Teaching power cycles by comparative first- and second-law analysis of their evolution

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The teaching of power cycles in courses of thermodynamics or thermal engineering was traditionally based on first-law analysis. Second-law analysis was typically taught later, and not integrated with it. This approach leaves the student ignorant of the effect of operating parameters and cycle modifications on the accompanying exergy (availability) magnitudes and component irreversibilities, which are necessary for evaluating the potential for further system improvements. It also leaves many of the students with an ambiguous understanding of the exergy concept and its use. Consonant with the gradual changes in this educational approach, which increasingly attempt to integrate first- and second-law analysis, this paper recommends a strategy which integrates exergy analysis into the introduction and teaching of energy systems, demonstrated and made didactically appealing by an examination of the historical evolution of power plants, emphasizing the objectives for improvements, accomplishments, constraints, and consequently the remaining opportunities. Important conclusions from exergy analysis, not obtainable from the conventional energy analysis, were emphasized. It was found that this approach evoked the intellectual curiosity of students and increased their interest in the course.

NOMENCLATURE

a_{CH}	specific chemical exergy, kJ/kg mol
a_f	specific flow exergy, kJ/kg mol
a_{TM}	specific thermal mechanical exergy, kJ/kg mol
$\dot{A}$	exergy rate, kJ/s
$\dot{A}_d$	exergy destruction rate, kJ/s
c_p	molar specific heat, kJ/kg mol-K

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$\dot{E}$ convective energy rate, kJ/s
h specific enthalpy, kJ/kg mol
$\dot{m}$ matter flow rate, kg/s
$\dot{N}$ molar flow rate, kg mol/s
N_m sum of total number of moles of mixture at reactor exit
$\dot{N}_p$ molar production rate, kg mol/s
P pressure, kPa
P_o atmospheric pressure, kPa
$\dot{Q}$ heat transfer rate, kJ/s
R universal gas constant, kJ/kg mol-K
s specific entropy, kJ/kg mol-K
T temperature, K
y stoichiometric coefficient of combustion product, kg mol
$\dot{W}$ work rate, kJ/s
z stoichiometric coefficient of combustion reactant, kg mol

η efficiency
μ electrochemical potential, kJ/kg mol
χ mole fraction

Subscripts

i mass stream index
j species index
0 reference ('dead state') conditions

Superscripts

$\dot{o}$ rate (per unit time)

INTRODUCTION

The teaching of power cycles in courses of thermodynamics or thermal engineering was, and in large part still is, typically based on first-law analysis, starting with the simplest possible cycle configuration and progressing through a number of modifications of the basic cycle which have been implemented over time [1–5]. These modifications, each made at some additional cost of equipment and complexity, are typically evaluated for their contribution to performance criteria such as first-law efficiency, power output, or system reliability and life. For example, in teaching Rankine cycles there might be a sequence of (1) a plant consisting of a pump, boiler and turbine exhausting into the ambient, (2) the addition of a condenser to close the loop, (3) the addition of a superheater, (4) the addition of reheat, (5) the addition of regenerative feedwater heating, (6) the addition of air-preheaters from stack gas, and (7) possible combinations of steps (3)–(7) above. While this gradual approach is didactically clear and introduces the students well to how plants operate and to their energy analysis, the students remain ignorant of the effect of operating parameters and cycle modifications on the accompanying exergy magnitudes and irreversibilities of subsystems which are the direct indicator of the ability to produce useful work.

Second-law (exergy) analysis, which leads to the evaluation of exergy states, and, consequently of process and system irreversibilities is being gradually introduced into thermodynamics and energy engineering curriculum but not always effectively. Most often this is done in a separate chapter, not integrated with the teaching of energy conversion systems. This separation deprives the students of the realization of the vital importance of exergy analysis in the process of power plant design and evaluation, demonstrated, for example, in the powerful insights which it produces. The separation also makes it easier to regard exergy analysis as a topic for specialists (or advanced students) only, one of the chapters in the textbook which may remain inadequately studied (if at all) in the time crunch of the term. In the learning of contemporary engineering thermodynamics, the concepts of exergy and of its use often also remain somewhat ambiguous. There are several ingrained causes for this ambiguity. People have a much better operational (see reference [6]) understanding of energy conservation than of energy quality or of reversibility or irreversibility of processes, let alone of the definition of exergy (availability) and entropy [7]. Almost anecdotally, the magnitude of the first-law (energy) and second-law (exergy) efficiencies of thermal power plants are almost the same, coincidentally because the fuel energy and exergy (based on typical ambient dead state conditions) are almost of the same magnitude, creating a possible (but false) disincentive for conducting the more difficult second-law analysis.

In fact, the first- and second-law efficiencies may differ markedly for each of the power plant components. For example, neglect of second-law analysis produces the potentially misleading first-law analysis conclusion that Rankine cycle efficiency may be improved by tending to the major heat energy loss in the condenser (of the order of 50% of the fuel energy input), while clearly even the most efficient use of this large amount of low-temperature energy could improve the power production efficiency by a few percent only. The second-law analysis points correctly to the boiler, and in it to the combustion process irreversibility, as the most inefficient component (the boiler, ironically, is highly efficient from the first-law standpoint in contemporary units), and focuses R&D needs correctly on that component [8, 9].

Our concern about this educational deficiency, presented in an earlier paper [10] was shared by many in the thermodynamics education community. The new educational strategy proposed is intended to remedy both of the abovementioned problems: it integrates second-law analysis into the gradual introduction of the power plants, and it motivates quite clearly the need for second-law analysis. This approach has indeed been seen to gradually enter several recent engineering thermodynamics textbooks [11, 12], with more extensive integration in [13–15].

Out of many possible variants of this strategy, which educators could adapt, develop, and experiment with, we show in this paper one approach which thermodynamically examines the historical evolution of power plants, emphasizing the objectives for improvements, accomplishments, constraints, and consequently the remaining opportunities. We have found that this approach was also didactically appealing to students.

To assist those who are interested in adopting this approach in their teaching, we have presented the material below in more detail than would normally be provided in a research paper.

POWER PLANT EVOLUTION: A BRIEF HISTORICAL NOTE

A brief historical review would be useful in setting the stage for the subsequent analysis.

The generation of steam for power dates back to the initial operation of steam engines, originally in demand for pumping water from mines in England. The first commercially successful steam engine was patented by Thomas Savery in 1698 [16]. Throughout the following two centuries, a number of improvements to the steam engine were implemented, directed at fuel economy (efficiency improvement) and safety. It is during that time, for example, that Carnot [17] developed his seminal work on cycle efficiency and the second law of thermodynamics.

In the latter part of the eighteenth century and throughout the nineteenth century, steam was used to provide heat and power for local industrial use. With the advent of practical electric power generation and distribution, electric utility companies were formed to served industrial, commercial, and residential users.

The first centralized electrical power generating station in America was built in the year 1881 by the Brush Electric Light Company in Philadelphia [16]. This plant (as well as others subsequently designed before the year 1920) employed reciprocating steam engines to drive electric generators [5].

In the year 1903, Commonwealth Edison (CE) became the first utility to use steam turbines exclusively for electric power generation by starting operation of the Fisk Street Station in Chicago. The steam turbine inlet conditions of this power plant were basically saturated vapour at a pressure of 1.2 MPa (170 psia [16]).

Since that time (the turn of the century), power plant modifications continued and, correspondingly, plant efficiency increased primarily as a result of increases in operating steam temperature and pressure. Modern-day boilers are designed to withstand pressures as high as 1500–4000 psia (17.2–27.6 MPa) and temperatures as high as 1100°F (close to 600°C). To quantify these improvements of plant efficiency, the relevant conditions, configurations, and limitations of each subsystem of the central station for the relevant periods of time are presented in the next section.

For both technical and educational reasons it is important to note that (1) exergy and energy losses are incurred in fuel extraction, preparation and transportation, prior to combustion in the boiler, (2) power plants are implementing, particularly in the last two decades, an increasing degree of environmental impact control, reflected in increased hardware, energy (and exergy, or fuel), costs, and (3) such losses and costs are also incurred in the transmission of the produced power to the points of final use. While not considered in the analysis below, it would be worthwhile to include these components in future analyses, so that the students learn the quantitative energy and exergy implications of fuel-related operations, environmental protection, and power transmission.

THE ANALYSIS

To study this progress in power plant performance, two fuel-fired Rankine cycle power plant configurations which exemplify designs (i) of the 1910–1920 era and (ii) those of today will be analysed.

The simple Rankine cycle plant (1910–1920 era)

By the latter part of the nineteenth century, economizers, evaporators, superheaters and air preheaters were all commonly employed components of the steam generator [18]. By 1900 the Westinghouse Electric Company was manufacturing multistage steam turbines of the Parsons type and the General Electric Company was developing an impulse turbine of the DeLaval type with Curtis velocity staging [5]. Pump technology was, of course, already in

common use by that time, dating back to the early years of coal mining (almost 300 years ago [16, 19]). Watt had developed the condenser in the latter part of the eighteenth century [20].

Plants constructed in the 1910–1920 era were of the simple Rankine cycle type with little or no gas clean-up equipment. Fig. 1 displays a schematic diagram of such plants, with the typical operating conditions (modelled after the Crawford Avenue Station constructed by CE in the 1920s). As shown, these electrical power generating stations incorporated: (i) a steam generator (i.e., a combustion chamber and a series of heat exchangers: economizer, evaporator, superheater, and at times, an air preheater) and auxiliary equipment, (ii) a single-stage turbine, (iii) a condenser, and (iv) a single-stage pump. The turbine isentropic efficiency is assumed to be 80%, and that of the single-stage pump 60%.

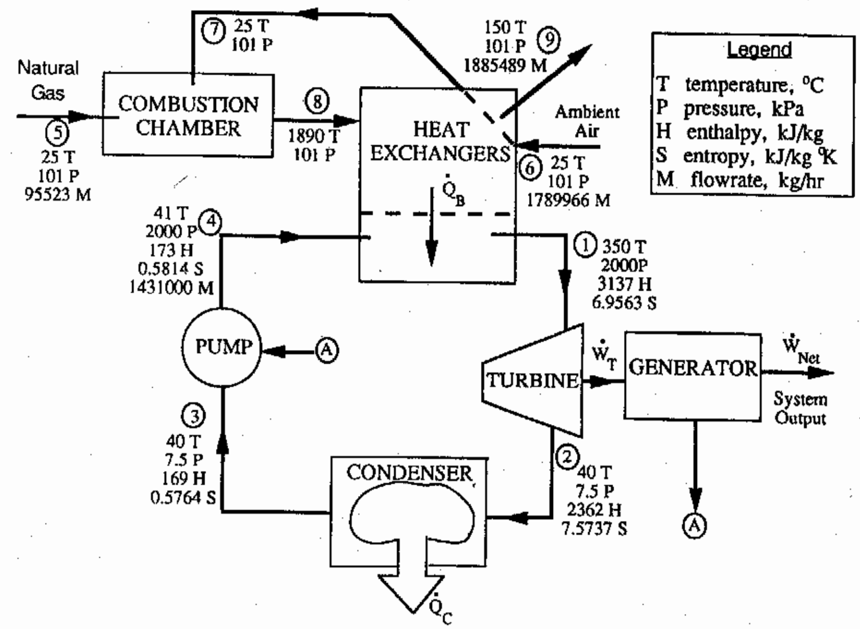

Fig. 1. Thermodynamic property data for the simple Rankine cycle power plant (1910–1920 era, without combustion air preheat).

The boiler pressure in such plants was in the neighbourhood of 2 MPa (290 psia), with superheated steam exiting at a temperature of about 350°C (660°F). The steam is expanded in the turbine to a condenser pressure of about 7.5 kPa (1.09 psia). The turbine isentropic efficiency was about 80%. It was assumed in the analysis that condensation occurs isobarically, with the H_2O exiting the condenser as a saturated liquid. The pump discharge pressure is 2 MPa.

Although coal was then and is now the principal fuel used in power stations, it was assumed in the analysis below that the fuel was natural gas, because it is simpler to consider in a course of instruction (the chemical reactions are simpler, and ideal gas relations may be used). Its composition was assumed here to be 83.4% methane, 15.8% ethane and 0.8%

nitrogen by volume (representative of natural gas extracted from fields in Pennsylvania [16]). It was also assumed in this paper that an amount of 10% excess air is employed to assure complete oxidation of the fuel, and that the stack gas temperature is set at 150°C (302°F) to prevent condensation in the stack and consequent corrosion.

The modern subcritical 2600 psig Rankine cycle power plant

During the 1920s, both regenerative feedwater heaters and reheaters were introduced as a means to improve plant efficiency [5]. Modern fossil-fuel power plants employ one or two stages of reheat and between five and eight feedwater heating stages [21].

Boiler and system technology continued to improve rapidly: by the year 1930, common turbine inlet steam conditions were 385°C (725°F) and 3.8 MPa (nearly 550 psia [16]). Modern-day boilers are designed to withstand conditions of up to about 28 MPa (4000 psia) and 565°C (1050°F). Utility steam generators are essentially of two basic types: (i) the subcritical water-tube drum type, and (ii) the supercritical once-through type. The supercritical units usually operate around 24 MPa (3500 psia); the subcritical units usually operate at either 13.2 MPa (1900 psig) or at 18 MPa (2600 psig). The majority of utility steam generators purchased in the 1970s and 1980s are of the 2600 psig variety, producing superheated steam at temperatures of about 540°C (1000°F) with one or two stages of reheat [21].

Fig. 2 displays the schematic diagram and operating conditions of the plant which is studied in this analysis. As shown, the steam power cycle has one stage of reheat (typical of modern-day plants) and one stage of feedwater heating (as mentioned above, modern-day plants typically have five to eight feedwater heating stages, and the effects of additional feedwater heating stages on second-law efficiency will be discussed in the Results section).

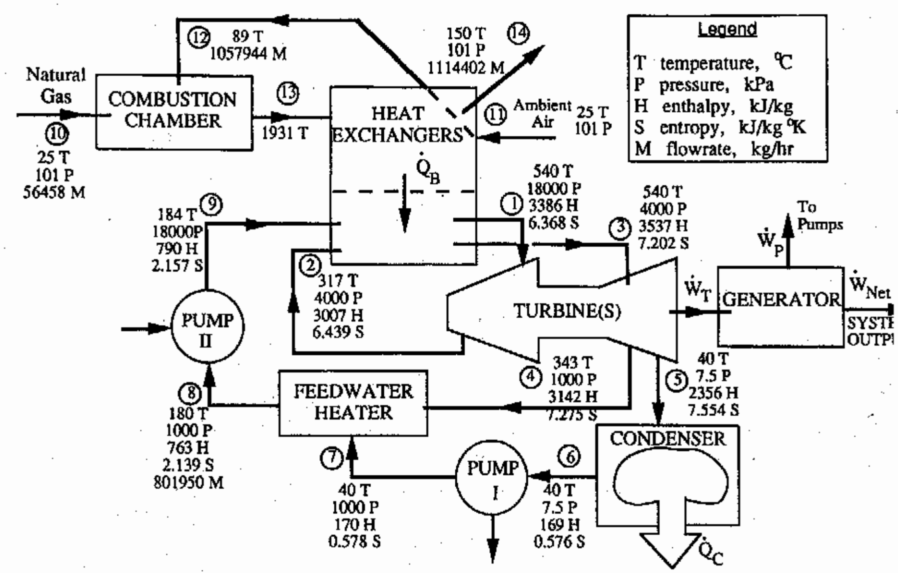

Fig. 2. Thermodynamic property data for the subcritical (2600 psig) Rankine cycle power plant (1980–1990 era, with one feedwater heating stage).

Again based on data gathered from the literature [5, 16, 19, 21], the operating conditions representative for plants constructed in the 1980s and used in this study are as follows. The boiler pressure is 18 MPa. The steam leaves the boiler superheated at 540°C. Process 1–2 in Fig. 2 is a steam expansion in the first stage turbine to the reheat pressure of 4 MPa. The steam is reheated in process 2–3 at constant pressure to a temperature of 540°C. The steam then expands in the second stage turbine to the regenerative feedwater heating pressure of 1 MPa.

At this point, approximately 20% of the steam (an amount determined by an energy balance on the feedwater heater) is extracted and delivered to the feedwater heater. The remainder of the steam is expanded in the third stage turbine to the condensation pressure of 7.5 kPa. The isentropic efficiency of all turbine stages is assumed to equal 90% [21].

Process 5–6 is a constant-pressure condensation where the exit state is saturated liquid. Process 6–7 is a pumping process where the pump exit state is at the feedwater heater pressure of 1 MPa. The feedwater is of the open-type, whereby the H_2O streams 4 and 7 mix, exiting the feedwater heater as saturated liquid at 1 MPa. Finally, process 8–9 is a water pumping process where the exit pressure is 18 MPa, implying a constant-pressure transfer of heat in the boiler heat exchangers. The pump efficiency is 70%.

THE MATHEMATICAL MODELLING

The system of equations which comprise the mathematical model consists of (1) balances for energy, matter, and exergy, (2) thermochemical property relations, (3) performance characteristics of the energy conversion units, and (4) boundary conditions.

Rather than going through a detailed description of the calculation method, which is well-known, only the principles are briefly described below.

The balances

For each subsystem of the power plant, balances for energy, exergy (and therefore, implicitly for entropy) and for each chemical species are satisfied.

The first law for a steady open system analysis requires the energy balance (positive Q for a heat input, positive W for a work output).

$$\sum_{\text{input}} \dot{E}_i + \dot{Q} = \sum_{\text{output}} \dot{E}_i + \dot{W} \tag{1}$$

The balance of matter for each chemical species j (for each subsystem) is:

$$\sum_{\text{output}} \dot{N}_j = \sum_{\text{input}} \dot{N}_j + \dot{N}_{P_j} \tag{2}$$

where $\dot{N}_j$ is the flow rate of species j in or out of the control volume, and $\dot{N}_{P_j}$ is the net production rate of species j within the system due to chemical reactions (where the production term is negative if the species is a reactant, positive if the species is a product of reaction).

Denoting the amount of exergy brought in or out of the system by the flow of stream i as $\dot{A}_i$, and the rate of exergy destruction as $\dot{A}_d$, the exergy balance (the students should note that there is no law of exergy conservation, the equation is balanced by the destruction term) is (see [22]):

$$\sum_{\text{input}} A_i + \int \left(1 - \frac{T_o}{T}\right) d\dot{Q} = \sum_{\text{output}} A_i + \dot{W} + A_d \tag{3}$$

expressing the fact that the sum of the exergy associated with the entering matter, and of the exergy of the net heat addition, is equal to the sum of the exergy of the exiting matter, of the useful work output, and of the irreversible destruction of exergy associated with all real processes. T here is the temperature at the boundary of the system where the heat interaction occurs.

Equations (1) and (2) may be regarded as the governing balances of the mathematical model, since they are used in determining the thermodynamics states of the cycle fluids, and consequently energy efficiencies. Equation (3) is the basis for the second-law analysis. It is employed for analysing irreversibilities and exergy efficiencies in the component processes and devices, as well as in the entire system which depends on the specific choice of the configuration of these components.

Thermochemical property relations

For the **gas side** of the plant, the energy transport due to flow of matter through a control volume (equation (1)) was expressed by the enthalpies of all chemical species j in each gas stream i using (see [23]):

$$\dot{E}_i = \sum_j \dot{N}_{ij} h_j \tag{4}$$

Ideal gas behaviour was assumed for the fuel, air, and product gas streams. This assumption is deemed justifiable at atmospheric pressure, especially for the product gas at the experienced elevated temperatures. Moreover, this assumption is made consistently in both cases treated and is adequate for the purpose of comparing different plant configurations. It may be didactically valuable to perform one set of calculations using real gas data and compare the results to those obtained with the ideal gas assumption: the differences would be small. Consequently

$$h_j = h_{j_0} + \int_{T_0}^{T} c_{p_j}\, dT \tag{5}$$

The exergy value of flow stream I is:

$$\dot{A}_i = \dot{N}_i a_{f_i} \tag{6}$$

The specific exergy of stream i is given by:

$$a_{f_i} = \sum_j \chi_{ij} a_{f_{ij}} \tag{7}$$

where the summation takes place over all species j present in stream i.

In turn, the specific flow exergy a_f can be expressed as composed of two available energy contributions: (i) the specific thermomechanical exergy a_{TM} and (ii) the specific chemical exergy a_{CH} [24]:

$$a_{f_{ij}} = a_{TM_{ij}} + a_{CH_{ij}} \tag{8}$$

where

$$a_{TM_{ij}} = \int_{T_o}^{T} c_{p_{ij}} \left(1 - \frac{T_o}{T}\right) dT + RT_o \ln \frac{P}{P_o} \tag{9}$$

and

$$a_{CH_{ij}} = h_{ij}(T_o) - T_o s_{ij}(T_o, P_o) + RT_o \ln \chi_{ij} - \mu_{ij}^{o} \tag{10}$$

Appropriate gas tables and correlations for enthalpies of formation, absolute entropies, chemical exergies, and ideal gas heat capacity coefficients were consulted [1, 3, 24].

For the **steam side** of the plant, steam tables were consulted for evaluating the enthalpies and entropies in equation (1) and (3), in which

$$\dot{E}_i = \dot{m}_i h_i \tag{11}$$

and

$$A_i = \dot{m}_i \left[(h_i - h_o) - T_o(s_i - s_o)\right] \tag{12}$$

where i represents the station number, specifying the location in the power cycle.

Performance characteristics and boundary conditions

The performance characteristics and boundary conditions (in addition to those already mentioned in the discussion of the two analysed power plants) are the following: (1) the reference state of the atmosphere is taken to be defined by $T_o = 298.15$ K (25°C, 77°F) and $P_o = 1$ atm. The composition of the reference atmosphere is shown in Table 1. As shown in Table 1, the stable configuration for carbon, oxygen, and nitrogen, is taken to be that of CO_2, O_2, and N_2, respectively, as they exist in air saturated with liquid water at (T_o, P_o). Hydrogen is assumed stable in the liquid phase of water saturated with air at (T_o, P_o). It would be didactically useful to introduce the students at this point (or earlier) to the dead state concept, at least in an operational manner, (2) the inlet fuel and air temperatures are equal to the ambient temperature of 25°C, (3) the stack temperature is fixed at 150°C (to assure that the H_2O in the stack gas will not condense), (4) all pressures on the combustion gas side of the plant are 1 atm, (5) all components except the steam generator have adiabatic boundaries, (6) the heat loss value from the steam generator to the environment is equal to 3% of the higher heating value (HHV) of the fuel [16], (7) the reference environmental water (lake, river or pond) temperature is 20°C, and (8) the generator electromechanical efficiency is assumed to be 98%.

Table 1. The composition of the reference atmosphere

Substance	Mole fraction
Argon, Ar	0.0091
Carbon dioxide, CO_2	0.0003
Water vapour, H_2O	0.0312
Oxygen, O_2	0.2034
Nitrogen, N_2	0.7560

PLANT SUBSYSTEM ANALYSIS

Boiler unit analysis

The boiler unit of a modern power plant consists of a combustion chamber and a series of heat exchangers (e.g., economizer, evaporator, superheater, air preheater). To be able to calculate separately the irreversibilities associated with combustion and with heat transfer, these various devices within the boiler are hypothetically separated in this study, as detailed below. This is done by first calculating the exergy destruction due to combustion with the assumption that no heat transfer occurs at the same time (i.e., adiabatic combustion). Using an approach common in the literature (see, for example, reference [8]), the combustion and heat transfer processes are studied here as though they occur in series, rather than the real situation of combustion progressing in parallel with heat transfer. Performing, in addition, the global second-law analysis on the entire boiler system gives the combined combustion/heat transfer exergy losses. These computations then also allow the determination of the extent of exergy destruction increase above that incurred by combustion without heat transfer.

An adiabatic combustion analysis is thus first performed on both a first- and second-law basis. The heat transfer processes from the product gases to the steam in the power cycle and to the air in the air preheater are subsequently analysed. This approach will therefore provide a breakdown of the irreversibilities within the boiler.

Combustion chamber analysis

The combustion reaction of natural gas with 10% excess air is basically

$$z_1CH_4 + z_2C_2H_6 + z_3N_2 + z_4O_2 + z_5H_2O \rightarrow y_1CO_2 + y_2H_2O + y_3N_2 + y_4O_2 \tag{13}$$

where z_i refer to the reactants and y_i to the reaction products. Complete adiabatic combustion is assumed here, as described above, and the theoretical adiabatic flame temperatures are in the vicinity of 1840–1950°C. This assumption will produce a combustion process exergy efficiency which is somewhat higher than that of the real case because in the real case the temperatures are lower due to heat transfer to the water and steam in the boiler tubes. Since the real combustion reaction thus occurs at lower flame temperatures, it is less efficient.

Assuming adiabatic boundaries (the steam generator heat loss term is included in the heat exchanger analysis), zero work extraction and negligible changes in the kinetic and potential energy values of the flow streams, the energy equation (1) for the combustion analysis thus reduces to

$$\sum_i (\dot{N}_i h_i)_{\text{products}} = \sum_i (\dot{N}_i h_i)_{\text{reactants}} \tag{14}$$

This energy balance, in conjunction with the reaction equation (13) and the matter balances for all chemical species, equation (2), allows solution for the unknowns, namely, the product gas temperature and composition. From equation (3), the combustion irreversibility is evaluated using the equation

$$A_d = \sum_{\text{input}} A_i - \sum_{\text{output}} A_i \tag{15}$$

Heat exchanger analysis

For the heat exchanger analyses, global thermodynamic evaluations are performed on each unit. Assuming no work is extracted from the system and negligible changes in kinetic and potential energy of each flow stream, the energy balance for each fluid stream reduces to

$$\sum_{\text{input}} (\dot{N}_i h_i) + \dot{Q} = \sum_{\text{output}} (\dot{N}_i h_i) \tag{16}$$

Because all chemical reactions are assumed to have occurred prior to heat exchange, the gas composition is fixed during the heat exchange process. Equation (2) then reduces to a simple balance of matter on all chemical species j without the production term N_p.

The heat exchanger global irreversibility and exergy lost to the environment because of the heat loss through the walls are evaluated by

$$A_d + A_{\text{heat loss}} = \sum_{\text{input}} A_i - \sum_{\text{output}} A_i \tag{17}$$

To assure credible magnitudes of the heat exchange temperature differences and compliance with the second law (i.e., possible erroneous crossover of temperature profiles implying heat transfer in a direction opposite to the one intended), the relevant temperature differences between the hot and cold fluid streams are monitored during the calculations. The relevant temperature–heat transfer (T–Q) diagrams are displayed in Figs 3 and 4.

Turbogenerator analysis

Assuming adiabatic boundaries and negligible changes in kinetic and potential energy of the steam flow streams, the energy equation for the steam turbines is

$$(\dot{m}h)_{\text{input}} = (\dot{m}h)_{\text{output}} + \dot{W}_T \tag{18}$$

For steady flow through the turbine, equation (2) reduces to a simple matter balance on H_2O. The exergy destruction associated with turbine operation is determined from equation (3) as

$$A_d = A_{\text{input}} - A_{\text{output}} - \dot{W}_T \tag{19}$$

Finally, turbine performance is introduced into the analysis through the isentropic efficiency, defined as

$$\eta_{T_s} = \frac{\dot{W}_T}{\dot{W}_{T_s}} \tag{20}$$

where W_T is the actual turbine shaft work, and W_{T_s} the shaft work produce in an isentropic process.

Pump analysis

The pump isentropic efficiency is

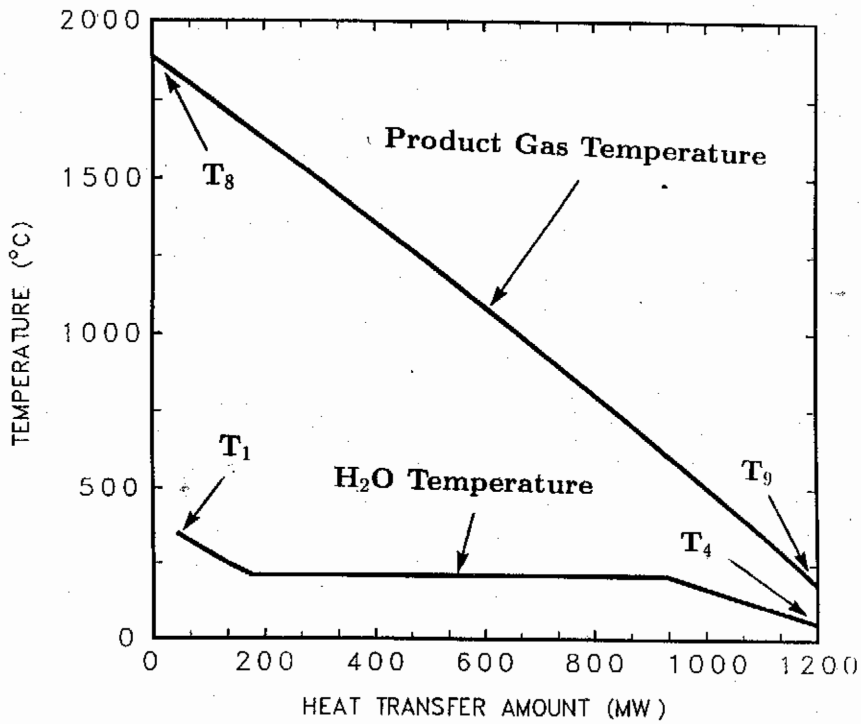

Fig. 3. Temperature–heat (T–Q) diagram for the simple Rankine cycle power plant (1910–1920 era, without combustion air preheat).

$$\eta_{P_s} = \frac{\dot{W}_{P_s}}{\dot{W}_P} \tag{21}$$

where W_P is the actual work required by the pump, and W_{P_s} that required in an isentropic pumping process. The relevant governing equations (energy and matter balances) are identical to those of the turbine analysis except for the work term W_T which is replaced by the pump work term W_P (a negative quantity). The relevant exergy balance is equation (19) with the same work term replacement.

RESULTS: FIRST-LAW ANALYSIS

By employing the energy and matter balances, a first-law analysis was performed on the two previously defined electrical power generating stations. Figs 1 and 2 display the relevant thermodynamic states for the various plant components. The corresponding *energy* 'flow' diagrams are given in Figs 5 and 6. The numbers shown alongside the flow paths between units represent the amount of energy 'flowing' past that station based on a boiler fuel energy input value of 100 units. The energy input with fuel is based on the higher heating value of the fuel (HHV = 989 200 kJ/kmol [16]).

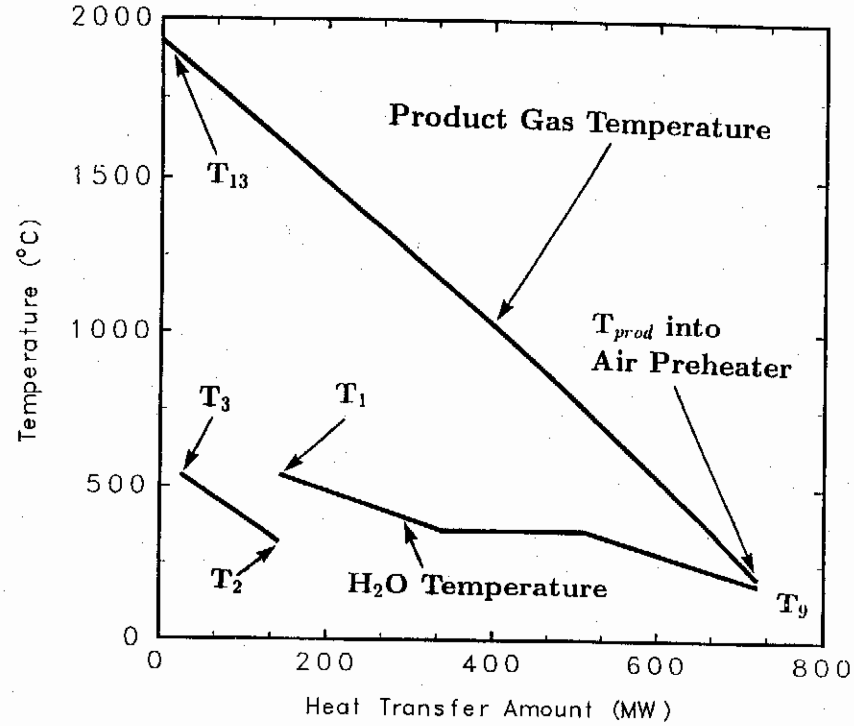

Fig. 4. Temperature–heat (T–Q) diagram for the subcritical (2600 psig) Rankine cycle power plant (1980–1990 era, with one feedwater heating stage).

The overall energy efficiency of the 1910–1920 era electrical power generating station is 21% (Fig. 5). According to this first-law analysis, the primary system losses are the discharge of heat from the condenser and with the stack gases. As seen in Fig. 5, 61% of the fuel energy is expelled in the condenser, and 14.6% with the stack gases. Without the advantage of second-law analysis one could have concluded that the most promising avenue for improving plant efficiency would be by extracting more power from either the condenser steam or the stack gases.

Reviewing the modifications of electrical power station design and operation since the year 1920, one observes, however, that little if any changes have occurred in the thermodynamic states of the condensing steam and the stack gases: modern-day plants do not incorporate any additional capital equipment which utilizes more of this energy expelled in the condenser or up the chimney for the production of electrical power. Since the exit temperature of the stack gas is the same in both old and new plants, any excess energy of the stack gases has been used in the older plants in the economizer section of the boiler, to preheat the feedwater, and in the modern plant it has been used in an air preheater, with no further ability to increase exergy efficiency in either of the two cases.

Rather, the primary modifications to the steam power cycle since the year 1920 were (1) increased turbine inlet temperatures and pressures, (2) the incorporation of reheat, and (3) the incorporation of regenerative feedwater heating. Raising the working fluid (H_2O)

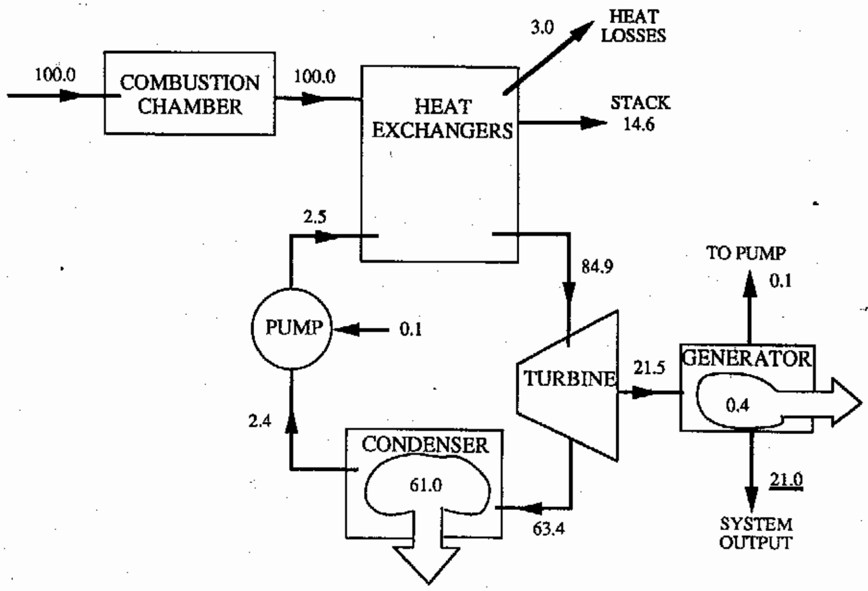

Fig. 5. Energy flow diagram for the simple Rankine cycle power plant (1910–1920 era, without combustion air preheat).

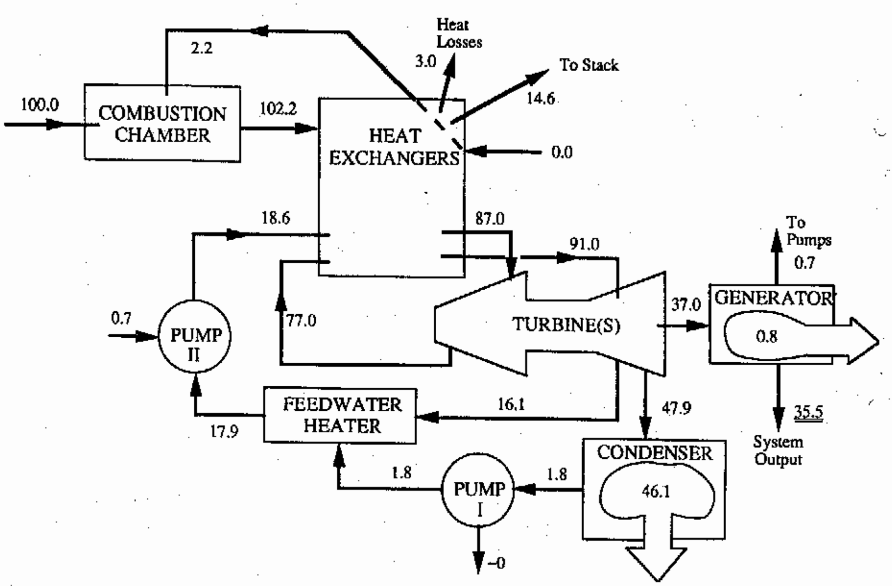

Fig. 6. Energy flow diagram for the subcritical (2600 psig) Rankine cycle power plant (1980–1990 era, with one feedwater heating stage).

temperature closer to the combustion temperature, all of these changes reduce inefficiencies associated with heat transfer from the combustion product gases to the steam in the cycle. The results of the energy analysis shown in Fig. 5 did not indicate that this was an area in which to improve system efficiency.

The first-law analysis of the 'modern' power plant, depicted in Figs 2 and 6, discloses that by (1) increasing the turbine inlet steam pressure and temperature to values employed in modern-day stations, (2) incorporating one stage of reheat, and (3) employing one stage of regenerative feedwater heating, overall plant energy efficiency has increased to 35.5%, which is an absolute gain of 14.5 percentage points (i.e., a relative gain of 69%). By comparing Figs 5 and 6, first-law analysis leads one to believe that these gains in plant efficiency over the last 70–80 years were accomplished by expelling less heat in the condenser, although basically no changes were made in the condensation process. Furthermore, these first-law analyses do not reveal that any improvement in boiler heat exchange performance has been accomplished.

The extent to which these first-law analyses are misleading in that respect can be demonstrated by an example of calculation of the maximum possible work production from using the condenser steam as the high-temperature heat source for a Carnot cycle engine. The saturation temperature corresponding to the condensation pressure of 7.5 kPa is 40.3°C (313.4 K). The reference environmental water temperature is 20°C (as described in the boundary conditions). The maximal efficiency of a hypothetical Carnot engine operating between these two temperatures is 6.5%. Considering that 61% of the fuel energy is expelled in the condenser (Fig. 5) and thus hypothetically available as heat input for that bottoming Carnot cycle, it follows that the maximum possible work production from such a bottoming cycle is only 4% of the fuel energy. This amount of useful work should coincide with the amount of exergy destroyed in the condensation process and this is verified in the next section.

This clearly demonstrates the inadequacy of first-law analysis in attempting to pinpoint prospective areas for improving the efficiency of electrical power production. The first-law analysis is necessary for the modelling of processes in that it helps determine system states and flow rates. Second-law analysis, as pointed out by Keenan [25] and demonstrated below, serves to pinpoint the work-production inefficiencies of energy conversion systems.

RESULTS: SECOND-LAW ANALYSIS

Exergy analyses were performed on these same two plants. The relevant results are summarized in Figs 7 and 8, where the numbers alongside the flow streams at the entrance/exit of each unit represent the amount of exergy 'flow' past that station, as a percentage of the fuel exergy. The negative numbers located within the units themselves represent the amount of exergy destruction due to that particular process, again based on a fuel exergy value of 100 units.

As shown in Fig. 7(a), the overall second-law efficiency of the 1910–1920 era power plant is only 22.4%. The difference between the overall plant first- and second-law efficiencies (21% in Fig. 5 and 22.4% in Fig. 7(a), respectively) is a consequence solely of the difference between the exergy and the HHV of the fuel. This difference, however, is not the focus of this discussion. The discussion is aimed at the significant differences of component inefficiencies as determined by the first- and second-law analyses.

In the 1910–1920 era power plants, the largest irreversibility occurs in the heat transfer processes within the boiler wherein 34.1% of the fuel exergy is destroyed. This is due to the

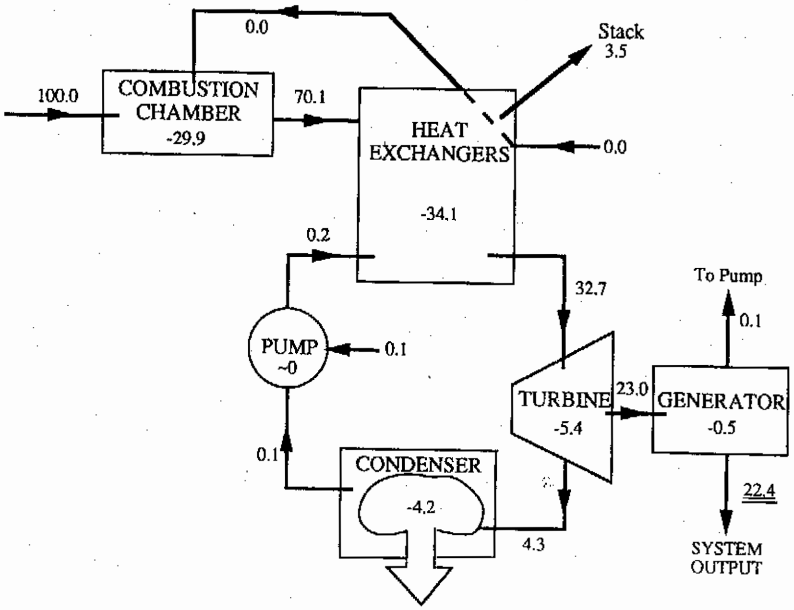

Fig. 7(a). Exergy flow diagram for the simple Rankine cycle power plant (1910–1920 era, without combustion air preheat).

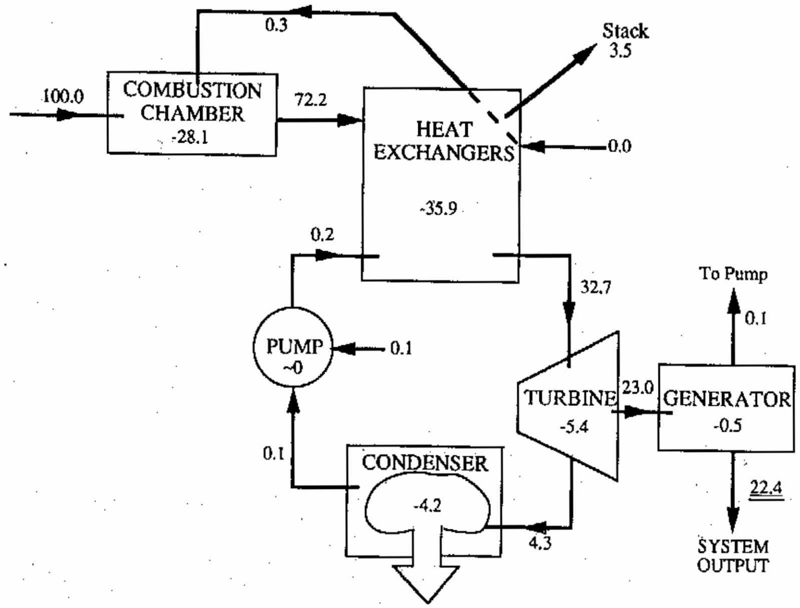

Fig. 7(b). Exergy flow diagram for the simple Rankine cycle power plant (1910–1920 era, with combustion air preheat to 89°C).

exceedingly large temperature differences which were experienced in the steam generator heat exchangers in those days (see Fig. 3). The second most inefficient process is combustion, destroying 29.9% of the incoming fuel exergy. Thus, in the power plants constructed in the early 1900s, the combustion and heat transfer processes were responsible for destroying 64% of the fuel exergy (these two processes were therefore responsible for about 82% of the total system losses). The results of second-law analysis back in the 1910–1920 time period would have thus revealed that the foremost modifications to be made should be those which will reduce heat exchange irreversibility. As shown below, these indeed are the changes that were implemented by power plant technologists over the years since that time, which resulted in improved overall efficiency.

Fig. 7(a) is the exergy flow diagram for the 1910–1920 power plant with no air preheat. Air preheat was unnecessary in this plant because the boiler feedwater temperature was only 41°C (see Fig. 1). For comparison to the results of the exergy study of the 'modern-day' power station, consider the air to be preheated (air preheater technology was available in those days) to a temperature of 89°C (192°F; the temperature to which air is preheated in the analysis of the 'modern-day' power plant). As shown in Fig. 7(b), the air preheat to this temperature causes a 1.8% (of the input fuel exergy) increase in the heat exchange irreversibility, and the same reduction in combustion exergy destruction. Air preheating thus provides no net gain in the efficiency of this plant, but it allows the incorporation of regenerative feedwater heating which, as shown in the analysis of the modern-day power plant, does increase the efficiency by reducing the overall heat transfer irreversibility in the steam generator.

The results of the second-law analysis of the 'modern-day' power station are shown in Fig. 8. By modifying the turbine inlet steam conditions to those representative of today and incorporating single stages of reheat and regeneration, the overall plant efficiency increased to a value of 37.9%, which is an absolute gain of 15.5 percentage points (a relative gain of 69%). Owing to these modifications, the 'targeted' heat exchange irreversibilities within the boiler decreased by 14.5%. That is, the power cycle modifications, primarily reheat and feedwater heating, implemented in centralized power stations since the 1910–1920 era reduced the temperature differences in the heat exchangers within the steam generator and, according to these results, are responsible for over 93% of the gain in plant efficiency attained over the last 70–80 years. The remaining 7% improvements are the modest gains experienced within the turbogenerator and the condenser.

Compared to the flow sheet shown in Fig. 8, modern power plants actually employ five to eight stages of regeneration and correspondingly, the heat exchange irreversibilities in the steam generator are about 5–6 units of input fuel exergy lower than the 21.4 value shown in Fig. 8. This reinforces the conclusion from the second-law analysis of the modern-day power plant that current-technology improvements in heat exchangers are at the stage of diminishing returns, and that thus future efforts for improving electric power plant efficiency should be directed at reducing combustion irreversibility. Consistent with historical evidence, just as technologists of the past targeted the largest irreversibility—the heat transfer losses in the boiler, technologists of today and the future must target the largest irreversibility of today's systems—the combustion losses.

CLOSURE

A way to integrate second-law analysis into teaching engineering thermodynamics, by examining the history of power plant improvements, was introduced. In this way the differences

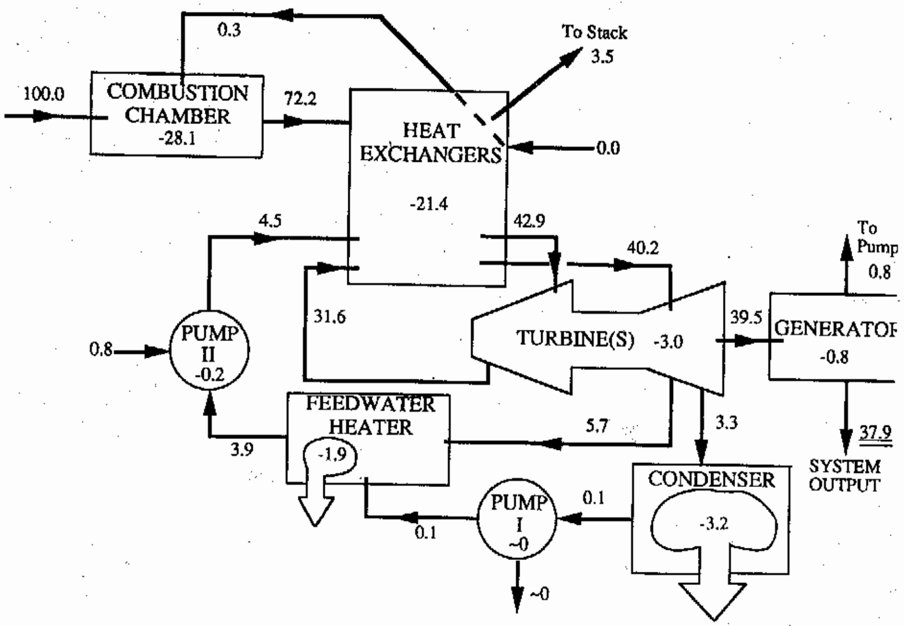

Fig. 8. Exergy flow diagram for the subcritical (2600 psig) Rankine cycle power plant (1980–1990 Era, with one feedwater heating stage).

between the conclusions of the first- and second-law analyses, and their limitations and advantages are presented and discussed. Explicit motivation for performing second-law analysis is generated by quantitative demonstration of its unique advantages in identifying the inefficiency of processes and thus the areas in which R&D has the potential of creating major improvements.

For example, it clearly identifies the combustion process in modern-day fossil fuel power plants as the most inefficient process deserving appropriate attention.

As recommended in the Introduction, it would be worthwhile to include in future analyses energy conversion system components which are used for environmental protection, so that the students learn the quantitative energy and exergy implications of this endeavour.

Naturally, a course which would use the approach described in this paper would have to start with the conventional basic definitions, description of equilibrium and the laws of thermodynamics, and thermodynamic properties.

Implicit in the recommended educational approach are the additional prerequisites of teaching the concepts of exergy, some introduction to energy conversion system components, system modelling (conservation equations, thermodynamic property relations, component performance characteristics, and boundary conditions, and methods for the solution of the resulting system of equations), behaviour of non-reacting ideal gas mixtures, and reacting ideal gas mixtures (including combustion), prior to approaching the proposed modelling and analysis of the power plants or other energy systems.

We found in practice that the proposed approach was not only effective in introducing the proper role of exergy analysis, but that in its realistic analysis of accomplished and potential

power plant improvements it also made the course much more interesting and motivating for the students.

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