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ENERGY CONSERVATION IN CITRUS PROCESSING

Technical Progress Report No. 2 for the Period April 1, 1980—
February 28, 1981

DOE/CS/40263--T2

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March 15, 1981

Work Performed Under Contract No. AC03-79CS40263

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U. S. DEPARTMENT OF ENERGY

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Prepared by

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Prepared for
U.S. Department of Energy
Office of Industrial Programs
Agriculture and Food Processes

PREFACE

The project, "Energy Conservation in Citrus Fruit Processing," is designed to define and evaluate alternative energy conserving systems for citrus processing plants with the objective of identifying systems that reduce energy usage and are economically viable. The operations of citrus processing including conveying and extraction, concentration, peel drying, refrigeration, and canning. An examination of the manner in which these operations are performed and how their energy requirements are met provides an opportunity for the application of innovative system concepts that substantially reduce the total energy demand of the citrus processing plant. The alternative energy conserving systems are synthesized from available components and subsystems that are arranged to make use of energy cascading and thermodynamic regeneration to reduce the overall energy usage of the plant.

The work is being carried out in two phases, each phase requiring one year to complete. The first phase is a systems definition study which terminates with a general description of alternative system concepts and a preliminary economic and energy usage evaluation. The second phase is a more detailed systems evaluation study in which the alternative system concepts provide the basis for performance analysis and optimization via computer simulation. The energy usage, costs, and system operating characteristics thus established serve as the basis for the evaluation of the alternative systems and their subsequent ranking.

The preliminary results of phase one of the project are presented in this report. The technical parameters, economic characteristics, and energy usage of alternative energy conserving systems for citrus processing plants are presented in this report. These results are subject to revision as the more detailed evaluation of phase two of the project is carried out.

TABLE OF CONTENTS

PREFACE	i
1. INTRODUCTION	1
2. OBJECTIVES	3
3. APPROACH	4
3.1 Types of Alternative Systems	4
3.2 Components of Operational Units & Subsystems	5
4. WORK PLAN	11
4.1 Overview of the Project	11
4.2 Task Description - Phase I, System Definition	11
4.3 Task Description - Phase II, System Evaluation	11
5. PROGRESS OF TECHNICAL WORK	15
5.1 Project Status	15
5.2 Work Completed - Task 1A - Requirements	15
5.2.1 Energy Price Projection	17
5.2.2 Atmospheric Emission Requirements	17
5.3 Work Completed - Characterization and Thermodynamic Analysis	17
5.3.1 Existing Peel Drying System - Process Description	19
5.3.2 Existing Peel Drying System - Material and Energy Balance	19
5.4 Task 1C - IE, Thermodynamics, Synthesis and Preliminary Evaluation of Systems	28
5.4.1 Central Brayton Co-generation	28
5.4.2 Central Rankine Co-generation	31
5.4.3 Central Reciprocating Co-generation	40
5.4.4 Peel Dryer with Brayton Engine	40
5.4.5 Peel Dryer with Reciprocating Engine	56
5.4.6 Thermal Evaporator with Steam Turbine	56
5.4.7 Dual-Mode Evaporator	70
5.4.7.1 System Description	70
5.4.7.2 Process Constraints	71
5.4.7.3 Specifications and Performance Parameters	71
5.4.8 Heat Pump for Pasteurization	78

LIST OF FIGURES

1. Existing Operation Units and Energy Flow Paths of the Sunkist Plant in Ontario	6
2. Central Energy Supply System	7
3. Modular Energy Supply System	8
4. Gulf Dryer - Material Inputs & Outputs	21
5. Summary of Material and Heat Balances of Dryer/WHE System	25
6. Central Brayton Co-generation	30
7. Central Rankine Co-generation	36
8. Central Reciprocating Engine - Co-generation	41
9. Peel Dryer with Brayton Engine - Parallel Mode	46
10. Gas Turbine Size Vs. Dryer Exhaust Humidity	50
11. Peel Dryer with Brayton Engine - Series Mode	54
12. Peel Dryer with Reciprocating Engine - Parallel Mode	60
13. Thermal Evaporator with Steam Turbine	64
14. Dual-Mode Juice Evaporator	72

LIST OF TABLES

1. List of Tasks Comprising the Project	12
2. Plant Parameters and Energy Costs (Typical Values, Sunkist Plant, Ontario, California)	16
3. Peel Dryer Mass Balance and Heat Contents of Reactants	24
4. Specifications and Performance Parameters - Existing Peel Dryer System	27
5. System Specifications and Performance Parameters for Central Brayton Co-generation	29
6. Capital Cost Breakdown for Central Brayton Co-generation System	32
7. Economic Characteristics of the Central Brayton System	33
8. Energy Savings for Central Brayton Co-generation	34
9. System Specifications and Performance for Central Rankine Co-generation	35
10. Capital Cost Breakdown for Central Rankine Co-generation System	37
11. Economic Characteristics of Central Rankine Co-generation System	38
12. Energy Savings for Central Rankine Co-generation	39
13. System Specification and Performance Parameters for Central Reciprocating Engine Co-generation	42
14. Capital Cost Breakdown for Central Reciprocating Engine Co-generation System	43
15. Economic Characteristics of Central Reciprocating Engine Co-generation	44
16. Energy Savings for Central Reciprocating Engine Co-generation	45
17. System Specifications and Performance Parameters for Brayton Engine Only	47
18. Energy Savings for Brayton Engine Only Adapted to Citrus Peel Dryer	49

LIST OF TABLES (CONT'D)

19.	System Specifications and Performance Parameters for Brayton Engine in Parallel with Direct Firing	51
20.	Energy Savings for Brayton Engine in Parallel with Burner for Citrus Peel Dryer	52
21.	System Specifications and Performance Parameters of Brayton Engine with Afterburner	53
22.	Capital Cost Breakdown for Dryer Co-generation Via Brayton Engine with Afterburner	55
23.	Economic Characteristics of Dryer Co-generation Via Brayton Engine with Afterburner	57
24.	Energy Savings of Brayton Engine with Afterburner Adapted to Citrus Peel Dryer	58
25.	System Specifications and Performance Parameters for Reciprocating Engine Parallel with Direct Firing	59
26.	Capital Cost Breakdown Dryer Co-generation Via Reciprocating Engine - Parallel Mode	61
27.	Economic Characteristic of Dryer Co-generation Via Reciprocating Engine - Parallel Mode	62
28.	Energy Savings for Reciprocating Engine Adapted to Citrus Peel Dryer	63
29.	System Specifications and Performance Parameters for Both Existing Supply to Evaporation and for Modular Thermal Evaporator/Steam Turbine	66
30.	Capital Cost Breakdown for Modular Thermal Evaporator/ Steam Turbine Combination	67
31.	Economic Characteristics of Modular Thermal Evaporator/ Steam Turbine Combination	68
32.	Energy Savings for Modular Thermal Evaporator/Steam Turbine	69
33.	System Specification and Performance Parameters for Dual-Mode Orange Juice Evaporator	73
34.	Capital Cost Breakdown for Modular Dual-Mode Evaporator	76
35.	Economic Characteristics of Dual-Mode Evaporator with Brayton Engine	77
36.	Specifications and Performance Parameters for Dual-Mode Orange Juice Evaporator	79
37.	Energy Savings and Economic Characteristics of Juice Pasteurization Via Heat Pump	80

1. INTRODUCTION

The Sunkist Citrus Plant in Ontario, California processes about six million pounds of citrus fruit per day to make a variety of products which include frozen concentrated juice; chilled, pasteurized, natural strength juice; molasses from peel; dried meal from peel; pectin; citrus oil; and bioflavonoids. The entire plant typically requires about 50×10^6 kilowatt-hours of electrical power per year and about 10×10^{11} BTU per year of thermal energy from the combustion of natural gas or fuel oil. Presently, all electrical power is purchased from Southern California Edison and all the thermal energy is produced on-site in the furnace of steam boilers and of dryers.

The preferred fuel for steam generation and for drying has been natural gas. However, the price of natural gas is being deregulated, resulting in a very rapid escalation of fuel costs. Since citrus fruit processing is presently an energy intensive operation, the substantially higher cost of fuel results in much higher processing costs and ultimately in higher market prices for citrus products. Some products may be discontinued if the cost of fuel makes them too expensive.

The Sunkist citrus fruit processing plant was originally designed when natural gas was considered to be plentiful and was the most economical energy source. Consequently, energy utilization efficiency was not a predominant factor in its original design nor in the original design of other types of food processing plants. The energy intensive operations at the Sunkist plant include concentration, drying and refrigeration.

An examination of the manner in which these operations are executed and the manner in which their energy requirements are satisfied provides an

1. INTRODUCTION (CONT'D)

opportunity for the application of innovative system concepts which will significantly reduce the total energy demand of the plant. These concepts are based on available technology. The evaporation, drying and refrigeration operations in citrus processing are typical of other food processing industries, and energy conserving systems developed for citrus processing could be adapted to other types of food processing.

2. OBJECTIVE

The principle objective of the work is to identify an economically viable alternative to the existing method of meeting the energy requirement of citrus fruit processing that will substantially reduce the overall energy usage of citrus processing plants. This objective will be accomplished in a two-phase project consisting of a systems definition study in phase one and a systems evaluation study in phase two. The project is designed to identify the overall configuration, equipment specifications, the analytically predicted performance and operating characteristics, and the projected total cost of several alternative systems. This information provides the basis of the selection of the system that is most desirable both in terms of economic viability for the citrus processing industry and of reduced energy usage. The final results of the project consist of the selected alternative system, a strategy for its implementation, and the technology transfer activity.

The project is structured so that currently available components are used in different arrangements to constitute the different types of alternative systems. The components which will make up the alternative systems include: evaporators, dryers, refrigeration units, heat pumps, heat engines, heat exchangers, thermal storage units, and ancillary components. These components will be used to form the five operational units of the citrus processing plant. These operational units are: evaporation, drying, refrigeration, pasteurizing and canning, and the plant electrical load that consists of operations such as conveying and juice extraction. The five operational units are then interrelated to varying degrees with respect to energy exchange to form different types of alternative systems. These alternative systems differ from existing systems in that they make optimum use of thermodynamic regeneration, cascaded energy utilization, and thermal storage to minimize the energy usage of the citrus processing operations without changing the product process stream variables.

3. APPROACH

3.1 Types of Alternative Systems

The study of alternative systems to reduce energy usage in citrus processing focused on the manner in which the energy requirements of the various basic operations in citrus processing are satisfied. These fundamental energy requirements are taken as they presently exist, except where there is an advantage in changing a particular operation where known flexibility of process requirements may be advantageously exploited. The different types of alternative systems for meeting these operational energy requirements will consist of different arrangements of some or all of the following major components: evaporators, dryers, heat engines, heat pumps, refrigeration machines, heat exchangers, compressors, and pumps, thermal storage units, and microprocessors. The alternative systems feature cascaded energy utilization wherein the heat source, work absorbing or producing and heat rejection characteristics of the various components and subsystems are properly matched to satisfy the specific energy requirements of the basic operations of citrus processing.

The basic operations of citrus processing and their supporting operations which will be considered in this project are:

- A. evaporation of citrus juice to produce a concentrate, and evaporations of peel liquor to produce molasses;
- B. refrigeration for chilling, freezing, and storage of citrus juice products;
- C. pasteurizing and canning of citrus juice;
- D. drying of peel and pulp to produce dried citrus meal;
- E. the various electrically driven operations, such as conveying, grinding, pressing, juice extraction, centrifuging, fluid pumping, compressing, pelletizing, and ventilation.

3.1 Types of Alternative Systems (Cont'd)

These basic operations are performed by individual assemblages of components that are referred to as "Operational Units." The operation units that are to be considered in this study are shown in Figure as they exist within the present energy utilization system of the Sunkist plant in Ontario, California. Each of the alternative energy utilization systems to be considered in this study is comprised of the five operational units shown in Figure 1, which are Evaporation, Pasteurization and Canning, Refrigeration, Drying and the Plant Electrical Load. The energy supply subsystem, which is a separate entity, will be partially or wholly incorporated as part of the operational units of certain types of alternative systems. The alternative systems to be considered in this project are classified in four general categories, denoted as Central, Modular, Integrated, and Combined. This Technical Progress Report deals only with the Central and Modular systems.

The Central systems feature a separate central energy supply subsystem from which the energy requirements of the operational units are supplied. The conceptual layout and energy flow paths for one of the several central systems to be considered are illustrated in Figure 2.

A Modular system consists of the five operational units wherein an individual energy supply subsystem is contained within each separate operational unit. The aforementioned balance between thermally-driven and work-driven operations within the modular operational units will be adjusted to optimally exploit the heat source, work producing, and heat rejection characteristics of the energy supply subsystem. An illustration of a Modular system is shown in Figure 3.

3.2 Components of Operational Units and Subsystems

The components from which the operational units and the energy supply subsystems will be synthesized are listed and briefly described below. These components are the basic elements that will be used in different combinations and arrangements to form all the alternative systems:

FIGURE 1: Existing operational units and energy flow paths of the Sunkist plant in Ontario, California.

FIGURE 2: An illustration of the concept of Central energy utilization systems in which the energy supply subsystem could incorporate either a Rankine, a Brayton, or a Diesel heat engine.

FIGURE 3: An illustration of the concept of Modular energy utilization systems in which the energy supply subsystem could incorporate either a Rankine, a Brayton, or a Diesel heat engine. Each module functions largely as an independent unit.

3.2 Components of Operational Units and Subsystems (Cont'd)

- A. **Evaporators:** Evaporation operations may be driven with energy in the form of either heat or work, and a single evaporator unit may utilize both thermally-driven stages and mechanically-driven stages. Multiple-effect, thermally-driven evaporators and mechanically-driven evaporators of both the vapor-recompression type and the heat-pump type will be analyzed for use in the evaporator subsystems.
- B. **Refrigeration Machines:** Refrigeration operations may also be driven with either heat or work. Thermally-driven absorption refrigeration and two types of mechanical refrigeration, vapor-compression and regenerative gas-expansion will be considered for the alternative systems.
- C. **Heat Pumps:** Heat pumps are work-driven devices which function to elevate heat from a low temperature reservoir to a higher temperature reservoir. For example, consider that a fluid effluent stream is available at a temperature of 200°F. (93°C). A heat pump could be employed to recover a portion of the heat of the effluent stream to supply some operation that requires heat at a temperature of 300°F. (149°C).
- D. **Thermal Storage Units:** Different types of thermal storage units will be parameterically examined for use in the alternative systems to mitigate any energy usage increase resulting from transient processing load and duty cycles.
- E. **Heat Exchangers:** Heat exchangers are essential subsystem components whose performance will be parametrically characterized. Waste heat boilers and fluid effluent heat recovery units are examples of heat exchangers to be utilized in the alternative systems.
- F. **Dryers:** The drying operations at the Sunkist plants require large amounts of heat at fairly high temperatures. The effect of this study will be directed toward the manner in which the heat requirements of the drying operation are met and, except for minimizing

3.2 Components of Operational Units and Subsystems (Cont'd)

direct thermal losses, will not explore extensive changes in the drying operation itself. Among the options to be examined is the use of gas turbine exhaust to provide heat for the drying operation.

G.. Heat Engines: The three fundamental types of heat engines upon which the energy supply subsystems can be based are the Brayton (gas turbine), Diesel, and Rankine (steam turbine). These engines have markedly different characteristics and are one basis for variation within a category of alternative systems.

In addition to the major subsystem components described above, the alternative systems will incorporate such components as compressors, pumps, steam ejectors, and electric motors.

4. WORK PLAN

4.1 Overview of the Project

The proposed work is being carried out in two phases, each phase requiring one year to complete. The first phase is a systems definition study, which terminates with a general description of alternative system concepts and preliminary economic and energy usage evaluation. The second phase is a systems evaluation study in which the alternative system concepts provide the basis for a more detailed performance analysis and optimization via computer simulation. The energy usage, operating cost, initial cost, and system operating characteristics thus established serve as the basis for the evaluation of the alternative systems and the subsequent selection of the most desirable system.

4.2 Task Descriptions - Phase I - Systems Definition

The individual tasks that make up the two phases of this project are listed in Table 1. The work that has actually been done since April 1980 is described in a subsequent section of this report.

4.3 Task Description - Phase II - Systems Evaluation

The more promising systems from Phase I will be evaluated as to their economic viability. A detailed fuel and electrical use profile will be used in determining the systems true energy savings per year. The time of day rate schedules will be used to determine the dollar savings. Projected energy costs will be used in the final economic calculations. As required, a computer program will be made to expedite certain calculations.

Table 1. List of Tasks Comprising the Project with JPL Responsibility

<u>TASK NUMBER AND TITLE</u>	<u>DESCRIPTION OF ACTIVITY</u>	
<u>PHASE I</u>	<u>SYSTEMS DEFINITION</u>	<u>JPL RESPONSIBILITY</u>
I.A Identification of Constraints, Requirements and External Factors	<ul style="list-style-type: none"> • Environmental and product quality factors • Government regulations • Establishment of total cost function • Future energy cost and availability projections • Uncertainty assessment of projections 	Assist Sunkist in establishing the total cost function, the energy cost and availability projections, and the uncertainty assessment
I.B Characterization of Basic Operations	<ul style="list-style-type: none"> • Analytical and experimental determination of process state variables and throughput in existing system 	Assist Sunkist in identifying the measurements to be made and in the formulation of the data reduction algorithm
I.C Thermodynamic Analysis of Operations and Components	<ul style="list-style-type: none"> • Formulation and validation of analytical models of components and operations 	JPL will perform the analysis based on the characterization provided by Sunkist
I.D Synthesis of Alternative Systems	<ul style="list-style-type: none"> • Formulation of conceptual layout of alternative systems and preliminary prediction of performance and operating characteristics 	JPL will perform the systems synthesis based on the criteria provided by Sunkist
I.E Preliminary Evaluation	<ul style="list-style-type: none"> • Selection of more promising alternative systems on basis of preliminary economic and energy usage characteristics 	Assist Sunkist in preliminary application of total cost function and in the establishment of energy usage and operating characteristics of alternative systems

Table 1. (continued)

<u>TASK NUMBER AND TITLE</u>		<u>DESCRIPTION OF ACTIVITY</u>	
<u>PHASE II</u>		<u>SYSTEMS EVALUATION</u>	<u>JPL RESPONSIBILITY</u>
II.A	Simulation of Components and Operations	<ul style="list-style-type: none"> Formulation of computer codes based on analysis of Task I.C 	JPL will formulate the computer codes to be validated according to criteria provided by Sunkist
II.B	System Model Integration	<ul style="list-style-type: none"> Assemblage of component simulation codes to form complete system models for alternative systems identified in Task I.E 	JPL will assemble the component simulation models to form system models representing the alternative systems. Sunkist will assist in system model formulation and provide criteria for checkout
II.C	System Optimization	<ul style="list-style-type: none"> Variation of system configuration specifications to establish the system configurations that yield (1) minimum total cost and (2) minimum energy usage under the requirement of economic viability 	JPL will perform the technical work required to exercise the system models according to the optimization criterion developed by Sunkist
II.D	Implementation	<ul style="list-style-type: none"> Assessment of impact of implementation of alternative systems on existing processing plant; acquisition time phasing, construction scheduling and planning, long term uncertainty analysis 	JPL will provide technical support regarding system characteristics
II.E	Evaluation and Selection	<ul style="list-style-type: none"> Selection of a system to be put into use based on: <ul style="list-style-type: none"> total cost function rate of return on capital implementation options uncertainty assessment 	JPL will assist Sunkist in establishing the quantitative selection criteria

Table 1. (continued)

<u>TASK NUMBER AND TITLE</u>	<u>DESCRIPTION OF ACTIVITY</u>	
<u>PHASE II</u>	<u>SYSTEMS EVALUATION</u>	<u>JPL RESPONSIBILITY</u>
II.E (continued)	<ul style="list-style-type: none"> ● energy conservation assessment ● capital requirements and risk analysis 	
II.F Technology Transfer	<ul style="list-style-type: none"> ● Assessment of applicability of study methodology, techniques, and optimum systems to other food processing industries, and communication of study results to food processing industry and technical community via: <ul style="list-style-type: none"> ● industry workshop or technical conference discussion ● project report ● technical paper ● short course (?) 	JPL and Sunkist will share equal responsibility in preparing for and carrying out the Technology Transfer Task as described

5. PROGRESS OF TECHNICAL WORK

5.1 Project Status

The first Technical Progress Report was presented on June 15, 1980 covering the work done from October 1, 1979 to April 1, 1980. This second Technical Progress Report covers the work done from April 1, 1980 to March 1, 1981.

It was found best, in Phase I, to establish realistic, specific plant parameters and energy costs rather than to work in generalities or inconsistent assumptions. The typical values for the Sunkist Growers, Inc. plant in Ontario, California, were chosen. These parameters are listed in Table 2.

Specifications, performance parameters, flow diagrams, capital costs and economic characteristics have been developed for each of the following systems:

- Central Energy Supply with Brayton Engine
- Central Energy Supply with Rankine Engine
- Central Energy Supply with Reciprocating Engine
- Peel Dryer with Brayton Engine
- Peel Dryer with Reciprocating Engine
- Dual-Mode Juice Evaporator
- Thermal Evaporator with Steam Turbine
- Heat Pump for Pasteurization

5.2 Work Completed -- Task I.A - Requirements

As shown in Table 1, Task I.A includes identifying the environmental and the product quality requirements. The June 15, 1980, Technical Report presented a projection of the future prices for fuel and electrical energy.

Table 2: Plant parameters and energy costs
(Typical Values, Sunkist OPD Plant, Oxnard, CA)

Plant operating time, hrs/yr	6,024
Dryer operating time, hrs/yr	4,500
Fruit processing load, tons/yr	600,000
Steam usage, lbs/yr	580×10^6
Electricity consumption, kW-hrs/yr	51×10^6
Average electricity cost, \$/kW-hr	0.06
Average electricity credit rate ^a , \$/kW-hr	0.06
Natural gas cost, \$/therm ^b	
Normal users rate	0.38
Cogeneration rate	0.35
Fuel oil cost ^c , \$/therm	0.758

^aCalifornia cogeneration rate, January, 1981

^b1 therm = 10^5 Btu

^cJanuary, 1981

5.2 Work Completed -- Task I.A - Requirements (Cont'd)

It gave the basic air pollution requirements by the California South Air Quality Management District, the product quality constraints and established the total cost function for economic evaluation purposes.

5.2.1 Energy Price Projection

The price of diesel fuel and natural gas in California increased too rapidly in 1979. As a result, the price leveled off in 1980. This is not reflected in our projections in the June 15th Technical Report. However, the longer range projection is still valid.

In California, Governor Brown is backing the Co-generation movement, but the proposed favorable natural gas rate for Co-generation is still held up in litigation. The FERC has adopted a standard for efficiency for all new oil and natural gas fired topping-cycle Co-generation systems. (18CFR Part 292 March 20, 1980). A qualifying facility must meet the following efficiency criteria: Useful power + $\frac{1}{2}$ useful thermal energy = 42.5 percent of the fuel energy input.

This standard requires the electrical production and heat recovery systems to be fairly efficient. A gas turbine - generator set with an overall efficiency of 15% would not meet the standard unless the waste heat boiler had at least a 64.7% thermal efficiency.

5.2.2 Atmospheric Emission Requirements

The June 15, 1980, Technical Report covered the current regulations as applies in the California South Coast Air Quality Management District. The rules in Central California are essentially the same as in the S.C.A.Q.M.D.: 15 lbs./hr. and 150 lbs./day max. for specific pollutant (eg. NO_x). Best Available Control Technology (BACT) is required to stay below that pollution rate. Gas turbines, except for smaller sizes, are required to have water injection to control NO_x . (See Appendix A for average emissions from turbines and three types of reciprocating engines).

5.2.2 Atmospheric Emission Requirements (Cont'd)

Diesel engines are currently not required to use BACT because the technology is not perfected. However, the California ARB is heading in the direction of requiring ammonia injection and a catalytic converter on diesels to control NO_x by 1983.

The air pollution rules of The State of Florida are generally less stringent than in Southern California. Installing gas turbines, or even diesels, as Co-generation engines in citrus processing, would pose no problem with Florida's air pollution regulations.

5.3 Work Completed - Characterization and Thermodynamic Analysis

The Technical Progress Report, dated June 15, 1980, described the thermodynamics of the Heat Pump and Thermal Evaporator in some detail. It also covered the thermodynamics of co-generation with the Brayton and Diesel engines. This report elucidates the thermodynamics of the peel dryer and waste heat evaporator.

5.3.1 Existing Peel Drying System - Process Description

In a citrus fruit processing plant, the peel, seeds and pulp are normally conveyed to a section of the plant called the "Feed Mill," where they are converted into a valuable cattle feed. A wet peel surge bin is used to stabilize the peel drying operation. After the peel (including seeds and pulp) is ground, a metered amount of calcium hydroxide is added to coagulate the pectin. This releases the liquid, making pressing more effective. The press liquor is pumped to the waste heat evaporator to be concentrated. The press cake (70 to 80% moisture) is fed to the peel dryer to produce a dry product (8-10% moisture). The concentrated press liquor may be sold separately, or it may be mixed with the wet feed to the peel dryer, and thus increase the yield of dry product.

Citrus peel dryers consume more fuel than any other single operation in citrus processing. The typical dryer is a rotary drum with an oil or gas fired furnace supplying hot air to the drum. The wet peel is fed into the hot end of the drum, i.e.: the flow is parallel. The peel comes in direct contact with the combustion products of the fuel. The moisture of the dried peel is related to the temperature of the dryer exhaust gases. A thermometer in the exhaust is used to control the fuel flow to the dryer furnace. The burner is designed to maintain the proper air/fuel ratio (about 17/1 on natural gas).

5.3.2 Existing Peel Drying System -- Material and Energy Balance

A very common peel dryer in the citrus processing industry is the Gulf Machinery Company single pass, rotary drum dryer rated at 60,000 lbs./hr.

5.3.2 Existing Peel Drying System -- Material and Energy Balance (Cont'd)

evaporation. Since Sunkist Growers, Inc., operates such a dryer at the plant in Ontario, California, it was possible to obtain actual data for this study.

The schematic of the dryer system is shown in Figure 4. The system boundary, the material and heat inputs, and out puts, are also shown in this Figure.

The data collected for process characterization are:

<u>Description</u>	<u>Units</u>	<u>Value</u>
Water removal from peel	lbs./hr.	60,000
Wet bulb temp., exhaust gas	°F.	260
Dry bulb temp., exhaust gas	°F.	185
Moisture of peel in	wt. %	77
Moisture of product out	wt. %	8
Heat loss (insulated)	% of fuel	2
Temp. of material entering	°F.	60
Temp. of product out	°F.	200

The material and energy balance were calculated as follows:

<u>Material Input -</u>	<u>Mass Rate lbs./Hour.</u>
Fuel	F
Air	A
Press Cake	P

Material Output -

Dried Pulp	M
------------	---

Exhaust Gas - The combustion constants of the fuel are obtained from the literature and composition can be expressed as;

<u>Combustion Products (using natural gas)</u>	<u>lbs./Hour</u>
Water	1.87 F
Carbon Dioxide	2.52 F

GULF DRYER - MATERIAL IN-PUTS AND OUT-PUTS

FIGURE 4

5.3.2 Existing Peel Drying System -- Material and Energy Balance (Cont'd)

<u>Combustion Products (Cont'd)</u>	<u>lbs./Hour</u>
Nitrogen	11 F
Excess Air	A - 14.39 F
Water Removed from Peel	60,000

Since the peel and dried pulp moisture contents are known, together with the total evaporation rate, the mass rate of peel and dried pulp can be calculated as shown below:

Total Material Balance

$$\text{Peel} = \text{Dried Pulp} + 60,000$$

$$P = M + 60,000 \quad (1)$$

Solid Balance

$$\text{Solids in Press Cake} = \text{Solids in Dried Pulp}$$

$$0.23P = 0.92 M$$

$$\text{or } P = 4 M \quad (2)$$

Substituting equation (2) into equation (1) yields:

$$P = 80,000 \text{ lbs./hr.}$$

$$M = 20,000 \text{ lbs./hr.}$$

The composition of exhaust gas is known in terms of fuel and combustion air rate, so the humidity of exhaust gas can be expressed as:

$$\text{Humidity, } \frac{\text{Pounds of Water}}{\text{Pounds of Dry Air}} = \frac{1.87 F + 60,000}{A - 0.87 F} \quad (3)$$

The expression for Dry Air and Water is obtained as:

<u>Dry Air</u>	<u>Mass Rate lbs./hour</u>
Carbon Dioxide	2.52 F
Nitrogen	11.0 F
Excess Air	A - 14.39 F
Total	A - 0.87 F

5.3.2 Existing Peel Drying System -- Material and Energy Balance (Cont'd)

<u>Water</u>	<u>Mass Rate</u> <u>lbs./hour</u>
Combustion Water	1.87 F
Evaporated Water	60,000
Total	1.87 F + 60,000

Since the dry and wet bulb temperatures of exhaust gas is known, the humidity may be obtained from the psychrometric tables. In this case, it is 0.7897 pounds dry air.

Substituting the value of humidity in equation (3) and re-arranging yields:

$$A - 3.24 F = 75.978 \quad (4)$$

An energy balance yields:

$$F \times (\text{Fuel HHV, BTU/lb.}) + \text{Feed} \times (\text{Heat Content*}, \text{BTU/lb.}) = \\ (\text{Components leaving dryer, lbs./hr.}) \times (\text{Heat Content*}, \\ \text{BTU/lb.}) + \text{Heat Loss (BTU/hr.)}$$

The material, inputs and outputs, mass rate and heat content are shown in the following table.

*Heat contents use 60°F. as datum point. We assume the feed to be at 60°F., so its heat content is zero.

5.3.2 Existing Peel Drying System -- Material and Energy Balance (Cont'd)

Table 3: Peel Dryer Mass Balance
and Heat Contents of Reactants

<u>Component</u>	<u>Mass Rate lbs./hour</u>	<u>Heat Content Above 60°F. BTU/lb.</u>
Fuel	(From Eq. 6)	20,000
Air	(From Eq. 6)	Ø
Press Cake	80,000	Ø
Dried Pulp	20,000	84.6
Exhaust Gas		
Combustion Water	1.87 F	1,139.24
Carbon Dioxide	2.52 F	44.1
Nitrogen	11.0 F	51.9
Excess Oxygen	0.23 (A - 14.39 F)	45.4
Excess Nitrogen	0.77 (A - 14.39 F)	51.9
Evaporated Water	60,000	1,139.24

Substituting the products of each mass rate times its heat content in Equation (5) and re-arranging yields:

$$347.51 \bar{F} - A = 1.39 \times 10^6 \quad (6)$$

Solving Equation (4) and Equation (6) simultaneously, yield:

$$F = 4,255 \text{ lbs./hr. of fuel}$$

$$A = 89,765 \text{ lbs./hr. of air}$$

The exhaust gas mass flow equals the sum of the fuel, air and water evaporated. $(4,255 + 89,764 + 60,000 = 154,019 \text{ lbs./hr.})$

The material and heat balances of the dryer/W.H.E. are shown in Figure 5.

Energy conservation in citrus peel drying took a giant step forward with the development of the waste heat evaporator. This is an evaporator for peel press liquor and selected wastewater streams. The heat supply to

FIGURE 5

Summary of Material and Heat
Balances of Dryer/WHE System

DESCRIPTION	MASS RATE lb./hr.	HEAT RATE $\times 10^6$ BTU/hr.
DRYER - 60,000 lb./hr. Evaporation		
1. Fuel	4,255	85.1
2. Air	89,760	—
3. Peel	80,000	—
4. Meal	20,000	—
5. Exhaust Gas	154,020	81.7
WASTE HEAT EVAPORATOR - 60,000 lb./hr.		
6. Press Liquor	75,000	0.7
7. Conc. Press Liquor	15,000	1.6
8. Black Water	36,030	4.0
9. Clean Condensate	30,800	2.6
10. Vapors to Condensor	29,200	36.7
11. Stack Gas	118,000	37.5

DRYER - WHE SYSTEM

5.3.2 Existing Peel Drying System -- Material and Energy Balance (Cont'd)

The evaporator is the dryer exhaust. Typically the waste heat evaporator (W.H.E.) will have about the same evaporating capacity as the peel dryer. With the advent of the W.H.E., the fuel requirement for drying peel was cut in half. By investing in more tube surface, it is possible to obtain almost twice the evaporation on the waste heat evaporator as on the dryer.

The specifications and performance parameters for the existing peel dryer system are listed on Table 4.

TABLE 4: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
EXISTING PEEL DRYER SYSTEM

DRYER

Evaporation, Lb./Hr.	60,000
Air Leakage, Lb./Hr.	16,290
Drum Entry Temp. °F:	1,065
Feed Peel	
Temperature, °F.	60
Moisture, %	77
Rate, Lb./Hr.	80,000
Product Peel	
Temperature, °F.	200
Moisture, %	8
Rate, Lb./Hr.	20,000
Exhaust	
Temperature, °F.	260
Flow, acfm	58,710
Humidity, Lb. W.V./Lb. D.A.	0.7896
Wet Bulb, °F.	185

BURNER

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	4,255
Heat Input, BTU/Hr.	85.1×10^6
Heat Loss, % Fuel Heat	2
Air	
Rate, Lb./Hr.	73,470
Air to Fuel Mass Ratio	17.3
Exhaust	
Rate, Lb./Hr.	77,730
Heat Output, BTU/Hr.	83.4×10^6
Humidity, Lb. W.V./Lb. D.A.	0.1138

ENGINE

None

5.4 Work Completed - Task IC - IE, Thermodynamics, Synthesis and Preliminary Evaluation of Systems

The previous technical report covered the thermodynamics of the

- o Heat pump (open and closed cycle)
- o Thermal evaporator
- o Brayton engine
- o Diesel engine

The current report includes the above, but takes the step to co-generation by the use of the

- o Brayton engine
- o Steam turbine
- o Reciprocating gas engine

This Phase I Technical Report combines the Tasks IC, ID and IE into this section. Each system is presented in an integrated fashion, rather than to divide it into thermodynamic analysis, synthesis and preliminary economic evaluation.

5.4.1 Central Brayton Co-generation

The use of a combustion turbine to generate electricity and steam from the same fuel is a form of co-generation that is being used successfully in some industries. It's feasibility in the citrus processing industry depends on many factors, some of which have become more favorable with the passage of laws affecting the relationships between co-generators and the public utilities.

A co-generation system using a 3.2 MW gas turbine was studied and the specifications and performance parameters are shown in Table 5. The schematic for the Central Brayton Co-generation System is illustrated in Figure 6. This system has an engine heat rate of 12,000 BTU/KWH, and an overall fuel efficiency of: $\frac{(3060)(3415) + 17.5 \times 10^6}{100 \times 38.4 \times 10^6} = 72.8\%$ The ratio of the useful BTUs and KWH is $17.5 \times 10^6 / 3060 = 5,719$ BTU/KWH.

A gas turbine works best at full load and continuous operation. The useful heat from the exhaust should be matched to the base heat load of the plant, not to the peaks. It is important to study the steam use profile in detail.

**Table 5: System specifications and performance parameters
for Central Brayton cogeneration
(60°F, Sea level)**

Gas turbine rated continuous power output, kW(e)	3,200
Engine heat rate, Btu/kW-hr	12,000
Fuel energy input to engine, 10^6 Btu/hr	38.4
Engine heat loss, 10^6 Btu/hr	0.5
Engine exhaust temperature, °F	950
Energy in engine exhaust, 10^6 Btu/hr	27.0
Waste-heat boiler efficiency, %	65
Heat to boiler exhaust, 10^6 Btu/hr	9.5
Heat transferred to steam, 10^6 Btu/hr	17.5
Steam production rate, lb/hr	17,100
Boiler feedwater temperature, °F	175
Enthalpy of feedwater, Btu/lb	145
Gear box efficiency, %	98.5
Generator efficiency, %	95
Net electrical power output, kW(e)	3,060
Steam to plant processes:	
Steam pressure, PSIA	40
Steam temperature, °F	267
Steam quality, %	100
Enthalpy, Btu/lb	1,170

CENTRAL BRAYTON CO-GENERATION

5.4.1 Central Brayton Co-generation (Cont'd)

Time increments should be no more than hourly. It is necessary to calculate the operating hours per year for the engine at full load. One should not plan to start and stop the engine more frequently than daily.

The Capital Cost Breakdown for the Central Brayton Co-generation System is shown in Table 6. The installed cost of a 3,000 KW system including waste heat boiler is \$509/KW.

The Economic Characteristics of the Central Brayton Co-generation System are given in Table 7. The payback period is 2.25 yrs. and the internal rate of return is 45% using a capital cost factor of 0.0805.

5.4.2 Central Rankine Co-generation

Steam turbine co-generation was used in the early days of the citrus processing industry, but became out-of-fashion with low energy costs. With the promulgation of Section 210 of Public Utility Regulatory Policies Act, and the increases in energy costs, all co-generation has again come alive.

Steam turbines are most efficient when there is the greatest difference between the inlet and outlet pressures. For Central Rankine Co-generation this study uses a boiler pressure of 650 psia and an exhaust pressure of 40 psia. The turbine output is approximately 4 MW with a net power from the generator of 3,760 KW. The complete specifications and performance parameters for the system are shown in Table 9. The schematic drawing for the Central Rankine Co-generation System is given in Figure 7. The ratio of useful heat and power output is $(1170)(80,000)/3760 = 24,890$ BTU/KWH.

The Capital Cost Breakdown for the Central Rankine Co-generation System is listed in Table 10. The installed cost of the 4 KW turbine generator set and high pressure boiler is \$465/KW.

The Economic Characteristics of the Rankine Co-generation System is given in Table 11. The payback period is about 2.6 years and the internal rate of return is 40% when using a capital cost factor of 0.0805. Table 12, Energy Savings for Central Rankine Co-generation shows an overall energy savings of 10%

Table 6. Capital cost breakdown for
Central Brayton cogeneration system

<u>3060 kW</u>	
<u>Cost Item</u>	$10^3 \$$
Energy conversion unit	980
Turbine-generator unit with accessories and controls	780
Heat recovery boilers	200
Generator substation & connections	117
Foundation and installation	412
Automatic monitoring & control system	84
Subtotal	1,593
Engineering and Supervision (10%)	159
Total cost of installed system	1,752
Installed cost per unit capacity	573/kW

Table 7. Economic characteristics of Central Brayton system

Electrical generation capacity, MW	3.06	6.0	9.0	12.0	15 MW
Capital, 10 ⁶ \$	1.75	3.44	5.2	6.9	8.6
Fuel cost	----- Annual Costs, 10 ⁶ \$ -----				
Cogen. firing Natural gas ^a	0.774	1.52	2.28	3.04	3.79
Supplementary firing Natural gas ^b	2.42	1.92	1.40	0.89	0.38
Purchased electricity cost	1.95	0.89	-0.19	-1.28	-2.36
Total energy cost	5.14	4.32	3.49	2.65	1.81
Total energy cost reduction ^c	0.86	1.68	2.51	3.35	4.19
Maintenance cost (\$.0022/kW-hr)	0.04	0.08	0.12	0.16	0.20
Payback ^d /IRR ^{d,e} , yrs/%	←----- 2.6/38 -----→				

a) Nat. gas @ current cogen. rate of \$.35/therm, January 1981.

b) Nat. gas @ std. rate of \$.38/therm, January 1981.

c) Relative to present system shown in Table 2.

d) Capital cost factor of 0.0805.

e) Internal Rate of Return before income taxes.

Table 8: Energy savings for Central Brayton cogeneration

	<u>Baseline System</u>	<u>Brayton System</u>
Steam production rate, lb/hr	17,100	17,100
Fuel energy to steam production ^a , 10 ⁶ Btu/hr	22.7	-
Electrical power demand/output, kW	3,060	3,060
Fuel energy to power production ^b , 10 ⁶ Btu/hr	31.6	38.4
Total fuel energy consumption, 10 ⁶ Btu/hr	54.3	38.4
Energy savings, 10 ⁶ Btu/hr	-	15.9
Percent energy savings	-	29.3

^aBaseline case steam raised at 75 lb/therm

^bUtility generated electricity delivered @ 33% overall efficiency

Table 9: System specifications and performance parameters
for Central Rankine cogeneration
(Two 40,000 lb/hr boilers)

Boiler size, lb/hr	80,000
Boiler exit steam temperature, °F	750
Boiler steam pressure, PSIA	650
Condensate return fraction	0.90
Condensate temperature, °F	185
Make-up water temperature, °F	90
Boiler feedwater temperature, °F	175
Enthalpy increase of steam across boiler, Btu/lb	1,229
Boiler efficiency, %	75
Fuel energy input to boiler, 10 ⁶ Btu/hr	131
Steam turbine efficiency, %	65
Turbine exhaust pressure, PSIA	40
Isentropic enthalpy drop across turbine, Btu/lb	264
Turbine power output, kW	4,020
Water pump efficiency, %	70
Pumping power, Kw	60
Gearbox efficiency, %	98.5
Generator efficiency, %	95
Net electrical power output, ww(e)	3,760
Steam to plant processes:	
Pressure, PSIA	40
Temperature, °F	267
Quality, %	100
Enthalpy, Btu/lb	1,170

FIGURE 7: CENTRAL RANKINE CO-GENERATION

Table 10: Capital cost breakdown for Central
Rankine cogeneration system

3760 kW

<u>Cost Item</u>	<u>10³\$</u>
Energy conversion unit	940
Turbine-generator unit with accessories and controls	490
High pressure boiler	450
Generator substations & connections	117
Foundation and installation	450
Automatic monitoring & control system	<u>84</u>
Subtotal	1,591
Engineering and supervision (10%)	<u>159</u>
Total cost of installed system	1,750
Installed cost	\$465/kW

Table 11. Economic characteristics of Central Rankine co-generation system

Electrical generation capacity, MW	1.88	2.82	3.76	4.5	4.7	5.64
Capital, 10 ⁶ \$	0.875	1.311	1.75	2.09	2.19	2.625
Fuel cost	----- Annual Costs, 10 ⁶ \$ -----					
Cogen. firing Natural gas ^a	1.38	2.07	2.76	3.31	3.46	4.15
Supplementary firing Natural gas ^b	1.72	1.11	0.497	0.016	-0.113	- .724
Purchased electricity cost	2.38	2.04	1.7	1.43	1.36	1.02
Total energy cost	5.48	5.22	4.96	4.76	4.7	4.45
Total energy cost reduction ^c	0.52	0.78	1.04	1.24	1.3	1.55
Maintenance cost (\$0.0094/kWhr)	0.106	0.160	0.212	0.255	0.266	.318
Payback ^d /IRR ^{d,e} , yrs/1	←----- 2.6/38 -----→					

a) Nat. gas @ current cogen. rate of \$.35/therm, January 1981.

b) Nat. gas @ std. rate of \$.38/therm, January 1981.

c) Relative to present system shown in Table 2.

d) Capital cost factor of 0.0805.

e) Internal Rate of Return before income taxes.

Table 12: Energy savings for Central Rankine cogeneration

<u>Parameters</u>	<u>Baseline System</u>	<u>Rankine System</u>
Steam production rate, lb/hr	80,000.	80,000.
Full energy to steam production ^a , 10^6 Btu/hr	107.	131.
Electrical power demand/production	3760.	3760.
Fuel energy to power production ^b , 10^6 Btu/hr	39.	-
Total fuel energy input, 10^6 Btu/hr	146.	131.
Energy savings, 10^6 Btu/hr		15.
Percent energy savings		10.

^aBaseline case steam raised @ 75 lb/therm

^bUtility generated electricity delivered 32@ 33% overall efficiency

5.4.3 Central Reciprocating Engine Co-generation

The reciprocating engine has several characteristics in its favor. The engine heat rate, BTU/Hp hr. is low, reflecting its high work to fuel efficiency. It also can be run at part loads without a serious loss of efficiency. The low RPM models have few maintenance problems.

However, reciprocating engines have a high capital cost, especially the high horsepower models. The heat recovery from the cooling jacket is at a relatively low temperature, which at times is useless. The exhaust is at a high temperature, but the mass flow is small. The schematic drawing for this system is shown in Figure 8.

The specifications and Performance Parameters for the Central Reciprocating Engine Co-generation are given in Table 13. This is for 1254 KW output and 2200 lbs./hr. steam production. The useful heat and power output have a ratio of $2.3 \times 10^6 / 1254 = 1,834$ BTU/KWH.

The Capital Cost Breakdown for the Central Reciprocating Engine Co-generation System is presented in Table 14. The installed cost for three 1254 KW engine-generator units with heat recovery boilers is \$1,165/KW. Table 15 gives the economic data on this system. The payback is 17 years with an unacceptable 1.5% internal rate of return. Table 16 shows a high overall energy savings of 11.8%.

5.4.4 Peel Dryer With Brayton Engine

The combustion gas turbine has an exhaust close to the proper temperature to enter the feed end of a peel dryer. The most simple arrangement would be the one shown in the schematic drawing, Figure 9, "Peel Dryer With Brayton Engine - Parallel Mode." The engine could provide any amount of the required heat up to 100 percent. The supplementary burner would act as a modulating control. The specifications and performance parameters for the Brayton engine only are shown in Table 17.

For 60,000 lbs./hr. of evaporation the turbine would generate almost 10 MW. However, moisture control of the product would be difficult without some

FIGURE 8: CENTRAL RECIPROCATING ENGINE
CO-GENERATION

Table 13. System specifications and performance parameters
for Central Reciprocating engine cogeneration
(One 1250 kW unit)

Engine power output, BHP	1,770
Engine speed, RPM	600
Engine heat rate, Btu/HP-hr.	8,000
Fuel energy input to engine, 10^6 Btu/hr.	14.2
Engine heat loss, 10^6 Btu/hr.	1.1
Engine exhaust temperature, $^{\circ}\text{F}$.	885
Energy in engine exhaust, 10^6 Btu/hr.	4.3
Engine efficiency (heat to work), %	31.8
Engine cooling (jacket + intercooler + lube oil)	4.3
Boiler exit temperature, $^{\circ}\text{F}$.	300
Heat transferred to steam, 10^6 Btu/hr.	2.3
Steam production rate, lb./hr.	2,200
Boiler feedwater temperature, $^{\circ}\text{F}$.	175
Enthalpy of feedwater, Btu/lb.	145
Generator efficiency (direct drive)	95
Net electrical power output, kW(e)	1,254
Steam to plant processes:	
Pressure, PSIA	40
Temperature $^{\circ}\text{F}$.	267
Quality, %	100
Enthalpy	1,170

Table 14: Capital cost breakdown for central
Reciprocating Engine cogeneration system

600 RPM/3762 kW
(Three 1254/kW Units)

Cost Item

Energy conversion units

(3) Engine-generator units with
accessories and controls

(3) Heat recovery boilers

Generator substation & connections

Foundation and installation

Automatic monitoring & control system

Subtotal

Engineering and supervision (10%)

Total cost of installed systems

Installed cost

\$x10³

2475.

117.

1240.

150.

3983.

398.

4381.

\$1165/kW

Table 15 . Economic characteristics of Central Reciprocating engine cogeneration
600 RPM engine

Electrical generation capacity, MW	1.2	3.76	6.0	8.4	12.0 MW
Capital, 10 ⁶ \$	1.40	4.38	6.99	9.79	14.0
- - - - - Annual Costs, 10 ⁶ \$ - - - - -					
Fuel cost	0.271	.851	1.36	1.90	2.71
Cogen. firing					
Natural gas ^a					
Supplementary firing	2.87	2.73	2.61	2.48	2.29
Natural gas ^b					
Purchased electricity cost	2.63	1.70	0.89	0.024	-1.28
Total energy cost	5.77	5.29	4.86	4.4	3.72
Total energy cost reduction ^c	0.23	.714	1.14	1.60	2.28
Maintenance cost (\$.0049/kW-hr)	0.035	0.111	0.177	0.247	0.354
Payback ^d /IRR ^{d,e} , yrs/%	← 17.7 < 2 →				

a) Nat. gas @ current cogen. rate of \$.35/therm, January 1981.

b) Nat. gas @ std. rate of \$.38/therm, January 1981.

c) Relative to present system shown in Table 2 .

d) Capital cost factor of 0.0805.

e) Internal Rate of Return before income taxes.

Table 16 : Energy savings for Central Reciprocating engine cogeneration

	<u>Baseline System</u>	<u>Engine System</u>
Steam production rate, lb/hr	2,200	2,200
Fuel energy to steam production ^a , 10 ⁶ Btu/hr	3.1	-
Electrical power demand/output, kW	1,254	1,254
Fuel energy to power production ^b , 10 ⁶ Btu/hr	13.0	14.2
Total fuel energy consumption, 10 ⁶ Btu/hr	16.1	14.2
Energy savings, 10 ⁶ Btu/hr	-	1.9
Percent energy savings	-	11.8

^aBaseline case steam raised @ 75 lb/therm

^bUtility generated electricity delivered @ 33% overall efficiency

FIGURE 9: PEEL DRYER WITH BRAYTON ENGINE
PARALLEL MODE

TABLE 17: SYSTEM SPECIFICATION AND PERFORMANCE PARAMETERS
FOR BRAYTON ENGINE ONLY

DRYER

Evaporation, Lb./Hr.	60,000
Air Leakage, Lb./Hr.	16,289
Drum Entry Temperature, °F.	910
Feed Peel	
Temperature, °F.	60
Moisture, %	77
Rate, Lb./Hr.	80,000
Product Peel	
Temperature, °F.	200
Moisture, %	8
Rate, Lb./Hr.	20,000
Exhaust	
Temperature, °F.	260
Flow, acfm	154,700
Humidity, Lb. W.V./Lb. D.A.	0.185
Wet Bulb, °F.	151

BURNER

None

ENGINE

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	7,089
Heat Input, BTU/Hr.	141.8×10^6
Air	
Rate, Lb./Hr.	385,000
Air to Fuel Mass Ratio	54.3
Exhaust	
Rate, Lb./Hr.	392,000
Heat Output, BTU/Hr.	105.1×10^6
Temperature, °F.	910
Humidity, Lb. W.V./Lb. D.A.	0.035
Power	
Efficiency, %	23.9
Electrical Work Output, KW	9,929

5.4.4 Peel Dryer With Brayton Engine (Cont'd)

supplementary burner firing. The exhaust flow from the dryer would be more than twice that of the present system. As shown in Table 18 the overall energy savings would be 12.2%.

One of the problems with the Brayton engine in the parallel mode is that the dryer exhaust humidity would be very low compared to the existing dryer system. A high humidity is very important for the efficient operation of the waste heat evaporator. The existing system recycles at least half of the exhaust back to the furnace, minimizing the fresh air, which would otherwise lower the humidity. As emphasized in Paragraph 5.3.2, the waste heat evaporator is an essential energy saving part of the peel drying system.

Figure 10 shows the relationship between the size of the gas turbine and the dryer exhaust humidity.

A small Brayton engine in the parallel mode would not lower the humidity excessively, but the electrical output would be hardly worth the effort. This is illustrated in Table 19 "System Specifications and Performance Parameters for Brayton Engine Parallel With Direct Firing." Table 20 shows the overall energy savings of the parallel mode to be - 4.8%. This is because the small co-generation gain is more than overcome by the loss in waste heat evaporator performance.

As shown in Figure 10 , the Series Mode causes only a slight reduction of humidity below about 2 MW of power. The Series Mode uses an afterburner in the turbine exhaust to provide the additional heat required to do the drying. The combustion gas turbine uses a great amount of excess air, so the exhaust can be thought of as preheated air to the supplementary burner. Recycle may be used almost to the same extent as with the existing system. The waste heat evaporator performance is only very slightly affected. The specifications and performance parameters for the Series Mode Brayton are shown in Table 21. The schematic flow diagram, Figure 11 illustrates the system.

Table 22 lists the capital cost data for the Brayton Series Mode Co-generation on the peel dryer. The installed cost, including the afterburner

Table 18: Energy Savings for Brayton Engine Only Adapted to Citrus Peel Dryer

	<u>Baseline System</u>	<u>Brayton Only</u>
Evaporation rate in dryer, lb./hr.	60,000	60,000
Evaporation rate in WHE, lb./hr.	60,000	--
Evaporation rate in multi-effect evaporator, lb./hr.		60,000
Fuel energy input to dehydrating, 10^6 Btu/hr.	85	23
Electrical power demand/output, KW	9,929	9,929
Fuel energy to electrical power production, 10^6 Btu/hr.	103(a)	142
Total fuel energy consumption, 10^6 Btu/hr.	188	165
Energy savings, 10^6 Btu/hr.	--	23
Percent energy savings	--	12.2

(a) Utility generated electricity delivered @ 33% overall efficiency.

FIGURE 10
GAS TURBINE SIZE
VS.
DRYER EXHAUST HUMIDITY

TABLE 19: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
BRAYTON ENGINE PARALLEL WITH DIRECT FIRING

DRYER

Evaporation, Lb./Hr.	60,000
Air Leakage, Lb./Hr.	16,290
Feed Peel	
Temperature, °F.	60
Moisture, %	77
Rate, Lb./Hr.	80,000
Product Peel	
Temperature, °F.	200
Moisture, %	8
Rate, Lb./Hr.	20,000
Exhaust	
Temperature, °F.	260
Flow, acfm	61,220
Humidity, Lb. W.V./Lb. D.A.	0.7221
Wet Bulb, °F.	183

BURNER

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	4,146
Heat Input, BTU/Hr.	82.92×10^6
Heat Loss, % Fuel Heat	2
Air	
Rate, Lb./Hr.	71,590
Air to Fuel Mass Ratio	17.3
Exhaust	
Rate, Lb./Hr.	86,140
Heat Output, BTU/Hr.	84.0×10^6
Humidity, Lb. W.V./Lb. D.A.	0.1138

ENGINE

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	188
Heat Input, BTU/Hr.	3.76×10^6
Heat Loss, % of Fuel Heat	2
Air	
Rate, Lb./Hr.	10,210
Air to Fuel Mass Ratio	54.3
Exhaust	
Rate, Lb./Hr.	10,400
Heat Output, BTU/Hr.	2.8×10^6
Temperature, °F.	910
Humidity, Lb. W.V./Lb. D.A.	0.035
Power	
Efficiency, %	23.9
Electrical Work Output, KW	263

Table 20: Energy savings for Brayton engine in parallel
with burner for Citrus Peel Dryer

	<u>Baseline System</u>	<u>Brayton in Parallel</u>
Evaporation rate in dryer, lb/hr	60,000	66,300
Evaporation rate in WHE, lb/hr	60,000	53,700
Fuel energy to burner, 10^6 Btu/hr	85.1	88.2
Electrical power demand/output, kW	263	263
Fuel energy to electrical power production ^a , 10^6 Btu/hr	2.7	3.8
Total fuel energy consumption, 10^6 Btu/hr	87.8	92.0
Energy savings, 10^6 Btu/hr	-	-4.2
Percent energy savings	-	-4.8

^aUtility generated electricity delivered @ 33% overall efficiency

TABLE 21: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS ON
BRAYTON ENGINE WITH AFTERBURNER

DRYER

Evaporation, Lb./Hr.	60,000
Air Leakage, Lb./Hr.	16,290
Drum Entry Temperature, °F.	1,069
Feed Peel	
Temperature, °F.	60
Moisture, %	77
Rate, Lb./Hr.	80,000
Product Peel	
Temperature, °F.	200
Moisture, %	8
Rate, Lb./Hr.	20,000
Exhaust	
Temperature, °F.	260
Flow, acfm	61,220
Humidity, Lb. W.V./Lb. D.A.	0.7385
Wet Bulb, °F.	183

BURNER

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	3,195
Heat Input, BTU/Hr.	63.9×10^6
Heat Loss, % Fuel Heat	2
Air	
Rate, Lb./Hr.	0
Air to Fuel Mass Ratio	0
Exhaust	
Rate, Lb./Hr.	85,590
Heat Output, BTU/Hr.	84.7×10^6
Temperature, °F.	0
Humidity, Lb. W.V./Lb. D.A.	0.1214

ENGINE

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	1,490
Heat Input, BTU/Hr.	29.8×10^6
Heat Loss, % of Fuel Heat	2
Air	
Rate, Lb./Hr.	80,910
Air to Fuel Mass Ratio	54.3
Exhaust	
Rate, Lb./Hr.	82,400
Heat Output, BTU/Hr.	22.1×10^6
Temperature, °F.	910
Humidity, Lb. W.V./Lb. D.A.	0.035
Power	
Efficiency, %	23.9
Electrical Work Output, KW	2,087

FIGURE 11: PEEL DRYER WITH BRAYTON ENGINE
SERIES MODE

Table 22 : Capital cost breakdown for Dryer cogeneration
via Brayton engine with afterburner supplying
hot gas to the dryer drum

2100/kW

Cost Item

Energy conversion unit	602
Turbine-generator unit with accessories and controls	402
Afterburner and controls	200
Generator substation & connections	117
Foundation and installation	412
Automatic monitoring & control system	84
Subtotal	1,215
Engineering and Supervision (10%)	122
Total cost of installed system	1,337
Installed cost per unit capacity	\$ 640/kW

5.4.4 Peel Dryer With Brayton Engine (Cont'd)

and controls is \$640/KW. Table 23 gives the economic information on this system. The payback is only 4.7 years and the internal rate of return is 21%. The overall energy savings shown in Table 24 is 12.1%.

5.4.5 Peel Dryer With Reciprocating Engine

The reciprocating engine can co-generate with a peel dryer without affecting the humidity of the dryer exhaust very much. This is because the mass flow of the engine exhaust is low. Table 25 shows the technical data for a 870 KW generating system using a reciprocating engine. The dryer exhaust retains a high humidity, 0.786 lbs. WV/lb. D.A.

Figure 12 illustrates, with a schematic flow diagram, how the engine block cooling is used to preheat the air to the burner. The engine exhaust enters the furnace in a parallel mode.

Table 26 shows the capital cost of an 870 KW system installed with an air preheater to be \$1,109/KW. Table 27 gives the payback as 9.1 years, and the ROR as 9%. Table 28 shows the overall energy savings of the system to be 7.3%.

5.4.6 Thermal Evaporator With Steam Turbine

Most boilers in the citrus processing industry produce steam at pressures of 150-250 psia. A major steam load is evaporation which requires steam at atmospheric or even sub-atmospheric pressure. In order to reduce boiler-steam pressure to that suitable for evaporation, steam is made to flow through pressure-reducing valves and/or orifices. Although theoretically no energy is lost in this process, it is an ineffective means of pressure reduction, because high pressure steam, which has potential to do work, is reduced to low pressure steam which has little potential to do work. A more effective means of pressure reduction is the steam turbine arrangement shown in Figure 13. In this scheme, no orifice or pressure-reducing value is present, but instead, all steam is made to flow through a single-stage steam turbine before it proceeds

TABLE 23. Economic Characteristics of Dryer
Co-generation Via Brayton Engine
with Afterburner

	<u>Present System</u>	<u>Brayton Co-generation</u>
Elect. generation capacity	none	1.952 MW
Additional capital	none	1.337 x 10 ⁶ \$
Operating costs	<u>10⁶ \$/yr.</u>	<u>10⁶ \$/yr.</u>
Co-gen. fuel, (NG. ^a)	--	469
Direct fuel, (NG. ^b)	1,450	1,090
Electricity credit	--	-527
Total energy cost	1,450	1,032
Energy cost reduction	--	418
Additional maintenance cost	--	21
Payback/IRR	--	4.7/21%

a) Natural gas @ current co-gen. rate of \$.35/therm, Jan. 1981

b) Natural gas @ standard rate of \$.38/therm, Jan. 1981

Table 21: Energy savings of Brayton engine with
afterburner adapted to Citrus Peel Dryer

	<u>Baseline System</u>	<u>Brayton/ afterburner</u>
Evaporation rate in dryer, lb/hr	60,000	60,000
Evaporation rate in WHE, lb/hr	60,000	60,000
Fuel energy input to burner, 10^6 Btu/hr	85.1	63.9
Electrical power demand/output, kW	2087	2087
Fuel energy to electrical power production ^a , 10^6 Btu/hr	21.6	29.8
Total fuel energy consumption, 10^6 Btu/hr	106.7	93.7
Energy savings, 10^6 Btu/hr	-	13.0
Percent energy savings	-	12.1

^aUtility generated electricity delivered at 33% overall efficiency

TABLE 25: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
FOR RECIPROCATING ENGINE PARALLEL/DIRECT FIRING

DRYER

Evaporation Rate, Lb./Hr.	60,000
Air Leakage Rate, Lb./Hr.	16,289
Drum Entry Temperature, °F.	1,065
Feed Peel	
Temperature, °F.	60
Moisture, %	77
Rate, Lb./Hr.	80,000
Product Peel	
Temperature, °F.	200
Moisture, %	8
Rate, Lb./Hr.	20,000
Exhaust	
Temperature, °F.	260
Flow, cfm	58,968
Humidity, Lb. W.V./Lb. D.A.	0.786
Wet Bulb, °F.	185

BURNER

Fuel	Natural Gas
Heat Content, BTU/Lb.	20,000
Rate, Lb./Hr.	3,884
Heat Input, BTU/Hr.	77.68×10^6
Air	
Rate, Lb./Hr.	67,187
Air to Fuel Mass Ratio	17.3
Exhaust	
Rate, Lb./Hr.	71,071
Heat Output, BTU/Hr.	80.65×10^6

ENGINE

Fuel	Natural Gas
Heat Content (HHV), BTU/Lb.	20,000
Rate, Lb./Hr.	464
Heat Input, BTU/Hr.	9.28×10^6
Heat Loss, % of Fuel Heat	4
Air	
Rate, Lb./Hr.	6,960
Air to Fuel Mass Ratio	15
Exhaust	
Rate, Lb./Hr.	7,424
Heat Output, BTU/Hr.	5.94×10^6
Temperature, °F.	1,034
Humidity, Lb. W.V./Lb. D.A.	0.132
Power	
Efficiency	32
Electrical Work Output, KW	870

FIGURE 12: PEEL DRYER WITH RECIPROCATING ENGINE
PARALLEL MODE

TABLE 26. Capital Cost Breakdown for Dryer Co-generation
Via Reciprocating Engine in Parallel Mode

<u>870 KW</u>	
<u>Cost Item</u>	<u>\$ x 10³</u>
Engine - generation unit with accessories and controls	394
Generator substation and connections	50
Installation and foundation	310
Automatic monitoring control system	<u>80</u>
Subtotal	834
Engineering and supervision (10%)	<u>83</u>
Total cost of installed system	917
Installed cost	\$1,109/KW

**TABLE 27. Economic Characteristics of Dryer
Co-generation Via Reciprocating
Engine - Parallel Mode**

	<u>Present System</u>	<u>Reciprocating Engine Co-generation</u>
Elect. generation capacity	none	827 MW
Additional capital	none	0.917 x 10 ⁶ \$
Operating costs	<u>10⁶ \$/yr.</u>	<u>10⁶ \$/yr.</u>
Co-gen. fuel (NG.) ^a	--	146
Direct fuel (NG.) ^b	1,450	1,330
Electricity credit	--	-223
Total energy cost	1,450	1,253
Energy cost reduction	--	197
additional maintenance cost	--	23
Payback/IRR		9.1/9%

a) Natural gas @ current co-gen. rate of \$.35/therm, Jan. 1981

b) Natural gas @ standard cost rate of \$.38/therm, Jan. 1981

**Table 28: Energy savings for Reciprocating engine
adapted to Citrus Peel Dryer**

	<u>Baseline System</u>	<u>Cogeneration System</u>
Evaporator rate in dryer, lb/hr	60,000	60,000
Evaporator rate in WHE, lb/hr	60,000	60,000
Fuel energy to burner, 10^6 Btu/hr	85.1	77.7
Electrical power demand/output, kW	870	870
Fuel energy to electrical power production ^a , 10^6 Btu/hr	9.0	9.3
Total fuel energy consumption, 10^6 Btu/hr	94.1	87.0
Energy savings, 10^6 Btu/hr	-	7.1
Percent energy savings	-	7.3

^aUtility generated electricity delivered @ 33% overall efficiency

FIGURE 13: THERMAL EVAPORATOR WITH STEAM TURBINE

5.4.6 Thermal Evaporator With Steam Turbine (cont'd)

to evaporation. The turbine reduces steam pressure and in addition, produces mechanical work. This work is used to produce electrical power.

Table 29 presents specifications and performance parameters for both this scheme and for the present system. Both systems perform the same function, namely supplying energy input necessary to run three, six-effect, 40,000 lb./hr. evaporations. Note; first that the turbine does not reduce boiler steam pressure completely to atmosphere. Turbine exhaust pressures above atmosphere are necessary to allow the steam to flow from the turbine to the evaporator. Also notice, that the steam turbine system consumes more steam than the existing system. This occurs because the quality and enthalpy of steam from a turbine are lower than those produced by a pressure-reducing valve or orifice. Since both systems must supply the same energy input to evaporation, the steam flow to the turbine system must be greater than that to the existing system.

The capital cost data for the Modular Thermal Evaporator/Steam Turbine System is given in Table 30. The installed cost of 487 KW turbine - generator set is \$478/KW. Table 31 lists the economic characteristics of this system indicating a fast payback of less than a year, and an internal rate of return of 60%. It is no wonder that several citrus processors have recently installed steam turbines ahead of their evaporators.

Although the turbine system requires more steam than the existing system, it requires less fuel energy input. Table 32 presents a comparison of the fuel energy inputs for both systems. The steam turbine system requires slightly more fuel input to make steam, but no more fuel input to make electricity, so that the net energy saving is positive. In this case, total fuel savings is 3×10^6 BTU/Hr. which represents a 7.7% reduction in current energy use.

Table 29: System specifications and performance parameters for both existing steam supply to evaporation and for Modular Thermal Evaporator/Steam Turbine (MTE/ST)

	<u>Present System</u>	<u>MTE/ST</u>
Number of evaporators, 6 effect units	3	3
Evaporation rate per unit, lb/hr	40,000	40,000
Boiler steam temperature, °F	366	366
Boiler steam pressure, psia	165	165
Boiler steam quality, %	97	97
Boiler steam enthalpy, Btu/lb	1,170	1,170
Steam energy input to evaporation, 10 ⁶ Btu/hr	28.8	28.8
Steam requirement for evaporation, lb/hr	24,600	26,100
Turbine exhaust pressure, psia	-	20
Turbine exhaust temperature, °F	-	228
Turbine exhaust quality, %	-	94
Isentropic enthalpy drop across turbine, $\frac{\text{Btu}}{\text{lb}}$	-	148
Evaporator steam pressure, psia	14.7	14.7
Evaporator steam temperature, °F	253	212
Evaporator steam quality, %	100	95
Evaporator steam enthalpy, Btu/lb	1,170	1,102
Turbine efficiency, %	-	46
Work output of turbine, kW	-	520
Gearbox efficiency, %	-	95.5
Generator efficiency, %	-	95
Net electrical power output, kW	-	487
Electrical power requirement of three evaporators, kW	450	450
Electrical output to plant, kW	-	37

Table 30: Capital cost breakdown for Modular Thermal
Evaporator/Steam turbine combination

487 kW

Cost Item

Energy conversion unit

Turbine-generator unit with
accessories and controls

Thermal Evaporator Steam
flow monitoring control system

Generator substation & connections

Foundation and installation

Subtotal

Engineering and supervision (10%)

Total cost of installed system

Installed cost per unit capacity

103\$

62

15

40

100

212

21

233.

\$478/kW

Table 31. Economic characteristics of Modular Thermal
Evaporator/steam turbine combination

	<u>Baseline</u>	<u>MTE/ST</u>
Electrical generation capacity, MW	-	0.49
Capital, 10 ⁶ \$		0.233
	Annual Costs	
	<u>10⁶ \$/yr.</u>	<u>10⁶ \$/yr.</u>
Fuel cost		
Cogen. firing		0.488
Natural gas ^a		
Present firing	0.500	
Natural gas ^b		
Electricity credit	-	-0.117
Total energy cost	0.500	0.371
Total energy cost reduction	-	0.129
Additional maintenance cost (@\$0.0069/kWhr)	-	0.021
Payback/IRR		2.6 yrs/38%

^aNatural gas @ current cogeneration rate of \$.35/therm, January, 1981.

^bNatural gas @ standard rate of \$.38/therm, January, 1981.

Table 32: Energy savings for Modular Thermal
Evaporator/Steam Turbine

<u>Parameter</u>	<u>Baseline System</u>	<u>MTE/ST</u>
Steam usage, lb/hr	24,600.	26,100.
Boiler fuel consumption ^a , 10 ⁶ Btu/hr	33.8	35.8
Electrical demand/production, kW	487	487
Fuel to electricity generation ^b , 10 ⁶ Btu/hr	5.0	-
Total fuel energy consumption, 10 ⁶ Btu/hr	38.8	35.8
Energy savings, 10 ⁶ Btu/hr		3.0
Percent energy savings		7.7

^aSteam production @ 75 lb/therm

^bUtility generated electricity delivered @ 33% overall efficiency

5.4.7 Dual-Mode Evaporator

The term "Dual-Mode" refers to a system for water removal in which a heat engine drives a mechanical vapor recompression evaporator and at the same time provides heat for a thermal evaporator. This concept reaps the benefits of mechanical vapor recompression, co-generation, and multiple-effect evaporation. The heat economy for the combined system is 193 BTU/Lb. evaporation compared to 243 BTU/Lb. for a seven-effect evaporator. In neither case does this include juice pumps, or the vacuum producing devices.

5.4.7.1 System Description

There are many possible combinations which could be assembled to create a Dual-Mode evaporator. The compressor may be driven electrically, with a reciprocating engine, with a steam turbine, or with a Brayton engine. The Brayton engine was chosen for detailed study because it seemed to be the most economical overall.

Electricity is generally too expensive to be used as the energy source for the MVR. There are some exceptions to this where hydroelectric power is available.

The reciprocating engine could be used to drive the compressor, but the heat output often falls short of satisfying the needs of the thermal finishing evaporator. This may not always be the case. The reciprocating engine may be a good candidate if the evaporation duty of the thermal evaporator is low.

The steam turbine would generate too little power for the heat available in the exhaust steam. Each application is different, so the Rankine alternate cannot be ruled out where the preconcentration step represents less than half of the total water removal. For orange juice, the preconcentration using MVR would probably represent 70-75% of the total water removal.

The Brayton engine was chosen as having the best overall potential to save energy. Figure 14 shows the general concept of this system. The feed juice is preheated with the MVR condensate so all the energy required in the MVR comes from the compressed vapor.

5.4.7.2 Process Constraints

Orange juice is usually heated to 200°F. for a few seconds to pasteurize it, and to inactivate enzymes which would otherwise cause gellation and "cloud" loss. Single strength juice has about 12% soluble solids (12°Brix). The most common commercial concentrated orange juice is 65% solids (65°Brix). While it is possible to concentrate all the way to 65°Brix using MVR, the heat transfer in the last stages is impeded by the high viscosity and 8°F. of point rise in the 65°Brix juice. For this reason, and in order to take advantage of co-generation, the Dual-Mode concept is presented here.

The MVR removes 72% of the water to be evaporated, while 28% is removed by three-effect thermal evaporator. The latter evaporator is supplied with steam from a boiler heated by the exhaust from the gas turbine driving the MVR unit.

5.4.7.3 Specifications and Performance Parameters

The MVR preconcentrator can be designed to make a product at any intermediate Brix. The example illustrated on the next page uses 30°Brix as the MVR product and 65°Brix as the finished product. Table 33 "System Specifications and Parameters for Dual-Mode Orange Juice Evaporator" lists the pertinent data for a 40,000 lbs./hr. system. The 517 Hp Turbine has an efficiency of 18%. The exhaust gases at 848°F. produce 3,926 lbs./hr. of steam. This is enough steam to operate the three-effect finishing evaporator, and provide some steam to the pasteurizer. A small amount of steam (319 lbs./hr.) would be provided from an independent source to provide the fine tune balancing for temperature and Brix control.

Although not shown in the schematics, the MVR evaporator would consist of several stages of concentration in series in order to go from 12°Brix to 30°Brix. The vapors from all stages are at the same pressure and converge to enter the suction of the compressor.

FIGURE 14: DUAL-MODE JUICE EVAPORATOR

TABLE 33: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
FOR DUAL-MODE ORANGE JUICE EVAPORATOR

MVR EVAPORATOR (Pre-evaporator in Dual-Mode)

Mass Flows	
Feed Rate, Lbs./Hr.	49,060
Product Rate, Lbs./Hr.	19,630
Evaporation, Lbs./Hr.	29,430
Solids in Feed, % by Weight	12
Solids in Product, % by Weight	30
Temperatures, °F.	
Feed	128
Product	141
Low Slide Vapor	140
High Side Saturation	156
Compressor Discharge	237
Pressures psia	
Low Side	2.89
High Side	4.34
Pressure Ratio	1.5
Heat Content, BTU/Lb.	
Feed	89
Product	98
Condensate	124
Low Side Vapors	1,122
High Side Vapors	1,129
Compressor Discharge	1,166
Work and Efficiencies	
Isentropic Work, BTU/Lb.	33
Compressor Work, BTU/Lb.	44
Compressor Efficiency, %	75
Heat Loss, %	0.3

Continued on next page.

TABLE 33: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
FOR DUAL-MODE ORANGE JUICE EVAPORATOR (CONTINUED)

MECHANICAL VAPOR COMPRESSOR

Vapor to Compressor, Lbs./Hr.	29,430
Water to Desuperheat, Lbs./Hr.	990
Steam to Maintain Temperature, Lbs./Hr.	None
Vapor to Tube Nests	30,420
Work Input to Compressor, Hp	509

GAS TURBINE

Gearbox Efficiency, %	98.5
Gas Turbine Size, Hp	517
Turbine Efficiency, %	18
Air/Fuel Ratio	67
Fuel Rate, BTU/Hr. (HHV)	7.3×10^6
Work Heat Rate, BTU/Hr.	1.32×10^6
Exhaust Heat, BTU/Hr.	5.98×10^6
Fuel Mass Rate, Lbs./Hr.	365
Exhaust Temperature, °F.	848

WASTE HEAT BOILER

Steam Pressure, psia	20
Heat to Generate Steam, BTU/Lb.	990
Boiler Efficiency	65
Steam Rate, Lbs./Hr.	3,926

PRE-HEATING - PASTEURIZING AHEAD OF MVR

Condensate/Feed Heat Exchanger	
Juice Rate, Lbs./Hr.	49,060
Condensate Rate, Lbs./Hr.	29,430
Temperature of Juice in, °F.	60
Temperature of Juice out, °F.	116
Temperature of Condensate in, °F.	156
Temperature of Condensate out, °F.	70
Regenerator	
Temperature of Juice in, °F.	116
Temperature of Juice out, °F.	188
Temperature of Pasteurized Juice in, °F.	200
Temperature of Pasteurized Juice out, °F.	128
Pasteurizer	
Temperature of Juice in, °F.	188
Temperature of Juice out, °F.	200
Steam Used, Lbs./Hr.	595

Continued on next page.

**TABLE 33: SYSTEM SPECIFICATIONS AND PERFORMANCE PARAMETERS
FOR DUAL-MODE ORANGE JUICE EVAPORATOR (CONTINUED)**

THERMAL MULTIPLE EFFECT EVAPORATOR

Mass Flows

Feed Rate, Lbs./Hr.	19,630
Product Rate, Lbs./Hr.	9,060
Evaporation, Lbs./Hr.	10,570
Solids in Feed	30
Solids in Product	65
Steam to First-Effect & Second Preheater, Lbs./Hr.	3,650

Temperatures, °F.

Feed Temperature	141
Exit First Preheater	150
Exit Second Preheater	180
Exit First-Effect	160
Exit Second-Effect	140
Exit Third-Effect (Product)	110
Condensate Temperature	140

Table 34. Capital Cost Breakdown for
Modular Dual-Mode Evaporation

40,000 Lbs./Hr.

<u>Cost Item</u>	<u>\$ x 10³</u>
Energy conversion unit	235
Gas turbine engine (500 KW) with accessories and controls	200
Heat recovery boiler	35
MVR evaporator unit	
Compressor	170
Evaporator	350
Three-effect thermal finisher (10,000 Lbs./Hr. evaporation and cooling tower)	300
Foundation, installation & ancillary components	200
Monitoring & control system	20
Subtotal	1,255
Engineering & Supervision (10%)	125
Total cost, installed	\$1,400

Table 35. Economic Characteristics of Dual-Mode
Evaporator/Brayton Engine (40,000 Lbs./
Hr. Evaporation Capacity)

<u>Cost Item</u>	<u>Conventional (Six-effect)</u>	<u>Dual-Mode (MVR/Thermal)</u>
Capital cost (\$ x 10 ³)	684	1,400
Fuel cost	10 ³ x \$/Yr.	10 ³ x \$/Yr.
Boiler firing @ \$.38/therm	250	
Co-gen. firing @ \$.35/therm		169
Purchased electricity cost @ \$.06/KWHR	54	0
Total energy cost	304	169
Additional maintenance cost @ \$.0033/KWHR	0	10
Net annual cost reduction		125
Payback/IRR		10 yrs./7%

5.4.8 Heat Pump for Pasteurization

Hot water from the condensed vapors from juice and peel dehydrating (juice evaporator and waste heat evaporator), is a candidate for heat pump energy recovery. There is about 180 gallons of 140°F. condensate results from each ton of fruit processed. Normally this energy is wasted.

Most plants have need for heat to pasteurize juice. The 140°F. condensate can be used to preheat the juice by means of a heat exchanger. If the juice is to be packaged hot, where regeneration is not practical, preheating to 130°F. would save $100 (130 - 70) + (190 - 70) = 50\%$ of the steam requirements.

However, heat recovery merely by heat exchange is not within the scope of this project. The heat pump method is the system which is within the regime of the project.

The proposal is to lift the heat from the temperature of the condensate to a temperature high enough to do the pasteurization of the juice. A closed heat pump was chosen. It was a pump readily available on the market. Table 36 shows the thermodynamic data for the heat pump system. Table 37 gives the overall energy savings as 25%. However, this would be costly electrical energy. The dollar "savings" is negative even without absorbing equipment depreciation and interest on investment. The heat exchanger saves 50% of the energy with relatively low operating and capital costs, therefore it would be the logical choice.

**Table 36: Specifications and performance parameters
for Pasteurization via Heat Pump System**

Plant fruit load, tons/yr	500,000
Plant capacity, hr/yr	6,000
Average hourly fruit load, 10^3 lb/hr	170
Condensate per lb of fruit	0.7
Condensate available, 10^3 lb/hr	100
Average condensate temperature, °F	130
Juice to be pasteurized, 10^3 lb/hr	15.3
Pasteurization temperature, °F	200
Specific heat of juice, Btu/lb°F	0.9
Temperature of incoming juice, °F	50
Pasteurization heat load, 10^6 Btu/hr	2.0
Heat supplied per heat pump, 10^3 Btu/hr	333
Number of heat pumps	6
Coefficient of performance	3.0
Power consumption per heat pump, kW(e)	33
Temperature of condensate leaving heat pump, °F	113
Total power consumption of heat pump system, kW(e)	198
Total heat output to pasteurization, 10^6 Btu/hr	2.0

Table 37: Energy savings and economic characteristics of
juice pasteurization via Heat Pump

<u>Parameter</u>	<u>Baseline System</u>	<u>Heat Pump System</u>
Pasteurization heat requirement, 10^6 Btu/hr	2.0	2.0
Steam demand ^a , lb/hr	1,938	-
Fuel energy required to supply steam, 10^6 Btu/hr	2.7	-
Electrical demand, kW	-	198.
Fuel energy to electrical power production ^b , 10^6 Btu/hr	-	2.0
Total fuel energy input, 10^6 Btu/hr	2.7	2.0
Energy savings, 10^6 Btu/hr	-	0.7
Percent energy savings		25.
Fuel cost @\$3.80/ 10^6 Btu, \$/hr	10.3	-
Electrical cost @\$0.06/kW-hr, \$/hr	-	11.9
Dollar savings, \$/hr	-	-1.62

A P P E N D I X

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Average Solar Turbine Emissions Compared to Average Emission for Large Bore Piston Engines (Units - Grams Per Horsepower Hour)

	NO _x	HC	CO
Diesel Engines	11	0.3	2
Natural Gas Engines	15	1.5	1.2
Dual Fuel Engines	8	2.0	2.5
Solar Turbines	2.0	0.18	1.2

COMPARISON OF PISTON ENGINE AND GAS TURBINE EMISSIONS

In general, large bore IC piston engines have much higher exhaust emissions when compared to gas turbines. This condition results from:

- Internal combustion piston engines (including gasoline, diesel, and gas) operate best at very high pressures and at near stoichiometric conditions. (For hydrocarbon fuel this is an air/fuel ratio of about 15 to 1 by weight).

This operating condition also results in the highest combustion temperature. The high combustion temperature encourages the chemical combustion of nitrogen to form NO_x.

- Internal combustion piston engines burn under intermittent explosive conditions resulting in less efficient flame propagation and incomplete combustion. The net results are that carbon monoxide and hydrocarbon (unburned fuel) emissions are also relatively high.
- The relatively cold cylinder walls and head of the piston engine combustion chamber tend to quench the flame encouraging combustion resulting in smoke and hydrocarbons (unburned fuel).

By contrast, the gas turbine:

- Operates at much lower combustion temperatures and pressures which are conducive to lower NO_x formation.
- Operates continuously at high air fuel ratios (lean mixtures at about 50 to 1). Higher combustion efficiencies are achieved with excess air firing, allowing a more complete combustion, which results in relatively low carbon monoxide and hydrocarbon emissions.