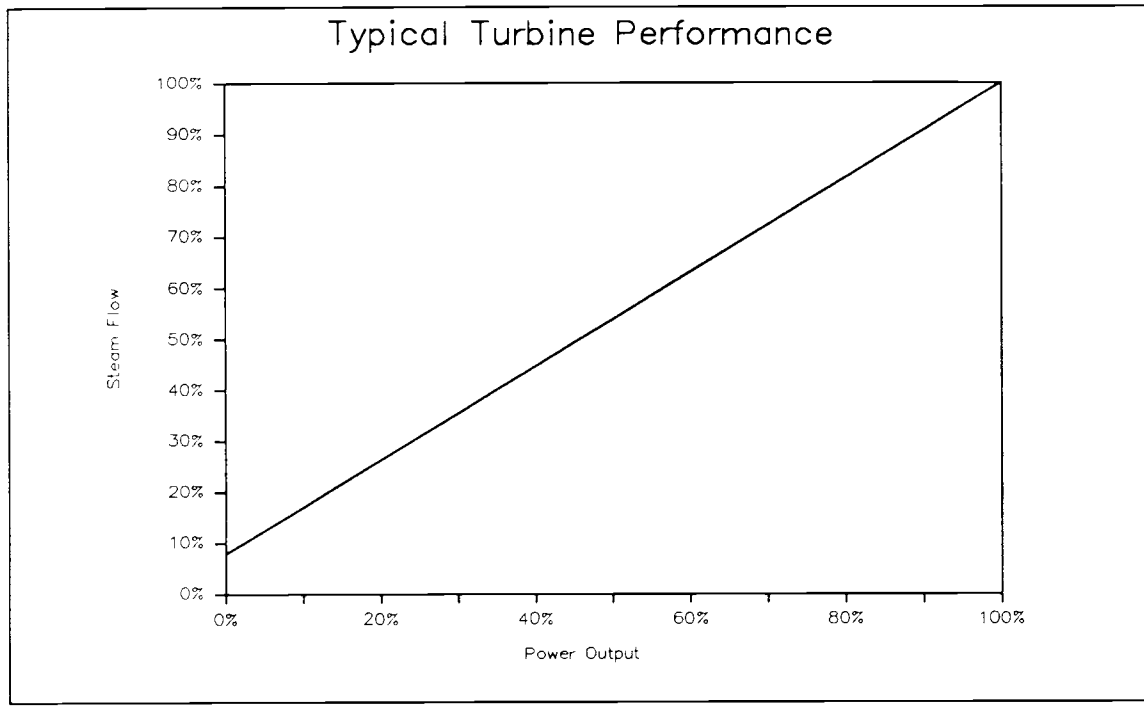

Figure 5. Throttle flow versus output.

The steam flow at no-load is required to overcome the no-load losses of the turbine-generator set. It typically varies from 5% to 30% of the full load flow depending on the type of turbine.

A plot of actual steam rate (ASR) versus turbine output can be generated using a performance curve like Figure 5. For several operating points on the Willans line the steam flow is divided by the power output to obtain the actual steam rate (see Table 5).

$$\text{ASR} = \text{Steam flow} / \text{Power output} \tag{12}$$

Table 5. Results of calculating the actual steam rate from Figure 5.

Output (A)	Steam Flow (B)	Steam Rate (B/A)
10.0%	17.1%	171.1%
20.0%	26.3%	131.6%
30.0%	35.5%	118.4%
40.0%	44.7%	111.8%
50.0%	53.9%	107.9%
60.0%	63.2%	105.3%
70.0%	72.4%	103.4%
80.0%	81.6%	102.0%
90.0%	90.8%	100.9%
100.0%	100.0%	100.0%

These values are then plotted against turbine power output. The steam rate, as a percentage of full load steam rate, is shown in Figure 6.

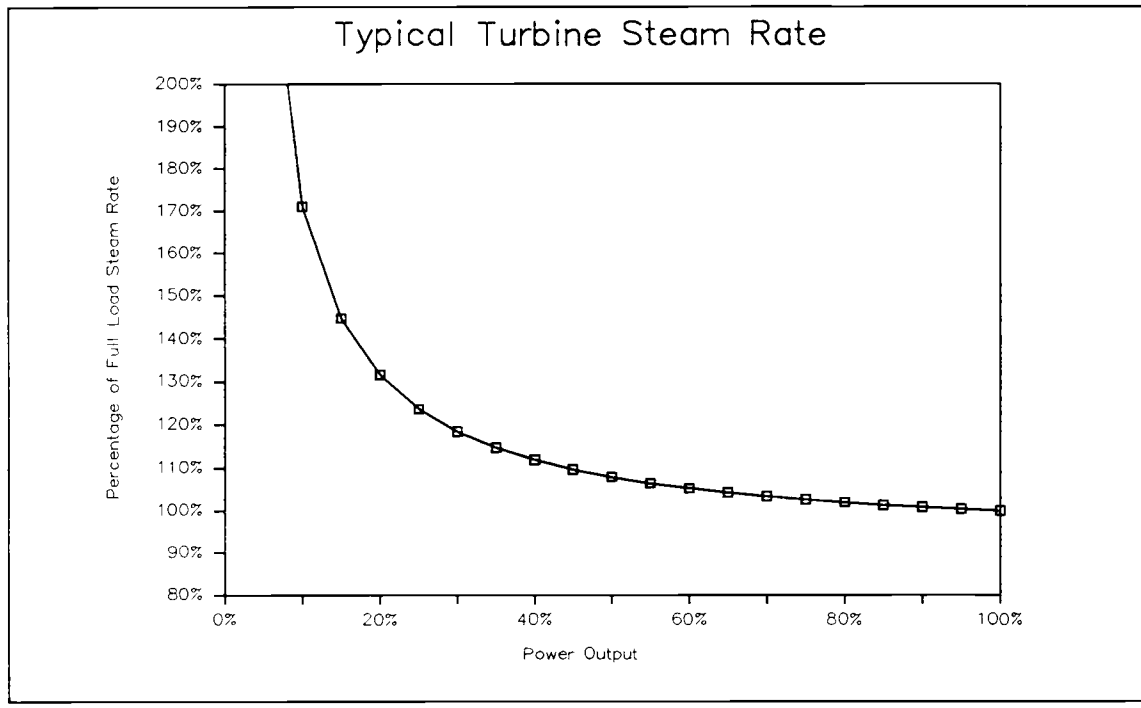

Figure 6. Steam rate versus output.

Similarly, the throttle flow versus turbine output can be determined from steam rate measurements under varying

loads by multiplying the steam rates by the corresponding outputs.

The steam rate provides sufficient information to determine cogeneration performance in terms of steam (and therefore costs) required for varying power production. Since turbine efficiency is the ratio of TSR/ASR, it can be calculated, as a function of turbine loading, from the operating conditions of the turbine (which lead to the TSR) and a curve of the actual steam rate (Figure 6). The Willans Line, steam rate, and efficiency curves are shown in Figure 7 for a 2000 kW turbine operating at 600 psig inlet pressure, 125 psig exhaust pressure. The theoretical steam rate is 22.9 lb/kW-hr.

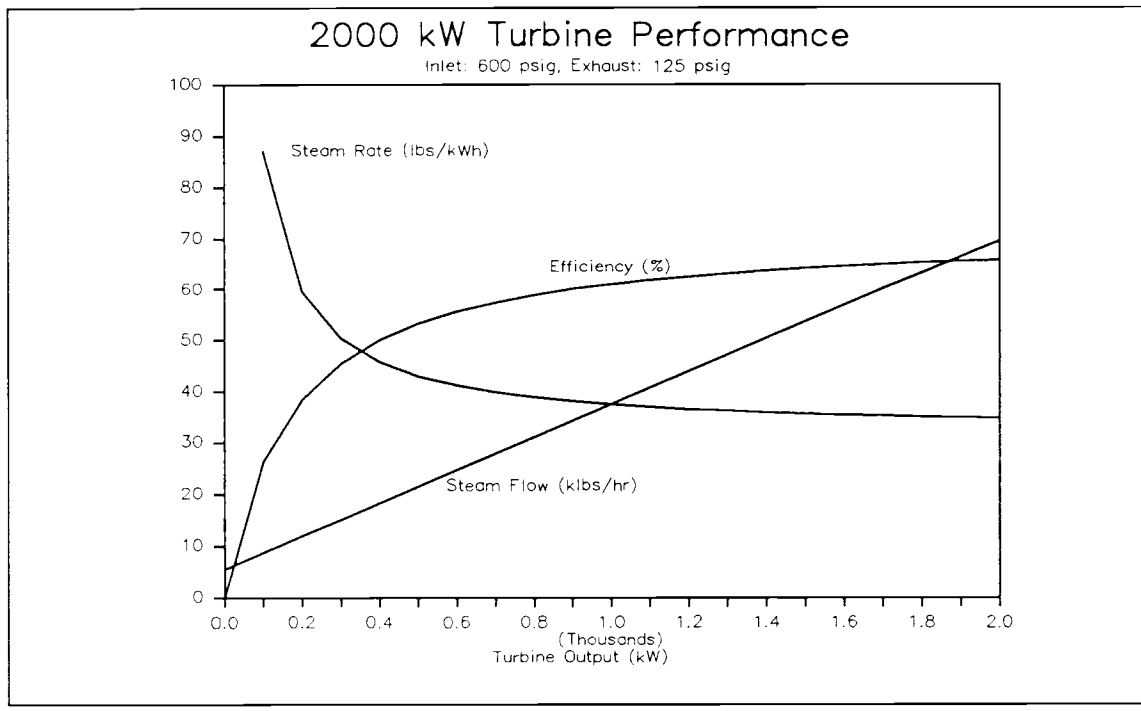

Figure 7. Performance curves for a 2000 kW turbine, 600 psig inlet, 125 psig exhaust.

As expected, maximum efficiency is achieved at full load operation. Note, however, that the turbine maintains relatively high efficiency from one-quarter to full load.

To construct these types of performance charts it is necessary to know at least two points from which to construct the Willans Line. Plots are available of full and half load TSR factors (defined as 1/efficiency) [6] for both condensing and noncondensing turbines ranging in size from 500 kW to 7500 kW. The TSR factors plotted are specifically for inlet pressures of 600 psia and 200 psia. Values for other operating pressures are interpolated by the model. Values of TSR factors for noncondensing turbines are shown in Table 6. Complete tables of the TSR factors are included in the appendix.

Table 6. Theoretical steam rate factors for noncondensing turbines (80% power factor) [6].

Turbine	Full Load		Half Load	
kW	200 psia	600 psia	200 psia	600 psia
500	1.700	1.890	2.060	2.230
750	1.600	1.760	1.930	2.055
1000	1.540	1.680	1.845	1.950
1500	1.470	1.580	1.750	1.821
2000	1.425	1.520	1.690	1.750
3000	1.380	1.440	1.630	1.670
4000	1.355	1.412	1.593	1.622
5000	1.345	1.390	1.572	1.590
6000	1.335	1.375	1.560	1.570
7500	1.330	1.360	1.550	1.560

Using the theoretical steam rate, the theoretical steam rate factors for full and half load, and the turbine rating, a complete set of performance charts can be constructed to predict system performance for all normal operating conditions.

6.3 Non-extraction Turbine Performance

The governing equation for the power (P) in kilowatts produced by a non-extraction steam turbine is based on the first law of thermodynamics. It can be written as

$$P = M_{THR} (h_{THR} - h_{EXH}) \eta / 3{,}413 \text{ Btu/kW-hr} \tag{13}$$

where

M_{THR} = Steam flow entering turbine: lb/hr
h_{THR} = Steam enthalpy at turbine inlet: Btu/lb
h_{EXH} = Steam enthalpy at turbine exhaust: Btu/lb
η = Turbine-generator efficiency
= $\eta_T \eta_G$
η_T = Turbine efficiency
η_G = Generator efficiency

This equation is often written utilizing the theoretical steam rate as follows

$$P = M_{THR} \eta / TSR \tag{14}$$

Rearranging to get the steam flow in terms of electrical power yields

$$M_{THR} = P \, TSR / \eta \tag{15}$$

Equations (14) and (15) are used to calculate system performance under any loading conditions using either steam flow or turbine power output as the independent variable.

As noted earlier, cogeneration systems are typically operated in one of three modes; electrical load following, thermal load following, or constant load operation. The approach to modelling non-extraction turbines in each of the three operating modes follows.

Electric Load Following. To model turbine performance operating in the electrical load following mode, it is necessary to determine an equation for the Willans line. Using Newman's TSR factors, the full and half load operating points can be calculated and a straight line equation can be determined. In general, a linear approximation is of the form

$$M = m P + b \tag{16}$$

where

M = Throttle steam flow: lb/hr
m = Slope of the Willans Line
P = Turbine output: kW
b = No load steam flow (y-intercept): lb/hr

The slope of the line is the change in steam flow divided by the corresponding change in turbine output and can be calculated from the full and half load operating points.

$$m = (M_{FULL} - M_{HALF}) \ / \ (P_{FULL}/2) \tag{17}$$

where

M_{FULL} = Full load steam flow: lb/hr
M_{HALF} = Half load steam flow: lb/hr
P_{FULL} = Full load turbine rating: kW

Using the theoretical steam rate and TSR factors for full and half load, the slope of the curve can be rewritten as

$$\begin{aligned} m &= [(TSR\ P_{FULL}/\eta_{FL}) - TSR\ (P_{FULL}/2)/\eta_{HL}] \ / \ (P_{FULL}/2) \\ &= TSR\ (2\ \eta_{HL} - \eta_{FL}) \ / \ (\eta_{FL}\ \eta_{HL}) \end{aligned} \tag{18}$$

where

η_{FL} = Full load turbine efficiency
= 1 / Full load TSR factor
η_{FL} = Half load turbine efficiency
= 1 / Half load TSR factor

The y-intercept (b) can be solved for by rearranging the equation and substituting for P and M at a known point (e.g., full load operation).

$$\begin{aligned} b &= M - m\,P \\ &= M_{FULL} - m\,P_{FULL} \\ &= (TSR\ P_{FULL}/\eta_{FL}) - (m\,P_{FULL}) \end{aligned} \tag{19}$$

Further substitution for the slope, m, yields the following expression for the y-intercept.

$$b = (TSR\ P_{FULL}/\eta_{FL})\ [(\eta_{FL} - \eta_{HL}) / \eta_{HL}] \tag{20}$$

The complete equation for non-extraction, electrical load following turbine performance can now be written as

$$\begin{aligned} M &= m\,P + b \\ &= [TSR\ (2\,\eta_{HL} - \eta_{FL}) / (\eta_{FL}\,\eta_{HL})]\ P + \\ &\quad \{(TSR\ P_{FULL}/\eta_{FL})\ [(\eta_{FL} - \eta_{HL})/\eta_{HL}]\} \end{aligned} \tag{21}$$

For a given electrical load profile, the required steam flow can be estimated using equation (21).

Thermal Load Following. When operating in the thermal load following mode, power production depends upon the steam demands on the cogeneration system. The model of a thermal load following non-extraction turbine is very similar to the electric load following model. The primary difference is that electrical power is the dependent variable and steam flow is the independent variable. Again, the straight line

approximation is used and the slope and y-intercept terms are determined.

The complete equation for non-extraction, thermal load following turbine performance can be written as

$$P = [(\eta_{FL}\ \eta_{HL}) / TSR\ (2\ \eta_{HL} - \eta_{FL})]\ M + P_{FULL}\ [(\eta_{HL} - \eta_{FL}) / (2\ \eta_{HL} - \eta_{FL})] \quad (22)$$

For a given thermal load profile, the required steam flow can be estimated using equation (22).

Constant Load Operation. Equations (21) and (22) for electrical and thermal load following operation are also used to model constant load. If the constant load to be met is electrical, equation (21) is used and the electrical load is the independent variable. If the constant load to be met is thermal, equation (22) is used and the thermal load is the independent variable.

6.4 Extraction Turbine Performance

A primary feature of autoextraction steam turbines for cogeneration systems is their flexibility in comparison to straight condensing or non-condensing steam turbines. They can supply varying demands for thermal and electrical energy whether these demands vary singly or simultaneously. The model simulates performance of both condensing and non-condensing single-automatic-extraction steam turbines.

Single-automatic-extraction steam turbines are similar to straight condensing or non-condensing turbines except that they are designed to allow for extraction of steam at one or more points. The pressure at these points is

controlled to maintain a pressure lower than the turbine inlet pressure but higher than the exhaust pressure. The controls allow the turbine to maintain thermal or electrical loading when the demand for extraction steam varies. This is done by varying the steam flow to the condenser to maintain the variable relation between the extraction steam flow and the turbine output.

Autoextraction turbines are inherently thermal load following; steam is extracted to meet the thermal requirements of a process. These turbines can be operated in both an electrical load following and constant load mode. The model assumes that the thermal load will be met by the extraction flow. The remaining steam flows to the turbine exhaust. In the program, turbine performance is modelled using the shorthand estimation method described by Newman in a series of articles in POWER PLANT ENGINEERING magazine [9].

> "The method is an empirical one that is based on the principle of estimating accurately the factors that influence greatly the result, and approximating roughly the factors that have little influence on the result [10]."

The following discussion will explain the method and show how to estimate performance graphically, based directly on Newman's method. This estimating method has been automated in the computer program. Many of the steps, which are outlined below, are performed by the program.

The governing thermodynamic equations for the performance of an autoextraction steam turbine-generator system can be written as

$$P = \{[M_{EXT}\ (h_{THR} - h_{EXT})\ \eta_{T1}] + [M_{EXH}\ (h_{THR} - h_{EXH})\ \eta_{T2}]\}\ \eta_G\ /\ 3{,}413 \qquad (23)$$

and

$$M_{THR} = M_{EXT} + M_{EXH} \tag{24}$$

where

kW = Generator output
M_{THR} = Steam flow at turbine inlet: lb/hr
M_{EXT} = Steam flow extracted from the turbine: lb/hr
M_{EXH} = Steam flow exhausted from the turbine: lb/hr
h_{THR} = Steam enthalpy at turbine inlet: Btu/lb
h_{EXT} = Steam enthalpy at turbine extraction: Btu/lb
h_{EXH} = Steam enthalpy at turbine exhaust: Btu/lb
η_{T1} = Turbine stage efficiency between inlet and extraction point
η_{T2} = Turbine stage efficiency between inlet and exhaust point

Assuming that the stage efficiencies are equal, and calling this efficiency η_T, the governing equation can be rewritten in terms of the theoretical steam rates as

$$P = [(M_{EXT}/TSR_2) + (M_{EXH}/TSR_1)]\ \eta \tag{25}$$

where

TSR_1 = 3,413 Btu/kW-hr / $(h_{THR} - h_{EXH})$
TSR_2 = 3,413 Btu/kW-hr / $(h_{THR} - h_{EXT})$
η = Turbine-generator efficiency
= $\eta_T\ \eta_G$

The governing equation can be rearranged to express the inlet steam flow in terms of the turbine output and the extraction flow.

$$M_{THR} = (P \times TSR_1/\eta) + M_{EXT} \times [1 - (TSR_1/TSR_2)] \tag{26}$$

The term $[1 - (TSR_1/TSR_2)]$ in equation (26) will be referred to as the theoretical extraction factor (TEF). It can be obtained from an enthalpy-entropy diagram for an extraction turbine.

Figure 8 shows the total theoretical isentropic enthalpy drop from turbine inlet (point 1) to turbine exhaust (point 2). The lines P1, P2, and P3 represent the inlet, extraction, and exhaust steam pressure in the turbine. The enthalpy drop from point 1 to point 2 consists of two parts. The first part, $h_1 - h_4$, is the difference in enthalpy between the inlet and extraction points. The second part, $h_4 - h_2$, is the enthalpy drop between the extraction point and the turbine exhaust.

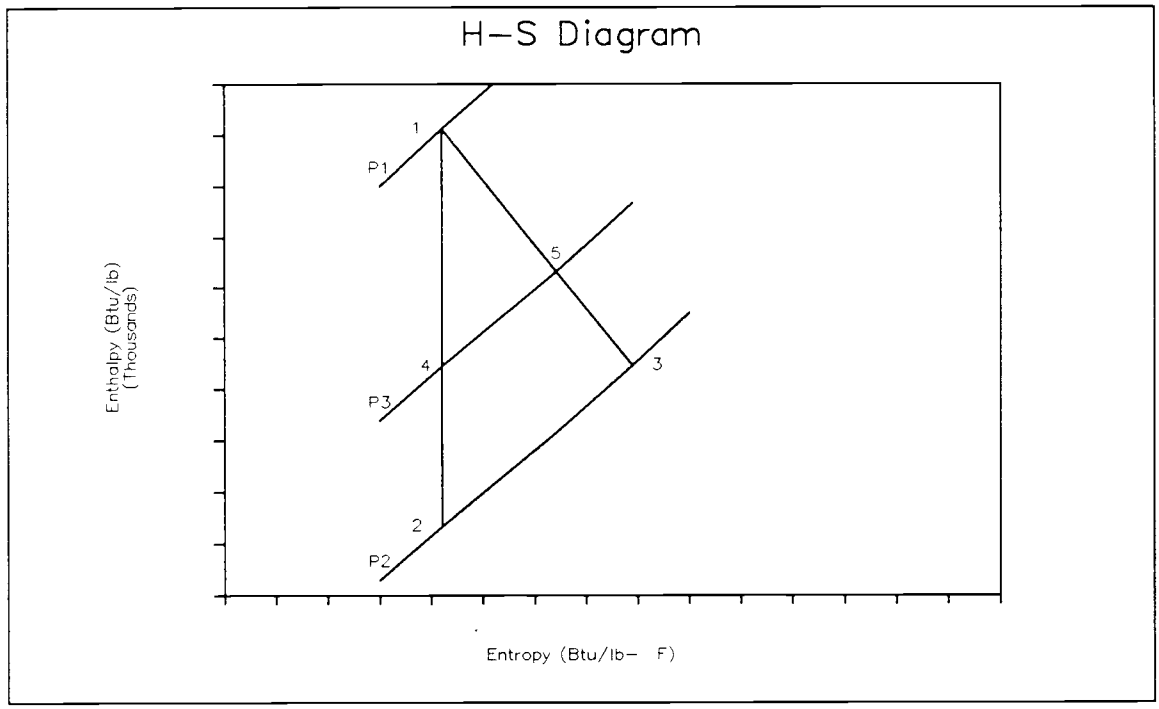

Figure 8. Typical expansion process across an extraction steam turbine on an enthalpy-entropy diagram.

The quantity $h_1 - h_4$ represents the amount of theoretical work in Btu/lb available from each pound of steam between the point of entry at the throttle and the point of extraction. For each pound of steam extracted the loss of energy available for generation resulting from the extraction is given by $h_4 - h_2$.

To maintain a constant electrical load on the turbine, additional steam must be added to the throttle flow to account for this loss of work. The theoretical extraction factor is the amount of steam (in pounds) which must be added to the throttle flow for each pound of steam extracted from the turbine. The theoretical extraction factor can calculated from

$$\begin{aligned} TEF &= [(h_1 - h_4) - (h_4 - h_2)] / (h_1 - h_4) \\ &= 1 - [(h_4 - h_2)/(h_1 - h_4)] \\ &= 1 - (TSR_1/TSR_2) \end{aligned} \tag{27}$$

The theoretical extraction factor must be modified for actual turbines. The actual extraction factor (EF) is shown in Figure 9 versus (TSR_1/TSR_2) for condensing single automatic extraction turbines [8]. A similar curve applies to noncondensing turbines.

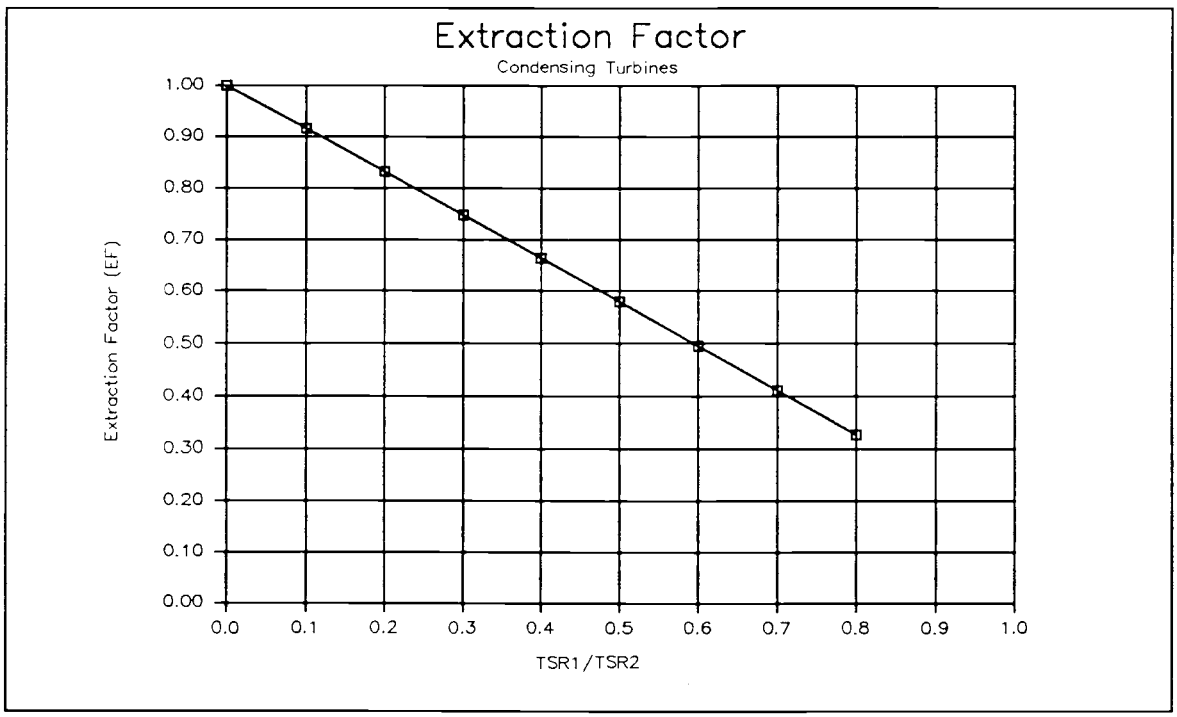

Figure 9. Extraction factor versus TSR_1/TSR_2 for condensing single automatic extraction steam turbines.

The extraction factor can be written as a function of the ratio of theoretical steam rates with a correction factor (C).

$$EF = 1 - C\ (TSR_1/TSR_2) \tag{28}$$

From the plots of EF versus TSR_1/TSR_2 given by Newman, C has been calculated to be 0.841 for condensing turbines and 0.902 for noncondensing turbines.

Therefore, the governing equation can be written

$$\begin{aligned} M_{THR} &= (P\ TSR_1/\eta) + M_{EXT}\ [1 - C\ (TSR_1/TSR_2)] \\ &= (P\ TSR_1/\eta) + M_{EXT}\ EF \end{aligned} \tag{29}$$

This equation is used to generate performance charts for single-automatic-extraction turbine-generator systems. Note that equation (29) is in the electrical load following form (i.e., power output is the independent variable and steam flow is the dependent variable). For the case of constant electrical loading, the power output is fixed and the throttle steam flow is calculated, fixed by the extraction steam flow requirement and the constant power output. Though the extraction flow may vary, the power output will stay constant by controlling the throttle steam flow.

7. EXAMPLE OF EXTRACTION TURBINE PERFORMANCE

To demonstrate the modelling method, an example will be presented. This will use the performance equations and show the limits on the operation of an autoextraction steam turbine. The tables and graphs are taken from Newman's articles [8]. The example will outline the steps required to construct a throttle flow versus power performance chart. Any mention of drawing parts of the chart refers to constructing the chart on paper. The model, however, performs these steps automatically during its performance simulation.

For the example, a throttle flow versus power performance chart will be constructed for a system with the following design parameters:

- 2500 kW, 80% Power Factor, 3125 kVA, 60 cycle condensing, single automatic extraction steam turbine generator set.

Steam Conditions:

- 600 psig, 600°F main (inlet) pressure and temperature.
- 2" Hg, abs (1 psia) exhaust pressure
- 150 psig extraction pressure
- 40,000 lb/hr maximum extraction flow

The process of constructing the chart consists of the following steps:

Step 1. Determine the steam enthalpies for the turbine inlet, extraction, and exhaust steam conditions from steam tables or Mollier diagram.

h_{INLET} = 1,287 Btu/lb (at turbine inlet)
h_{EXT} = 1,178 Btu/lb (at extraction point)
h_{EXH} = 851 Btu/lb (at turbine exhaust)

Step 2. Calculate the theoretical steam rates for the expansion from inlet to extraction and from inlet to exhaust.

TSR_1 = TSR from inlet to exhaust
= 3,413 Btu/kW-hr / (h_{INLET} - h_{EXH})
= 3,413 / (1,287 - 851)
= 7.83 lb/kW-hr

TSR_2 = TSR from inlet to extraction
= 3,413 Btu/kW-hr / (h_{INLET} - h_{EXT})
= 3,413 / (1,287 - 1,178)
= 31.4 lb/kW-hr

Table 7. Selected full load non-extraction turbine-generator efficiencies and half load factors (HLF) for condensing single-automatic-extraction steam turbines [9].

Rated kW	MAIN PRESSURE (psig) 150	200	250	300	400	600	850	HLF
	EFFICIENCY							
2500	0.700	0.695	0.690	0.685	0.675	0.660		0.580
3000	0.710	0.705	0.700	0.695	0.685	0.670		0.580
3500	0.715	0.710	0.705	0.700	0.690	0.680		0.575
4000	0.720	0.715	0.710	0.705	0.700	0.685		0.575
5000	0.725	0.720	0.715	0.710	0.705	0.695	0.685	0.575
6000	0.735	0.730	0.725	0.720	0.715	0.705	0.695	0.570
7500	0.740	0.735	0.730	0.725	0.720	0.715	0.705	0.570

Step 3. Calculate the full and half load non-extraction steam flows.

The full load nonextraction throttle flow (point A in Figure 12) is calculated from

$$M_{FULL} = (TSR_1)\ P\ /\ \eta_{FL}$$

where

η_{FL} = Full load non-extraction efficiency (Table 7)

Therefore,

$$M_{FULL} = (7.83 \text{ lb/kW-hr})\ 2500 \text{ kW} / 0.66$$
$$= 29{,}660 \text{ lb/hr}$$

The half load nonextraction throttle flow (point B in Figure 12) is calculated next from

$$M_{HALF} = M_{FULL}\ HLF$$

where

HLF = Half load factor (Table 7): 0.580

Therefore,

$$M_{HALF} = (29{,}660 \text{ lb/hr})\ 0.58$$
$$= 17{,}200 \text{ lb/hr}$$

The nonextraction performance line can be drawn by connecting points A and B in Figure 12. The next step is to construct the performance lines for various extraction flows.

Step 4. Calculate the extraction factor based on the theoretical steam rates. Use the constant for condensing turbines.

$$EF = 1 - C_{COND}\ (TSR1/TSR2)$$
$$= 1 - 0.841\ (7.83 \text{ lb/kW-hr} / 31.4 \text{ lb/kW-hr})$$
$$= 0.79$$

Step 5. Calculate the full and half load full-extraction steam flows with equation (29).

$$M_{THR} = (P\ TSR_1/\eta) + (M_{EXT}\ EF)$$

Therefore, at full load with full extraction the boiler must supply the turbine with the steam flow (point C in Figure 12)

$$\begin{aligned} M_{THR} &= [2500\ (7.83\ /\ 0.66)] + (40{,}000)\ 0.79 \\ &= 29{,}660 + 31{,}600 \\ &= 61{,}260\ \text{lb/hr} \end{aligned}$$

And at half load with full extraction the boiler must supply the turbine with the steam flow (point D in Figure 12)

$$\begin{aligned} M_{THR} &= M_{HALF} + (M_{EXT})\ EF \\ &= 17{,}200 + (40{,}000)\ 0.79 \\ &= 48{,}600\ \text{lb/hr} \end{aligned}$$

The performance line for full extraction can be drawn by connecting these two points. The same procedure is followed to determine full and half load performance for intermediate extraction flows.

Step 6. Determine limits of minimum and maximum exhaust steam flow, maximum generator output, and maximum throttle steam flow. The limits are explained in the following paragraphs.

7.1 Limits

Minimum Flow to Exhaust. The minimum turbine exhaust steam flow is limited either by the minimum flow needed to

cool the exhaust stages, or by the minimum flow that will leak around the shaft and through the extraction control valve under its closed condition [11].

Figure 10 is a plot of minimum flow to exhaust versus extraction pressure for several ranges of turbine ratings. This plot is read to the nearest 500 lb/hr.

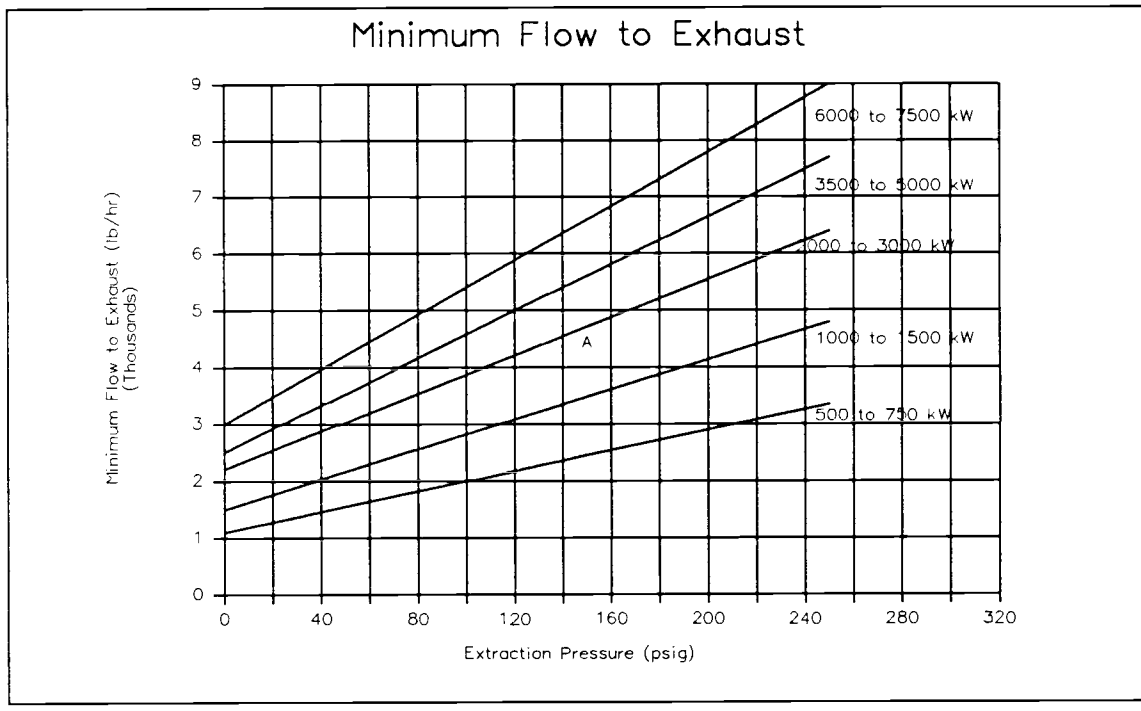

Figure 10. Minimum flow to exhaust versus extraction pressure [9].

This limit is added to the performance chart (Figure 12) by marking the point on the lines of constant extraction where the throttle flow is equal to the extraction flow plus the minimum flow to exhaust as taken from Figure 10. A line is then drawn connecting these points.

For this example, the minimum flow to exhaust from Figure 10 is 5,000 lb/hr (point A in Figure 10). Therefore, the point on the 40,000 lb/hr extraction flow line in Figure 12 will be where the throttle flow is equal to 45,000 lb/hr

(point E); the point on the 20,000 lb/hr extraction flow line will be where the throttle flow is equal to 25,000 lb/hr (point F). The minimum flow line is constructed by connecting points E and F in Figure 12.

The minimum flow to exhaust can also be estimated as the steam flow at no load from the Willans line for the case of no extraction.

Maximum Flow to Exhaust. The limit of maximum flow to exhaust is added in the same way as the minimum flow to exhaust. On each line of constant extraction, the point where the throttle flow is equal to the extraction flow plus the full load non-extraction flow is marked. The lines of constant extraction are extended to include these new points and a new line connecting these points is drawn as in Figure 12. This line represents the upper limit on exhaust steam flow.

In the example, the 20,000 lb/hr extraction flow line will be marked at 49,660 lb/hr (20,000 lb/hr plus the full load non-extraction flow, calculated in Step 3 as 29,660 lb/hr: point G in Figure 12). The limit can be drawn by connecting this point and the full load non-extraction point, points G and A, respectively.

> "In this estimating method the assumption has been made that all turbines will be designed with exhaust sections sufficiently large to enable the turbine to carry full rated output with the extraction pressure held constant but no extraction taken from the turbine. This is the usual practice with autoextraction turbines although cases are occasionally encountered when it is better to make the exhaust section larger or smaller than the general rule [12]."

Maximum Generator Output. A turbine-generator set usually has an 80% power factor (induction) generator and a turbine capable of carrying full kVa on the generator at a power factor of 100%. This operating point will be indicated on the performance chart as 125% capacity or 1.25 times the rated output of the turbine. In the example, this corresponds to an output of 3125 kW.

Maximum Throttle Flow. The maximum throttle flow should not exceed three times the full load non-extraction flow. It usually is taken as the full load throttle flow at the maximum required extraction flow. However, the limit of maximum throttle flow is not rigid. Turbines are built to accept throttle flows in excess of this limit but they are usually of a special design. The model determines this limit based on the maximum required extraction flow. Figure 11 shows maximum throttle flow versus turbine rating for several inlet steam pressures.

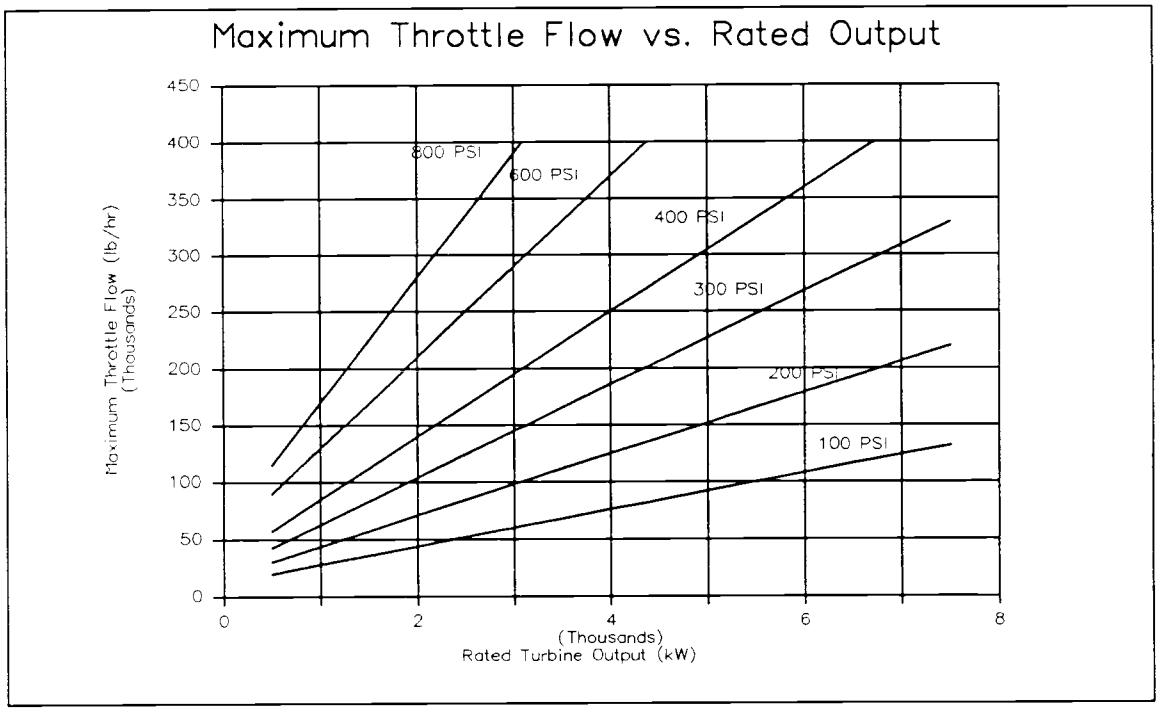

Figure 11. Maximum throttle flow versus rated output. Estimates should be limited to those coming within the maximum throttle flows shown on this chart [9].

In the example, the throttle flow at the maximum required extraction flow (61,260 lb/hr) does not exceed three times the full load non-extraction flow (88,980 lb/hr).

With the limits in place the performance chart (Figure 12) is complete.

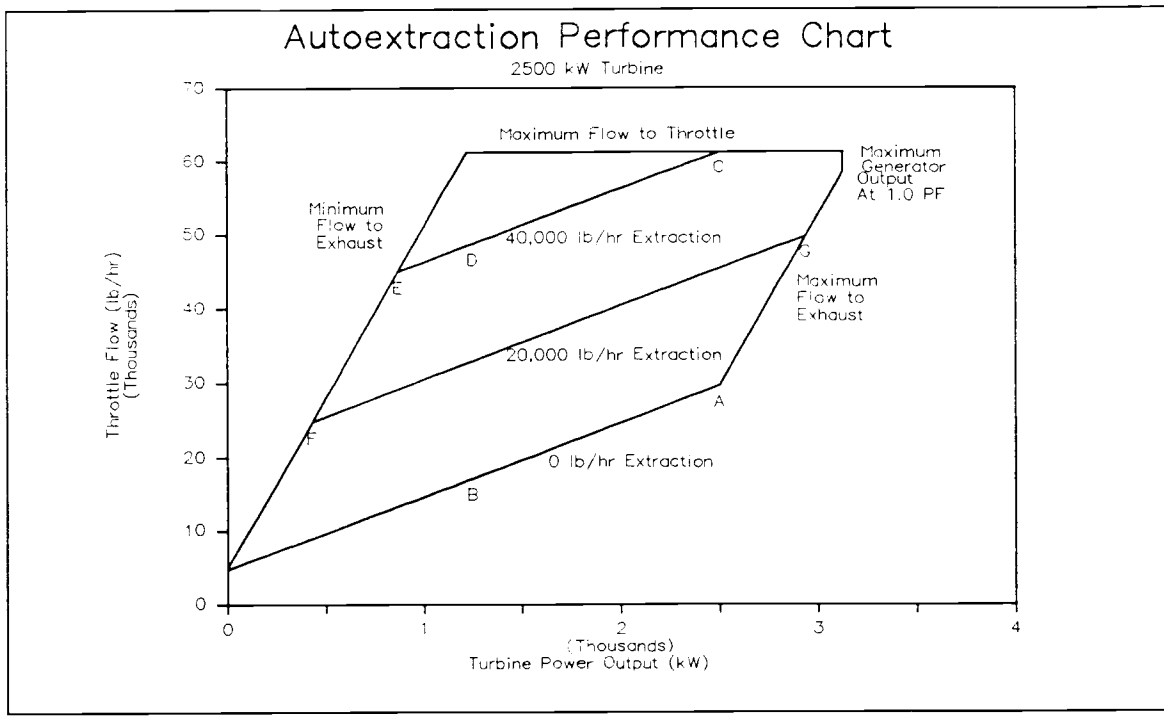

Figure 12. Throttle flow versus output performance chart for a 2500 kW autoextraction steam turbine. Steam conditions: 600 psig, 600°F inlet, 2 inches Hg, Abs. exhaust, extraction at 150 psig.

As a result of the assumptions made in simplifying this approach to turbine performance, Newman recommends not using this method to predict performance at less than half load [8]. Therefore, the model limits the performance of autoextraction turbines to 50% capacity as a minimum operating point.

8. COST ESTIMATION

When a new boiler or turbine-generator is required for the cogeneration application, an estimate of total system cost and installation must be made. Figures 13 and 14 are plots of estimated installed costs of boiler systems and steam turbine-generator sets, respectively. The model uses curvefits to estimate total system capital cost for each application.

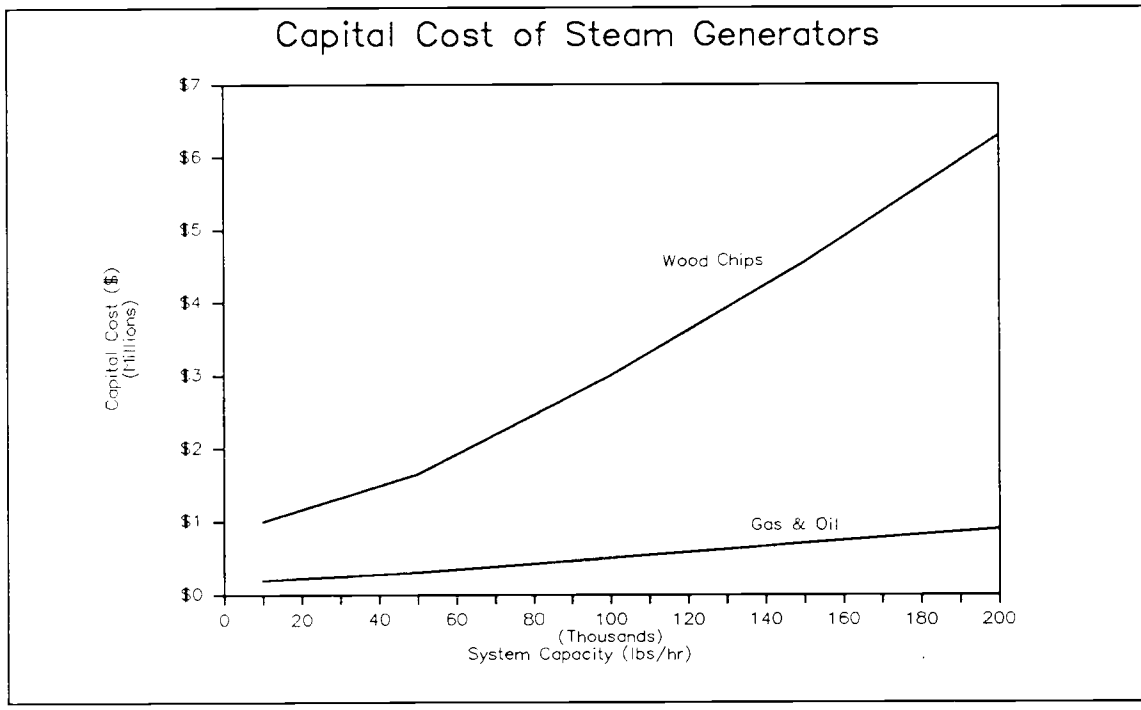

Figure 13. Estimated installed cost of steam generating equipment [13].

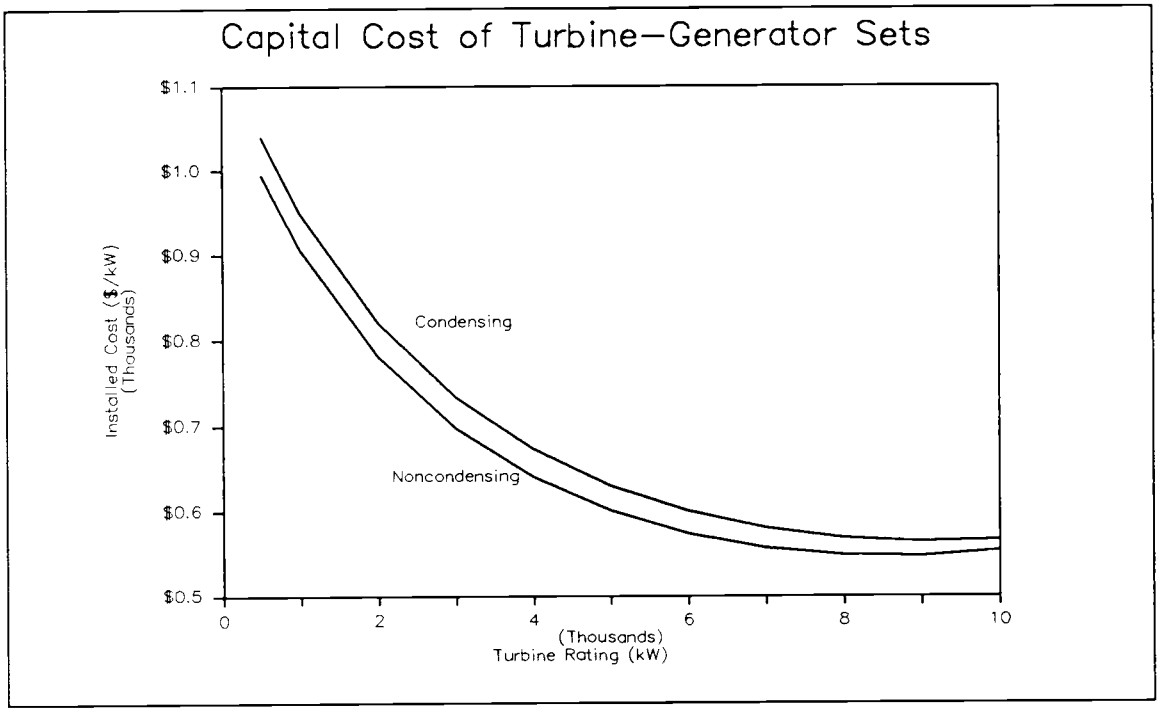

Figure 14. Estimated installed cost of steam turbine-generator sets [14].

9. CASE STUDY

The following study shows the application of the model to a sawmill/veneer plant study to estimate cogeneration feasibility.

Fuels. The mill processes 50% Sugar/Ponderosa Pine and 50% White Pine, each with a 50% wet basis moisture content. The boiler was fired with hogged-fuel produced on-site. No additional fuel was purchased for the boiler. Considerable excess hogged-fuel was produced by the mill. A portion (approximately 15 truckloads per week, 23 tons per truck) of the hogged-fuel was hauled from the site at a cost to the mill of $40 per truckload. With a heating value of 9,000 Btu/lb, the energy value of the excess mill-produced fuel is 150,696 MMBtu/yr. The dollar value of this waste is therefore -$0.19/MMBtu up to the amount of fuel which is currently hauled away. The net annual boiler fuel costs are only those associated with hauling away 16,744 tons from the mill, $29,120/yr. The annual energy consumption of the boiler is approximately 56,440 MMBtu/yr.

The boiler efficiency was not tested and was assumed to be 65% for a properly tuned hogged fuel-fired boiler.

Load Profiles. Steam at 150 psig is provided to the mill for drying veneer and to preheat logs prior to peeling.

The mill operates two shifts, five days per week. The mill operation (and therefore power requirements) was modelled using two day types; weekday and weekend. The estimated electrical load profiles for the two typical days are shown in Figure 15.

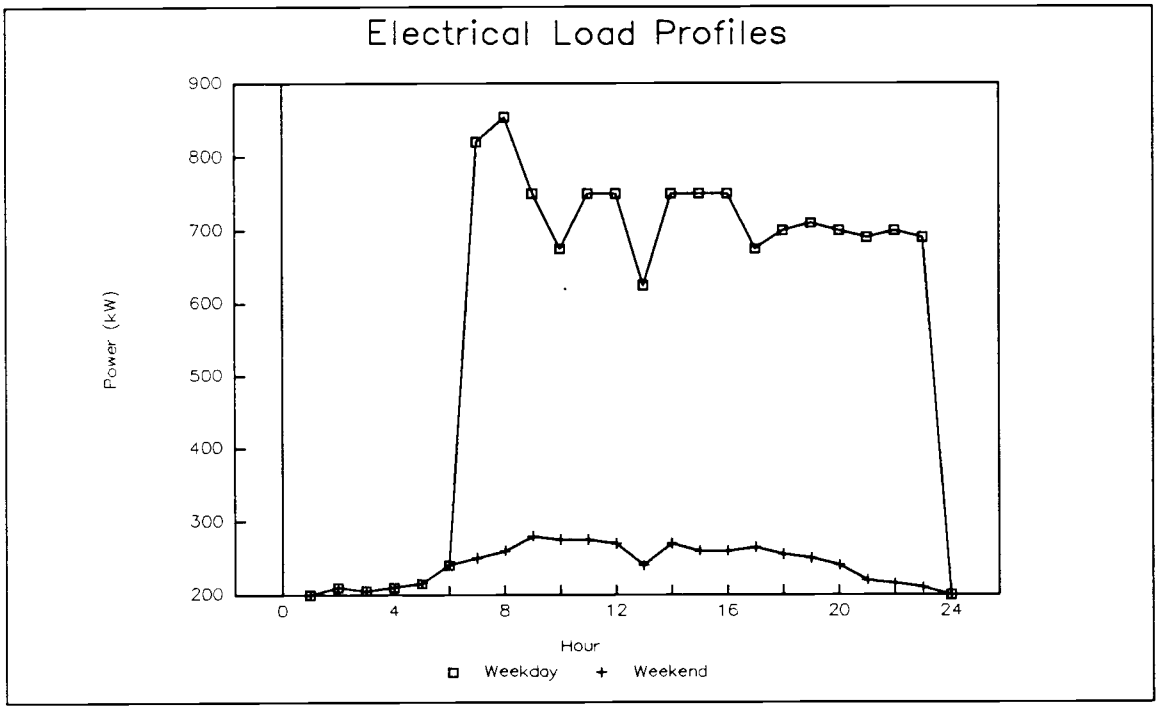

Figure 15. Case study electrical load profiles.

Steam flow measurements were not available. Mill personnel estimated that the average continuous steam flow is 4,000 lb/hr.

Utility Rate Schedule. The local utility assesses the following charges for electrical service:

Demand Charge:	$5.87/kilowatt per month
Energy Charge:	$0.021/kilowatt-hour

The existing annual electricity consumption and costs are summarized as follows:

Demand:	854 kW	$60,156
Energy:	3,036 MWh	$63,938

The value of any excess energy generation from the system (avoided cost of the local utility) was estimated at $0.022/kWh. No credit for power sales (kW) in excess of the plant demand is included. System operating and maintenance costs were estimated at $0.004/kWh [15].

Base Case Summary. Table 8 summarizes the base case annual fuel consumption and costs of the sawmill.

Table 8. Summary of existing annual fuel consumption and costs.

Boiler:	56,440 MMBtu	None
Excess Fuel:	150,696 MMBtu	$29,120
Electricity:	854 kW	$60,156
	3,036 MWh	$63,938
Total Annual Fuel Cost:		$153,214

Equipment. The mill has a new 300 psig, 20,600 lb/hr hogged fuel-fired boiler. The boiler operates at approximately one quarter load over the year. Since the steam pressure requirement of the mill is relatively high, a backpressure turbine is not practical for this application. Therefore, a full condensing autoextraction turbine has been used for the simulation.

Operation. The system was modelled using both the electrical load following and constant modes of operation. The turbines were run at their rated capacity for the constant mode simulations.

Results. The program was run using the parameters described above. The output from the program includes a generation summary, an annual fuel summary, and the simple and incremental payback analysis for turbines ranging in size from 500 kW to 2000 kW for the electrical load

following simulation and from 500 kW to 1000 kW for the constant load simulation. The maximum output from the boiler is the limiting factor in sizing a turbine for this application.

Tables 9 and 10 summarize the output from the program using an autoextraction turbine operating in the electrical load following mode (Table 9) and in the constant mode at full capacity (Table 10).

Table 9. Summary of cogeneration analysis model output operating in the electrical load following mode of operation.

Turbine Size (kW)	Elect. Cost Savings (E$)	Add'l Fuel Cost ($)	Total System Cost ($1,000)	O&M Cost ($/yr)	Net Savings ($/yr)	Simple Payback (yrs)
500	$104,876	($26,841)	$520	$13,268	$118,449	4.4
625	$130,676	($29,037)	$634	$16,505	$143,208	4.4
750	$154,139	($29,120)	$744	$19,185	$164,074	4.5
1000	$174,352	($29,120)	$950	$21,528	$181,944	5.2
1250	$186,240	($29,120)	$1,140	$23,689	$191,671	5.9
1500	$201,488	($29,120)	$1,317	$26,462	$204,146	6.5
2000	$248,668	($29,120)	$1,642	$35,040	$242,748	6.8

Table 10. Summary of cogeneration analysis model output operating in the constant mode of operation.

Turbine Size (kW)	Electrical Cost Savings ($)	Additional Fuel Cost ($)	Total System Cost ($1000)	O&M Cost ($/yr)	Net Savings ($/yr)	Simple Payback (years)
500	$127,373	($29,084)	$520	$17,520	$138,936	3.7
625	$160,268	($29,120)	$634	$21,900	$167,488	3.8
750	$193,163	($29,120)	$744	$26,280	$196,003	3.8
1000	$248,668	($29,120)	$950	$35,040	$242,748	3.9

Tables 11 and 12 show the complete output from the program for the sawmill study for the electrical load following and constant load operating modes..

GENERATION SUMMARY							ANNUAL FUEL SUMMARY				SYSTEM ECONOMICS				
	---- DEMAND ----		---- ENERGY ----		Elec.	T/G	Maximum	--- Additional ---		Boiler	Total				
Turbine Size (kW)	Billing Demand (kW)	Cost Savings (kW$)	Turbine Output (MWh)	Cost Savings ($)	Cost Savings (TE$)	Capital Cost (1000$)	Boiler Output (lb/hr)	Fuel Use (MMBtu)	Fuel Cost (F$)	Capital Cost (1000$)	System Cost (1000$)	O&M Cost (O&M$)	Net Savings ($/yr)	Simple Payback (Years)	Increm. Payback (Years)
500	354	$35,220	3,317	$69,656	$104,876	$520	11,706	83,796	($26,841)	$0	$520	$13,268	$118,449	4.4	
625	229	$44,025	4,126	$86,651	$130,676	$634	13,489	103,018	($29,037)	$0	$634	$16,505	$143,208	4.4	4.6
750	104	$52,830	4,796	$101,309	$154,139	$744	15,185	118,603	($29,120)	$0	$744	$19,185	$164,074	4.5	5.3
1000	0	$60,156	5,382	$114,196	$174,352	$950	16,730	134,283	($29,120)	$0	$950	$21,528	$181,944	5.2	11.5
1250	0	$60,156	5,922	$126,084	$186,240	$1,140	17,048	149,952	($29,120)	$0	$1,140	$23,689	$191,671	5.9	19.5
1500	0	$60,156	6,615	$141,332	$201,488	$1,317	17,433	168,784	($29,120)	$0	$1,317	$26,462	$204,146	6.5	14.2
2000	0	$60,156	8,760	$188,513	$248,668	$1,642	19,889	218,888	($29,120)	$0	$1,642	$35,040	$242,748	6.8	8.4

TURBINE SPECIFICATIONS	
Type:	AUTOEXTRACTION
Inlet Pressure:	290 PSIA
Inlet Temperature:	414 °F
Exhaust Pressure:	1 PSIA
Extraction Pressure:	165 PSIA
Operating Mode:	Electric Load Following

BOILER SPECIFICATIONS	
Fuel Type (Gas, Oil, Wood):	Wood
Maximum Steam Pressure:	315 PSIA
Operating Steam Pressure:	290 PSIA
Operating Steam Temperature:	414 °F

PLANT ELECTRIC LOADING	
Peak Demand:	854 kW
Annual Energy Use:	4,207 MWh

PLANT STEAM LOADING	
Steam Pressure:	165 PSIA

Table 11. Program output for sawmill study, electrical load following operation.

GENERATION SUMMARY							ANNUAL FUEL SUMMARY				SYSTEM ECONOMICS				
	---- DEMAND ----		---- ENERGY ----		Elec.	T/G	Maximum	--- Additional ---		Boiler	Total				
Turbine Size (kW)	Billing Demand (kW)	Cost Savings (kW$)	Turbine Output (MWh)	Cost Savings ($)	Cost Savings (TE$)	Capital Cost (1000$)	Boiler Output (lb/hr)	Fuel Use (MMBtu)	Fuel Cost (F$)	Capital Cost (1000$)	System Cost (1000$)	O&M Cost (O&M$)	Net Savings ($/yr)	Simple Payback (Years)	Increm. Payback (Years)
200	654	$14,088	1,752	$36,792	$50,880	$221	6,812	38,745	($18,135)	$0	$221	$7,008	$62,007	3.6	
500	354	$35,220	4,380	$92,153	$127,373	$520	11,706	106,153	($29,084)	$0	$520	$17,520	$138,936	3.7	3.9
625	229	$44,025	5,475	$116,243	$160,268	$634	13,489	130,721	($29,120)	$0	$634	$21,900	$167,488	3.8	4.0
750	104	$52,830	6,570	$140,333	$193,163	$744	15,185	154,089	($29,120)	$0	$744	$26,280	$196,003	3.8	3.8
1000	0	$60,156	8,760	$188,513	$248,668	$950	18,547	200,405	($29,120)	$0	$950	$35,040	$242,748	3.9	4.4

TURBINE SPECIFICATIONS

Type:	AUTOEXTRACTION	
Inlet Pressure:	290	PSIA
Inlet Temperature:	414	°F
Exhaust Pressure:	1	PSIA
Extraction Pressure:	165	PSIA
Operating Mode:	CONSTANT	

BOILER SPECIFICATIONS

Fuel Type (Gas, Oil, Wood):	Wood	
Maximum Steam Pressure:	315	PSIA
Operating Steam Pressure:	290	PSIA
Operating Steam Temperature:	414	°F

PLANT ELECTRIC LOADING

Peak Demand:	854	kW
Annual Energy Use:	4,207	MWh

PLANT STEAM LOADING

Steam Pressure:	165	PSIA

Table 12. Program output for sawmill study, constant mode operation.

The results of the analysis allow for the comparison of different sized systems as well as for different modes of operation. The results show simple paybacks under four years for several of the options. The turbines operating in the constant mode consistently yield a better payback than the turbines operating in the electrical load following mode. As a result of the expense of hauling away excess fuel, a distinct minimum payback does not occur. Based on obtaining a payback under four years while maximizing the base load power produced by the cogeneration system, the optimum arrangement is a 625 kW turbine operating in a constant mode at its rated capacity.

Since the payback is under four years, the mill is a good candidate for cogeneration. The next step would be to perform a detailed heat and mass balance on the facility for each of the typical days. A more detailed economic analysis would also be appropriate to help obtain a more accurate estimation of cogeneration feasibility.

10. SUMMARY

Detailed cogeneration analysis is an involved and costly process. The modelling program and methods described in this paper allow reasonable estimate of cogeneration feasibility.

The model was designed to serve as a preliminary screening tool for determining cogeneration feasibility. It offers the following system design options:

- Wood-, gas-, or oil-fired boilers.
- Monthly variation in boiler efficiency for wood-fired systems.
- Hourly steam and electrical load profiles for up to 6 typical operating days per year
- Condensing, backpressure, or autoextraction turbines.
- Full and part load performance modelling.
- Three operating modes:
  Thermal load following
  Electrical load following
  Constant load operation.

The model, a menu-driven spreadsheet application program, calculates fuel requirements, electrical energy and power production, costs, equipment size and cost, total cost savings, and simple payback for one or more cogeneration alternatives.

BIBLIOGRAPHY

1. Singer, Joseph G. (ed.), Combustion, Windsor, Connecticut: Combustion Engineering, 1981, Chapter 2, pp. 12.

2. The American Society of Mechanical Engineers, Steam Generating Units, Power Test Codes, New York, 1964.

3. Keenan, Joseph H. et al., Steam Tables, New York: John Wiley & Sons, Inc., 1969.

4. Kuck, I. Z. , "Thermodynamic Properties of Water for Computer Simulation of Power Plants," Department of Nuclear Engineering, University of Arizona, 1981.

5. Gonzalez-Pozo, Virgilio, "Formulas Estimate Properties For Dry, Saturated Steam," Chemical Engineering, May 12, 1986, pp.123.

6. Harold Pender (ed.), Electrical Engineers' Handbook (New York: John Wiley & Sons, Inc., 1936), Chapter 5, p. 50.

7. Oregon State University Extension Energy Program, Commercial Audit Manual (Corvallis, Oregon: 1986), p. 2.7.

8. Newman, Louis E. et al., Modern Turbines, New York: John Wiley & Sons, Inc., 1944), pp. 81.

9. Newman, Louis E., "Modern Extraction Turbines," Power Plant Engineering, Vol. 45, January, February, March, April, 1945.

10. Newman, Louis E., "Modern Extraction Turbines," Power Plant Engineering, Vol. 45, February, 1945, pp. 78.

11. Op Cit, pp. 81.

12. Louis E. Newman, "Modern Extraction Turbines," Power Plant Engineering 45 (March, 1945) pp. 85.

13. The Technology Applications Laboratory of the Georgia Institute of Technology, The Industrial Wood Energy Handbook, Van Rostrand Reinhold Company, New York, 1984, pp. 160.

14. Pacific Northwest Laboratory, Cogeneration Handbook for the Pulp and Paper Industry, Washington D.C., U.S. Department of Energy, 1984.

15. Southeastern Regional Biomass Energy Program (SERBEP), Design Manual For Small Steam Turbines, Tennessee Valley Authority, Muscle Shoals, Alabama, 1990, pp. 85.

16. Wilson, Pamela L., James W. Funck, and Robert B. Avery, Fuelwood Characteristics of Northwest Conifers and Hardwoods, Forest Research Laboratory, Oregon State University, Research Bulletin 60, 1987.

APPENDICES

APPENDIX A

Steam Property Curvefit Equations

Dry, saturated steam. The following equation is used in the model to estimate properties of dry, saturated steam as a function of absolute pressure (p):

$$Y = Ap + B/p + Cp^{\frac{1}{2}} + D\ln(p) + Ep^2 + Fp^3 + G \qquad (30)$$

Table 13. Constants used for calculating properties of dry, saturated steam.

	A	B	C	D	E	F	G
T:	-0.17724	3.83986	11.48345	31.1311	8.762969E-05	-2.78794E-08	86.594
h_f:	-0.15115567	3.671404	11.622558	30.832667	8.74117E-05	-2.62306E-08	54.55
h_{fg}:	0.008676153	-1.3049844	-8.2137368	-16.37649	-4.3043E-05	9.763E-09	1045.81
h_g:	-0.14129	2.258225	3.4014802	14.438078	4.222624E-05	-1.569916E-08	1100.5
s_f:	-1.67772E-04	4.272688E-03	0.01048048	0.05801509	9.101291E-08	-2.7592E-11	0.11801
s_{fg}:	3.454439E-05	-2.75287E-03	-7.33044E-03	-0.14263733	-3.49366E-08	7.433711E-12	1.85565
s_g:	-1.476933E-04	1.2617946E-03	3.44201E-03	-0.08494128	6.89138E-08	-2.4941E-11	1.97364

where

T	=	Saturation temperature:	°F
h_f	=	Liquid enthalpy:	Btu/lb
h_{fg}	=	Vaporization enthalpy:	Btu/lb
h_g	=	Vapor enthalpy:	Btu/lb
s_f	=	Liquid entropy:	Btu/lb-°R
s_{fg}	=	Vaporization entropy:	Btu/lb-°R
s_g	=	Vapor entropy:	Btu/lb-°R

Superheated steam. The following equations are used by the model to estimate the enthalpy (h_g) and entropy (s_g) of superheated steam as functions of pressure (press) and temperature (temp).

Enthalpy

$$h_g = [\text{Part1} + \text{R1}(\text{Part2} + \text{Part3}) + \text{h1}]\ h_{conv} \qquad (31)$$

where

$$\text{Part1} = C1(T-T1) + (Ts - T1s)C2/2 + (Tc-T1c)C3/3$$
$$\text{Part2} = Z1(P-P1) + Z3[(Ps/T)-(P1s/T1)] + (Ps-P1s)Z4/2$$
$$\text{Part3} = (Pc-P1c)Z6/3 + [(Pc/T)-(P1c/T1)]Z8(2/3) + [(Pc/Ts)-(P1c/T1s)]Z9$$

Entropy

$$s_g = [\text{Part1} - R1(\text{Part2}+\text{Part3}+\text{Part4}) + s1]s_{conv} \quad (32)$$

where

$$\text{Part1} = C1\ Ln(T/T1) + (T-T1)C2 + (Ts-T1s)C3/2$$
$$\text{Part2} = Z0\ Ln(P/P1) + (P-P1)Z2 + [(Ps/Ts)-(P1s/T1s)]Z3/2$$
$$\text{Part3} = (Ps-P1s)Z5/2 + (Pc-P1c)Z7/3$$
$$\text{Part4} = [(Pc/Ts)-(P1c/T1s)](-Z8/3) - [(Pc/Tc)-(P1c/T1c)]Z9(2/3)$$

The following constants and conversions are used in both of the above equations (Eqs. 31 and 32):

C1	=	1.7524
C2	=	2.4936E-04
C3	=	3.0978E-07
Z0	=	1.0001
Z1	=	-5.3391
Z2	=	6.1322E-03
Z3	=	-74.961
Z4	=	0.25547
Z5	=	-2.0580E-04
Z6	=	-8.9010E-03
Z7	=	4.3505E-06
Z8	=	6.1847
Z9	=	-1471.5
T1	=	273.16
P1	=	0.006113
h1	=	2501.3
s1	=	9.1571
R1	=	0.4619
T_{abzro}	=	-459.67
T_{conv}	=	1.8
P_{conv}	=	14.696
S_{conv}	=	0.23901

$$P = press/P_{conv}$$
$$T = (temp - T_{abzro})/T_{conv}$$

$$Ps = P^2$$
$$Pc = P^3$$
$$Ts = T^2$$
$$Tc = T^3$$
$$P1s = P1^2$$
$$P1c = P1^3$$
$$T1s = T1^2$$
$$T1c = T1^3$$

APPENDIX B

Cost Estimation Curvefits

Wood-fired boilers. The following equation is used in the model to estimate the installed cost of wood-fired boilers as a function of the boiler capacity in pounds per hour:

$$\text{Cost} = A + Bx + Cx^2 \tag{33}$$

where

A = 0.7888E+06
B = 0.1607E+02
C = 0.5809E-04

Gas- and oil-fired boilers. The following equation is used in the model to estimate the installed cost of gas- and oil-fired boilers as a function of the boiler capacity in pounds per hour:

$$\text{Cost} = A + Bx + C/x \tag{34}$$

where

A = 0.8317E+05
B = 0.4072E+01
C = 0.7594E+09

Condensing turbine-generating sets. The following equation is used in the model to estimate the installed cost of condensing turbines in dollars per kilowatt as a function of the turbine capacity in kilowatts:

$$\text{Cost/kW} = 1/[A\ (B + x)^2 + C] \tag{35}$$

where

$$A = -0.1083E-10$$
$$B = -0.9154E+04$$
$$C = 0.1773E-02$$

Non-condensing turbine-generating sets. The following equation is used by the model to estimate the installed cost of non-condensing turbines in dollars per kilowatt as a function of the turbine capacity in kilowatts:

$$\text{Cost/kW} = 1/[A\ (B + x)^2 + C] \tag{36}$$

where

$$A = -0.1242E-10$$
$$B = -0.8640E+04$$
$$C = 0.1828E-02$$

APPENDIX C

Wood Fuel Properties

The following wood species are included in the database used in the model. Table 13 lists the elemental fuel composition for each species in the list.

1. Douglas Fir Wood
2. Douglas Fir Bark
3. Western Hemlock Wood
4. Western Hemlock Bark
5. Lodgepole Pine
6. Lodgepole Pine Bark
7. Ponderosa Pine
8. Ponderosa Pine Bark
9. Redwood
10. Redwood Bark
11. Western Red Cedar
12. Western Red Cedar Bark
13. Port Orford Cedar
14. Port Orford Cedar Bark
15. Red Alder
16. Red Alder Bark
17. Oregon Ash
18. Oregon Ash Bark
19. Bigleaf Maple
20. Bigleaf Maple Bark
21. Oregon White Oak
22. Oregon White Oak Bark
23. Generic Softwood
24. Generic Softwood Bark
25. Generic Hardwood
26. Generic Hardwood Bark

Table 14. Elemental fuel composition for various wood species [16].

	C (%)	H (%)	O (%)	N (%)	Ash (%)	HHV (Btu/lb)	GMC (%)
1.	52.3	6.3	40.5	0.1	0.8	8600	33
2.	56.2	6.2	37.3	0.1	0.2	10000	33
3.	50.4	5.8	41.4	0.1	2.2	8500	44
4.	53.0	6.2	39.3	0.1	1.5	9400	41
5.	52.9	6.3	39.7	0.1	1.0	8600	36
6.	55.0	5.8	38.6	0.1	0.8	10000	50
7.	52.9	6.3	39.7	0.1	1.0	9100	52
8.	53.1	5.9	37.9	0.2	2.9	9500	23
9.	53.5	5.9	40.3	0.1	0.2	9000	55
10.	51.9	5.1	42.4	0.1	0.5	8350	12
11.	52.9	6.3	39.7	0.1	1.0	9000	33
12.	53.1	5.9	37.9	0.2	2.9	8850	32
13.	52.9	6.3	39.7	0.1	1.0	9000	30
14.	53.1	5.9	37.9	0.2	2.9	9000	25
15.	50.8	6.4	41.8	0.4	0.6	8000	50
16.	51.2	6.0	37.9	0.4	4.5	8600	45
17.	50.8	6.4	41.8	0.4	0.6	8200	32
18.	51.2	6.0	37.9	0.4	4.5	8200	32
19.	50.8	6.4	41.8	0.4	0.6	8250	38
20.	51.2	6.0	37.9	0.4	4.5	8300	38
21.	50.8	6.4	41.8	0.4	0.6	8110	41
22.	51.2	6.0	37.9	0.4	4.5	8110	41
23.	52.9	6.3	39.7	0.1	1.0	9000	50
24.	53.1	5.9	37.9	0.2	2.9	9100	40
25.	50.8	6.4	41.8	0.4	0.6	9000	50
26.	51.2	6.0	37.9	0.4	4.5	9100	40

where

C = Carbon
H = Hydrogen
O = Oxygen
N = Nitrogen
HHV = Higher heating value
GMC = Green moisture content (wet-basis)

APPENDIX D

Complete tables of Theoretical Steam Rate Factors

Table 15. Theoretical steam rate factors for straight condensing turbines.

Straight Condensing Turbines				
	Full Load		Half Load	
kW	200 psi	600 psi	200 psi	600 psi
500	1.690	1.798	1.880	1.990
750	1.605	1.700	1.780	1.870
1000	1.560	1.640	1.720	1.790
1500	1.505	1.560	1.640	1.700
2000	1.460	1.520	1.590	1.640
3000	1.420	1.460	1.540	1.570
4000	1.395	1.430	1.510	1.540
5000	1.380	1.410	1.490	1.515
6000	1.370	1.392	1.482	1.500
7500	1.360	1.380	1.470	1.490

Table 16. Theoretical steam rate factors for straight non-condensing turbines.

Straight Non-condensing Turbines				
	Full Load		Half Load	
kW	200 psi	600 psi	200 psi	600 psi
500	1.700	1.890	2.060	2.230
750	1.600	1.760	1.930	2.055
1000	1.540	1.680	1.845	1.950
1500	1.470	1.580	1.750	1.821
2000	1.425	1.520	1.690	1.750
3000	1.380	1.440	1.630	1.670
4000	1.355	1.412	1.593	1.622
5000	1.345	1.390	1.572	1.590
6000	1.335	1.375	1.560	1.570
7500	1.330	1.360	1.550	1.560

Table 17: Full load Non-extraction efficiencies for condensing single-automatic steam turbines.

Rated kW	MAIN PRESSURE (psig) 150	200	250	300	400	600	850	HLF
	EFFICIENCY							
500	0.600	0.595	0.585	0.580	0.565	0.545		0.590
625	0.615	0.610	0.605	0.600	0.580	0.560		0.590
750	0.630	0.625	0.620	0.610	0.595	0.575		0.590
1000	0.650	0.645	0.640	0.630	0.620	0.600		0.585
1250	0.665	0.660	0.650	0.645	0.635	0.615		0.585
1500	0.675	0.670	0.665	0.660	0.645	0.630		0.585
2000	0.690	0.685	0.680	0.675	0.665	0.645		0.580
2500	0.700	0.695	0.690	0.685	0.675	0.660		0.580
3000	0.710	0.705	0.700	0.695	0.685	0.670		0.580
3500	0.715	0.710	0.705	0.700	0.690	0.680		0.575
4000	0.720	0.715	0.710	0.705	0.700	0.685		0.575
5000	0.725	0.720	0.715	0.710	0.705	0.695	0.685	0.575
6000	0.735	0.730	0.725	0.720	0.715	0.705	0.695	0.570
7500	0.740	0.735	0.730	0.725	0.720	0.715	0.705	0.570

Table 18: Full load Non-extraction efficiencies for non-condensing single-automatic steam turbines.

Rated kW	MAIN PRESSURE (psig) 150	200	250	300	400	600	850	HLF
	EFFICIENCY							
500	0.590	0.580	0.570	0.560	0.540	0.500		0.640
625	0.615	0.605	0.595	0.580	0.560	0.520		0.640
750	0.630	0.620	0.610	0.595	0.575	0.535		0.640
1000	0.655	0.645	0.635	0.620	0.605	0.565		0.635
1250	0.670	0.660	0.650	0.640	0.625	0.585		0.635
1500	0.680	0.670	0.665	0.655	0.640	0.600		0.635
2000	0.700	0.690	0.685	0.675	0.660	0.630		0.630
2500	0.710	0.705	0.695	0.690	0.675	0.645	0.620	0.630
3000	0.720	0.715	0.705	0.700	0.690	0.660	0.635	0.630
3500	0.725	0.720	0.715	0.710	0.695	0.670	0.650	0.625
4000	0.730	0.725	0.720	0.715	0.705	0.680	0.660	0.625
5000	0.735	0.735	0.730	0.725	0.715	0.695	0.675	0.625
6000	0.740	0.735	0.735	0.730	0.725	0.705	0.685	0.620
7500	0.745	0.740	0.740	0.735	0.730	0.715	0.700	0.620

APPENDIX E

Nonextraction Thermal Load Following Equations

The model of a thermal load following non-extraction turbine is very similar to the electric load following model. The primary difference is that electrical power is the dependent variable and steam flow is the independent variable. Again, the straight line equation is used and the slope and y-intercept terms are determined.

$$P = m M + b \tag{37}$$

The slope of the line (m) is defined as the change in turbine output divided by the corresponding change in steam flow.

$$m = (P_{FULL}/2) \ / \ (M_{FULL} - M_{HALF}) \tag{38}$$

Using the theoretical steam rate and TSR factors for full and half load, the slope of the curve can be rewritten as

$$\begin{aligned} m &= (P_{FULL}/2) \ / \ \{(TSR \ P_{FULL}/\eta_{FL}) - [TSR(P_{FULL}/2)/\eta_{HL}]\} \\ &= (\eta_{FL}\eta_{HL})/[TSR \ (2\eta_{HL} - \eta_{FL})] \end{aligned} \tag{39}$$

The y-intercept (b) determined by rearranging the equation and substituting for M and kW at a known point (again, at full load operation).

$$\begin{aligned} b &= P - m M \\ &= P_{FULL} - m M_{FULL} \\ &= P_{FULL} - [m \ (TSR \ P_{FULL}/\eta_{FL})] \end{aligned} \tag{40}$$

Further substitution for the slope, m, yields the following expression for the y-intercept.

$$b = P_{FULL} \left[(\eta_{HL} - \eta_{FL}) / (2\ \eta_{HL} - \eta_{FL}) \right] \tag{41}$$

The complete equation for non-extraction, thermal load following turbine performance can now be written as

$$\begin{aligned} kW &= m\ M + b \\ &= \left[(\eta_{FL}\ \eta_{HL}) / TSR\ (2\ \eta_{HL} - \eta_{FL}) \right] M + \\ &\quad P_{FULL} \left[(\eta_{HL} - \eta_{FL}) / (2\ \eta_{HL} - \eta_{FL}) \right] \end{aligned} \tag{42}$$

For a given thermal load profile, the required steam flow can be estimated using the above equation.