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ECONOMICS AND THE SECOND LAW: AN ENGINEERING VIEW AND METHODOLOGY

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Abstract—An operationally convenient methodology is presented for pricing the penalties of thermodynamic irreversibilities occurring in equipment processes.

Starting with a recognition of the individual internal and relevant external irreversibilities thermodynamic arguments are used to formulate both entropy and energy measures in terms of operating conditions. The energy measures lead to economic pricing relating to system energy expenditure and sometimes system energy rating penalties. The analysis loop is closed by considerations relating to the reduction of the individual irreversibilities in terms of trade-off factors.

The usual available energy or 'exergy' analysis provides an answer to the overall costs of the collective internal irreversibilities. Depending on the system definition, relevant 'external' irreversibilities may be excluded. The lack of detail does not allow the development of trade-off factors and, moreover, inhibits judgments as to the relevance of an energy rating penalty in addition to an energy expenditure penalty.

NOMENCLATURE

- A, heat transfer and flow friction area;
- A_c, flow cross-section area;
- c_p, c, specific heat at constant pressure, perfect gas and incompressible liquid, respectively;
- h, enthalpy per unit mass;
- $\dot{Irrev}$, measure of irreversibility rate: subscript s entropy basis, subscript E energy basis;
- m, molal mass;
- N_{tu}, number of heat transfer units $[AU/(wc)]$, nondimensional;
- P, fluid pressure;
- P^*, pressure ratio (> 1) across pump or compressor;
- q, heat transfer rate;
- R, perfect gas constant, molal basis;
- s, entropy per unit mass;
- T, absolute temperature;
- U, overall heat transfer coefficient;
- v, specific volume, $1/\rho$: subscript fg is change on vaporization, subscript g denotes saturated vapor;
- $\dot{Wk}$, electrical or mechanical power;
- w, mass flow rate.

Greek symbols

- α, exponent related to specific heats ratio (see Table 2);
- Δ, δ, prefix denoting a small change;
- ε, heat exchanger effectiveness;
- η, energy conversion efficiency: subscript T denotes turbine, subscript G denotes electric generator, subscript P denotes pump, subscript M denotes electric motor;
- ρ, fluid density, $1/v$;
- $\phi(\ )$, denotes function of.

Miscellaneous

- $\dot{(\ )}$, denotes derivative with respect to time;
- $\approx$, approximate equivalence;
- $\triangleq$, a definition.

INTRODUCTION

IN GEORGESCU-ROGEN'S book, *The Entropy Law and the Economic Process* [1], an understandable description of the science of economics is presented as "the study of mankind in the ordinary business of life". A further statement is "that the true product of that (economic) process is an immaterial flux, the enjoyment of life". The engineer simplifies his economic thinking by expressing his design process output as monetary values, for example, cents per kWh at the busbar or ($/year)/kW rating, or as a trade-off factor, a 1% improvement here is worth x $/year to the business in operating costs, but this improvement is going to cost the business $y$$ in investment. These bottom-line factors are important ingredients of the decision making process. Quite possibly this is an oversimplification of the 'economic process', but it is both operationally convenient and reasonably defensible as a basis for action.

How does entropy, that extensive physical property which is an integral part of the conceptual basis of the Second Law of Thermodynamics, enter the discussion? Georgescu-Rogen's thesis is "that the basic nature of the economic process is entropic and that the Entropy Law rules supreme over this process and its evolution." Our Congress appears to agree with this view, as expressed in the National Energy Conservation Act PL95-619, 1978, as amended by PL96-294, 30 June 1980, Section 683, entitled "Second Law Efficiency Study." Quoting paragraph (a), "The Secretary of Energy, in consultation with the Director of the National Bureau of Standards and such other agencies as he deems necessary, shall conduct a study of the relevance to energy conservation programs of the use of the concept of energy efficiency as being the ratio of the minimum available work necessary for accomplishing a given task to the available work in the actual fuel used to accomplish that task." Clearly, some confusion may result if the task is performed with shaft power derived from hydroelectric power. This is a question for the Director of the National Bureau of

Standards to resolve. The objective of this presentation is to show, in a somewhat conventional manner, how entropy enters into the "ordinary business of life" using the economic methodology of developing the monetized bottom-line factors discussed previously.

THERMODYNAMICS

In thermodynamic analysis, one accounts for mass, energy and entropy. This accounting or 'bookkeeping' procedure can be unified by expressing the Second Law principle in the same format as the conservation of mass and energy principles. One such format is based on the following definition of the rate of creation of (something) *within the system*

$$CRE\dot{A}TION \text{ of (something)}$$
$$\triangleq \sum OUT\dot{F}LOW \text{ of ( )} - \sum IN\dot{F}LOW \text{ of ( )} + \sum INCR\dot{E}ASE \text{ of storage ( ).} \quad (1)$$

As a very important part of this definition, rate of creation is restricted to the concept of a system, or control volume, a closed region in space, *specified by the analyst*, for the purpose of investigating the (something) of interest. As a rate basis is specified, the accounting is done at an instant of time. The accounting principles of thermodynamics now become

$$CREA\dot{T}ION \text{ of mass} = 0, \quad (2)$$

$$CREA\dot{T}ION \text{ of energy} = 0, \quad (3)$$

$$CRE\dot{A}TION \text{ of entropy} \geq \Sigma(9/T)_{in} - \Sigma(9/T)_{out} \quad (4a)$$

In this last statement, the Second Law principle, the $(q/T)_{in}$ terms are for heat transfer rates *into* the system *from the surroundings*, with the T terms the absolute temperature just *within* the system boundaries where the individual q_{in} terms are received. In contrast, the $(q/T)_{out}$ terms are heat transfer rates *out* of the system to the surroundings, with the T terms again measured just *within* the system boundaries located where the individual q_{out} terms are delivered to the surroundings. The equal sign in the Second Law belongs to the idealized system where all processes internal to the system occur in a *reversible manner*. The inequality sign belongs to the real-life system, where *internal irreversibilities* may be minimized but never reduced to zero.

The irreversibilities that we all recognize as part of reality are listed in Table 1. Certainly there are more than the thirteen specified, so the listing must be considered as open-ended.

With the foregoing conceptual basis for the Second Law, including the concept that entropy is that particular extensive property of matter which fits the formulation equation (4a), a natural *entropy measure* of the collective *internal* irreversibilities is provided by the strength of the inequality sign. This leads to the definition

$$Ir\dot{r}ev_s \triangleq CRE\dot{A}TION \text{ of entropy} - [\Sigma (q/T)_{in} - \Sigma (q/T)_{out}]. \quad (5)$$

The fact that

$$Ir\dot{r}ev_s \geq 0 \quad (4b)$$

is then just a reformulation of the Second Law, equation (4a).

If the "Entropy Law rules supreme" and the "economic process is entropic," Georgescu-Rogen is electing, as is his privilege, to locate his economic system boundaries so as to have an adiabatic system.

Only for an adiabatic system* is the rate of creation of entropy always positive. Otherwise, it can be positive, negative, or zero, depending on the RHS (q/T) terms of equation (4a). What can be claimed is that the *entropy measure of irreversibilities in the economic process* always tends with time in one direction, as given by equation (4b). Some modern texts on thermodynamics, as, for example, ref. [2], refer to the RHS of equation (5) as the 'production of entropy', electing to

* Note that an isolated system is much more restrictive than an adiabatic system, as no boundary fluxes of mass and energies in addition to q terms are allowed.

Table 1. Irreversibilities in energy conversion systems

1.	Fluid friction in flow over solid surfaces
2.	Solid-to-solid surface friction
3.	Flow throttling
4.	Free expansion e.g. blowdown, explosion
5.	Mixing of dissimilar fluids
6.	Mixing of similar fluids at different temperatures
7.	Solution of a solid in a liquid
8.	Plastic deformation of a solid
9.	I^2R heating in an electrical conductor
10.	Electromagnetic hysteresis
11.	Virtually all chemical reactions e.g. combustion
12.	Heat transfer across a finite temperature difference
13.	Phase change when initial conditions are not in equilibrium
14.	Open-ended listing

interpret the (q/T) terms as 'flow rate of entropy with heat transfer'. This is unfortunate from a pedagogical viewpoint, because it undercuts the conceptual basis of the Second Law contained in the idea that there is an *extensive property* of matter, which we elect to call entropy, that has the characteristic implied in equation (4a). If a property is extensive, it can only flow in and out of a system with associated mass flows. This is not a characteristic of (q/T). From an operational point of view, accounting for entropy is simplified if it has a single nature, mass-associated only, and not a dual nature, both mass and heat transfer associated.

IRREVERSIBILITY EVALUATION

Bejan [3] and others have proposed a Keenan type "available energy" measure of irreversibility, as distinct from the entropy measure of equation (5). More recently, Bejan and Pfister [4] proposed that "the merit of a given heat transfer augmentation technique (be evaluated) by comparing the rate of entropy generation present in an augmented duct with the entropy-generation rate in a reference duct." This "entropy-generation rate" is identical to the entropy measure of irreversibility of equation (5) evaluated for the duct system under consideration.

An energy measure of irreversibility is defined relative to the entropy measure as follows:

$$Irrev_E \triangleq T_{WF} Irrev_s \tag{6}$$

for any particular irreversibility. Here, T_{WF} is a *temperature-weighting factor* to be specified by the analyst based on his judgment of its relevance to the system being studied. It can be demonstrated that if T_0, the temperature of the "natural thermal sink" located in the surroundings of the system, is selected as T_{WF}, then $Irrev_E$ becomes identical to an "availability" or "exergy" measure of the irreversibility in question [5–8]. The arguments in favor of selecting an entropy measure rather than an availability measure are as follows:

(i) The entropy measure ties in directly with the inequality sign of the Second Law. In this sense, as it is more basic, there should be general agreement on its usage.
(ii) It is operationally more convenient to account for entropy as a single extensive function of state than to account for the availability functions $(h - T_0 s)$ for the flow terms and $(u + P_0 v - T_0 s)$ for the storage terms, involving a multiplicity of both extensive and intensive state functions.
(iii) The conversion of the entropy measure to the availability measure can be achieved readily [equation (6)].
(iv) As will be seen, in some situations it is more reasonable to use a temperature-weighting factor other than T_0.

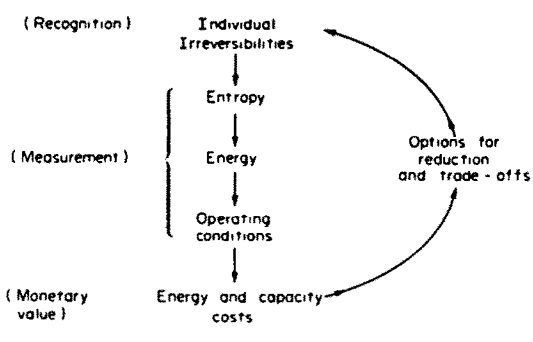

FIG. 1. Entropy and economics.

The author's position is that the operationally more convenient entropy measure will enhance both the teaching and professional application of 'Second Law Analysis' of energy systems.

IRREVERSIBILITY AND ECONOMICS

The schematic presentation of Fig. 1, entitled Entropy and Economics, provides a framework for the methodology of relating irreversibilities to monetized economic gains and costs. The position taken in this presentation is that the analysis should start from a *recognition of the individual irreversibilities* that significantly relate to the function of the system being analyzed. As an example, consider the condenser of a steam–electric power plant described in Fig. 2. The function of the condenser is to dump thermal energy to the surroundings (river or ocean), which is at a temperature T_0. This dumping is to be accomplished with as much thermodynamic grace as can be afforded. The more obvious irreversibilities are those internal to the condenser system A, namely, the heat transfer and two flow-friction irreversibilities. Less obvious are the following: a mixing irreversibility, where the heated cooling water is returned to the thermal sink (ocean or river); the flow friction and throttling occurring within the pump; the flow friction in the piping transporting the cooling water; the bearing friction windage and I^2R losses in the electric motor pump-drive; and the heat leak to the surroundings from the condenser shell. These are external irreversibilities that should be charged to the condenser, in view of its function to dump q with *affordable* thermodynamic grace. To establish the *external* (to the condenser) irreversibilities requires analysis of additional systems, such as B and C identified in Fig. 2.

One can justifiably argue that the q_{leak} irreversibility is not significant for this heat exchanger, because it aids rather than hinders the q dumping function. This would not be the case for the regenerative feed-water heaters in the power plant, or the cold-box of a cryogenic system used to liquefy natural gas.

The evaluation of the specific irreversibilities using an entropy measure requires bookkeeping according to the defining equations (1) and (5). Evident from equation (5) are the advantages of describing an adiabatic system when feasible. System A of Fig. 2, with the idealization of negligible q_{leak}, is such an

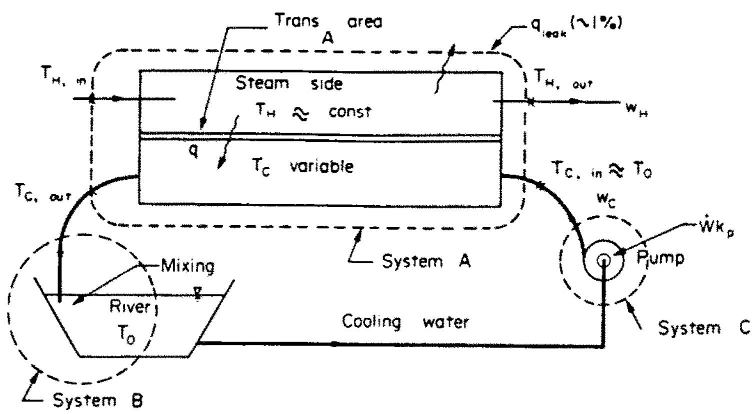

FIG. 2. Steam—electric power plant condenser, 1965 state-of-the-art. $A = 435\,000\ \text{ft}^2$ (~ 10 acres); 25 590 al. bronze 18 BWG tubes 1" OD, 65 ft long; $\dot{W}k_{net} = 740\,000$ kW; Heat rate = 8700 B.t.u. kWh^{-1}; $q/\dot{W}k_{net} = 1.117$; $T_0 = 520°R\ (60°F)$, $T_{C,in}$; $T_{C,out} = 540°R\ (80°F)$ $w_C/q = 1/20$ lbs/B.t.u.; $T_{H,in} = 544.7°R\ (84.7°F)$, sat. at 1.2 in Hg abs; $(\Delta P/\rho)_C = 22$ ft lb$_f$/lb $= 2.83 \times 10^{-2}$ B.t.u./lb; Pump head $= 1.5\,(\Delta P/\rho)_C$, $\eta_{pump} = 80\%$; $\Delta P_H = 0.20$ in Hg.

adiabatic system, and equation (5) will yield the entropy measure of irreversibility for the three *internal* components, the heat transfer and the two flow-friction components. Since entropy is an extensive property, the conservation of matter principle, equation (2), is useful in combination with equation (5). Additionally, conservation of energy, equation (3), and the appropriate equations of state, Table 2, for the fluids involved are introduced. The end results should be expressed in terms of *directly measurable operating conditions*, such as mass flow rates, energy ratings, pressures and temperatures, and not in terms of entropies or enthalpies. Further, when the conversion is made from the entropy to the energy measure, equation (6), a judgment is required as to the selection of a temperature-weighting factor T_{WF}. As an illustration, for the condenser of Fig. 2, the end results for the individual irreversibilities are, for system A,

$$\left.\frac{Ir\dot{r}ev_E}{q}\right|_q = \left.\frac{T_{WF}Ir\dot{r}ev_s}{q}\right|_q = \left[1 - \frac{T_{C,lm}}{T_{H,lm}}\right]\frac{T_{WF}}{T_{C,lm}} \tag{7}$$

where T_{lm} is the logarithmic average of T_{in} and T_{out}.

$$\left.\frac{Ir\dot{r}ev_E}{q}\right|_{\Delta P_C} = w_C\frac{(\Delta P_C/\rho)}{q}\frac{T_{WF}}{T_{C,in}} \tag{8}$$

$$\left.\frac{Ir\dot{r}ev_E}{q}\right|_{\Delta P_H} = \frac{(\Delta P_H/\rho_g)}{2h_{fg}}\frac{T_{WF}}{T_{H,ave}} \tag{9}$$

where an arithmetic average is suitable for $T_{H,ave}$.

Similarly, analysis for the other systems of Fig. 2 yields

$$\left.\frac{Ir\dot{r}ev_E}{q}\right|_{mix} = \frac{w_C c}{q}\left[\left(\frac{T_{C,out}}{T_0} - 1\right) - \ln\frac{T_{C,out}}{T_0}\right]T_{WF}, \tag{10}$$

$$\left.\frac{Ir\dot{r}ev_E}{q}\right|_{q_{leak}} = \frac{q_{leak}}{q}\left[1 - \frac{T_0}{T_{H,ave}}\right]. \tag{11}$$

Table 2. Equation-of-state summary

System	Equations of State
Single-phase fluids	$h(P, T)$, $s(P, T)$
Two-phase fluid (e.g. evaporating or condensing)	$h(P)$, $P(T)$; $\left.\frac{dP}{dT}\right\|_{sat} = \frac{h_{fg}}{Tv_{fg}}$ (The Clapyron equation)
Perfect gas with c_P = constant	$(h_2 - h_1) = c_P(T_2 - T_1) = \phi(T)$ only; $(s_2 - s_1) = c_P \ln\frac{T_2}{T_1} - \frac{R}{m}\ln\frac{P_2}{P_1} = \phi(T) + \phi(P)$; $\alpha \triangleq \frac{k-1}{k}$, $k \triangleq c_P/c_V$
Incompressible liquid with c = constant	$(h_2 - h_1) = c(T_2 - T_1) + \frac{(P_2 - P_1)}{\rho} = \phi(T) + \phi(P)$; $(s_2 - s_1) = c\ln\frac{T_2}{T_1} = \phi(T)$ only

$$\left.\frac{Irrev_E}{q}\right|_{pump} = \frac{(1-\eta_P)}{\eta_P}\frac{\dot{W}k_s}{q}. \quad (12)$$

In equation (12), the isentropic pump power requirement is

$$\dot{W}k_s = w_C \times \text{pump head}.$$

The pump head is specified in Fig. 2 as being greater than the $(\Delta P_C/\rho)$ of equation (8) by 50%, as a result of additional piping friction. Since the function of the condenser is to dump thermal energy to the surroundings, a temperature-weighting factor, $T_{WF} = T_0$, is appropriate. Numerical results for the situation described in Fig. 2 are summarized in the second column of Table 3. Note that the heat transfer and mixing irreversibilities dominate.

Reverting to Fig. 1, the tasks that remain are to relate the results of the component irreversibility measurements, first to the power plant energy and capacity costs, and then to consider options for their reduction.

Capital costs for the 1965 state-of-the-art plant described in Fig. 2 are specified as \$200/kW capacity at an annual interest rate of 18%. Average energy costs for fuel oil and/or natural gas are taken as \$5/10⁶ Btu. The plant heat rate is 8700 B.t.u./kWh; 4000 h/yr full power equivalent operation is assumed. These specifications yield annual charges of \$36/kW rating for capital and \$174/kW for busbar energy delivery. Energy charges dominate over capital charges because of the high value of petroleum fuels (oil and gas) at \$5/10⁶ B.t.u., amounting to 4.35¢/kWh for the heat rate of 8700 B.t.u./kWh.

A consideration of the irreversibilities listed in Table 3 leads to the following conclusions:

(i) *The heat leak irreversibility* has no rating or energy charge associated with it, because the condenser function is to dump thermal energy to the surroundings, and the heat leak contributes to this function. This zero charge would not obtain for other exchangers in the plant, such as a feedwater heater.

(ii) *The other irreversibilities* of Table 3 all have both rating (or capital) and energy charges. This is not necessarily the case for other exchangers in the plant. As an example, the heat transfer and mixing irreversibilities associated with the boiler exhaust-gas-to-combustion-air preheater would have only the energy charge, but no rating charge.

(iii) *The mixing, heat transfer* and the *steam-side flow friction irreversibilities* all subtract from busbar output through the energy-conversion processes of the steam turbine and the electrical generator. Consequently, these irreversibilities should be *discounted* by the inefficiencies of these energy conversion processes to express their busbar costs. Then

$$\left.\frac{\delta \dot{W}k_{net}}{\dot{W}k_{net}}\right|_{busbar} = (\eta_T\eta_G)\left(\frac{Irrev_E}{q}\right)\left(\frac{q}{\dot{W}k_{net}}\right).$$

A combined turbine-and-generator efficiency of 80% and a $(q/\dot{W}k_{net}) = 1.117$ (from Fig. 2) was used for producing the entries in the fourth column of Table 3. For example, for the mixing irreversibility, the *discounted cost* is

$$\left.\frac{\delta \dot{W}k_{net}}{\dot{W}k_{net}}\right|_{busbar} = 0.80 \times 0.0188 \times 1.117 = 0.0168$$

and the monetized value of this loss is

$$0.0168 \times 740000 \times (174 + 36) = \$2.61 \times 10^6/\text{yr}.$$

With the exception of the three pumping requirement irreversibilities associated with ΔP_C, the other busbar costs in Table 3 were obtained in a similar manner.

(ii) The condenser cooling water *flow-friction irreversibilities* associated with ΔP_C, water pipe friction, and pump losses $(1 - \eta_P)$, are paid for by a direct subtraction from busbar output to power an electric motor. Consequently, the motor losses are also part of the busbar charge, and

$$\left.\frac{\delta \dot{W}k_{net}}{\dot{W}k_{net}}\right|_{busbar} = \frac{1}{\eta_M}\left(\frac{Irrev_E}{q}\right)\left(\frac{q}{\dot{W}k_{net}}\right).$$

For an electric motor efficiency of 90%, the busbar *appreciated cost* for the cooling water ΔP_C is

$$\frac{1}{0.90} \times 0.00142 \times 1.117 = 0.00176$$

Table 3. Irreversibility accounting summary (Condenser of Fig. 2)

	$\frac{Irrev_E}{q}$	% of total	Busbar costs $\frac{\delta \dot{W}k_{net}}{\dot{W}k_{net}}$	10^6 \$/yr	(% of total)
Mixing	0.0188	35	0.0168	2.61	(35)
Heat transfer	0.0267	50	0.0239	3.71	(50)
Heat leak	0.00045	1	0	0	
Steam-side ΔP_H	0.00458	9	0.00409	0.64	(9)
Condenser ΔP_C	0.00142	3	0.00176	0.27	(4)
Pump $(1 - \eta_P)$	0.00053	1	0.00066	0.10	(1)
Pipe friction	0.00070	1	0.00087	0.14	(2)
Summation	0.0532		0.0481	7.47	

with a price of

$$0.00176 \times 740\,000(174 + 36) = \$0.274 \times 10^6/\text{yr}.$$

Pump inefficiency and pipe-friction irreversibilities are priced in a similar manner.

The total costs of the condenser irreversibilities identified in Table 3 are impressive. The impact on the plant heat rate is 4.8%, and their costs, including both capital and energy charges, amounts to $7.5 × 10^6/yr.

Again referring to Fig. 1, the analysis for this illustrative example has progressed to the stage of considering design (and/or operating) options for the reduction of the component irreversibilities and the development of trade-off factors.

The *mixing irreversibility*, equation (10), with $T_{WF} = T_0 \approx T_{C,in}$ and $q = w_C c_l (T_{C,out} - T_{C,in})$, reduces to

$$\left.\frac{Irrev_E}{q}\right|_{mix} = \left[1 - \frac{T_0}{T_{C,lm}}\right], \tag{13}$$

$T_{C,lm} \triangleq (T_{C,out} - T_{C,in})/\ln(T_{C,out}/T_{C,in})$.

The only option for reduction is by increasing the flow rate w_C to reduce $T_{C,out}$. Infinite w_C would result in $T_{C,lm} = T_{C,in} = T_0$, and the mixing irreversibility would vanish.

The similarity of equations (13) and (7) for the *heat transfer irreversibility* suggests an addition of the two irreversibilities to yield the very simple (and possibly obvious) result

$$\left.\frac{Irrev_E}{q}\right|_{mix+q} = \left[1 - \frac{T_0}{T_H}\right]. \tag{14}$$

Then, by differentiation,

$$\delta(Irrev_E/q)_{mix+q} = T_0 \frac{\delta T_H}{T_H^2}. \tag{15}$$

Thus, for the condenser problem under consideration, a 1 °R production in T_H, from a reduction of either a mixing or heat transfer irreversibility, results in 620 × 1/544.7² = 0.00175 reduction in the (0.0188 + 0.0267) = 0.0455 sum for the mixing plus heat transfer irreversibilities from Table 3. The monetized savings is (0.00175/0.0455) × 6.3 × 10^6 = $242,000/yr.*

This is about as far as one can go with thermodynamics alone. Further progress requires additional physics input, namely, heat-transfer rate considerations and fluid mechanics. The end results of this extended analysis are as follows:

$$-\left.\frac{\delta Irrev_E}{q}\right|_q = \frac{T_0}{T_H^2}(T_H - T_0) \times \left[\frac{(1-\varepsilon)}{\varepsilon} N_{tu}\right] \frac{\delta(AU)}{(AU)} \tag{16a}$$

$$-\left.\frac{\delta Irrev_E}{q}\right|_{mix} = \frac{T_0}{T_H^2}(T_H - T_0) \times \left[1 - \frac{(1-\varepsilon)}{\varepsilon} N_{tu}\right] \frac{\delta w_C}{w_C}. \tag{17a}$$

For the condenser example at hand, Fig. 2, ε = (540 − 520)/(544.7 − 520) = 0.8097 and N_{tu} = 1.6592. Equations (16a) and (17a) then become

$$-\left.\frac{\delta Irrev_E}{q}\right|_q = 0.0169 \frac{\delta(AU)}{(AU)} \tag{16b}$$

$$-\left.\frac{\delta Irrev_E}{q}\right|_{mix} = 0.0264 \frac{\delta w_C}{w_C}. \tag{17b}$$

Using the above relations and the busbar costs of Table 3, it is noted that a 1% increase in the (AU) product will result in [0.0169 × 10^{-2}/0.0267] × 3.71 × 10^6 = 23.4 × 10^3 $/yr saving, and a 1% increase in the cooling water rate will save [0.0264 × 10^{-2}/0.0188] × 2.61 × 10^6 = 36.7 × 10^3 $/yr. Apparently a 1% increase in the cooling water rate is 1.57 times more effective than a 1% increase in the (AU) product. In this manner, options for the reduction of irreversibilities can be compared.

Both increases, $\delta(w_C)/(w_C)$ and $\delta(AU)/(AU)$, will tend to increase $Irrev_{E,\Delta P_C}$. However, this tendency can be countered by increasing the flow area and thereby reducing the velocity. Differentiation of equation (8), and using the usual friction factor expression for relating the pressure drop to the flow velocity, flow area A_C, and friction (heat transfer) area A yields

$$\left.\frac{\delta Irrev_E}{Irrev_E}\right|_{\Delta P_C} = \frac{\delta A}{A} + \frac{3\delta w_C}{w_C} - \frac{3\delta A_C}{A_C}. \tag{18}$$

Equation (18) may be interpreted as follows. The 1% increase in transfer area A [to increase the (AU) product] can be compensated by a 1/3% increase of *flow area* A_C, while the 1% increase in cooling water rate w_C needs a 1% increase of A_C to compensate. However, after noting from Table 3 that ΔP_C and its associated pump and piping irreversibilities are contributing only 7% of the busbar impact of the condenser total irreversibilities, but costing $510 × 10^3/yr, the designer may decide that the increases of 1% in both w_C and A together are well worth the 4% increase (keeping flow area A_C constant) of the ΔP_C irreversibility at the busbar, since the net gain, using the numbers of Table 3†, is $42 800 per year. The reluctance to increase A_C

Table 4. Trade-off factors

1% increase in	Busbar costs* †($/yr)
Cooling water rate, w_C	− 36 700
Heat transfer surface, (AU)	− 23 400
Cooling water pressure drop†, ΔP_C	+ 5100
Steam side pressure drop, ΔP_H	+ 6400

* + increase, − decrease.
† Including the associated pump and pipe friction losses, assumed to scale with ΔP_C.

* From the Clapeyron equation of Table 2, δT_H of −1°R corresponds to a reduction of steam-side condenser pressure by 2.70 psf/ft² or 0.0383 in Hg.

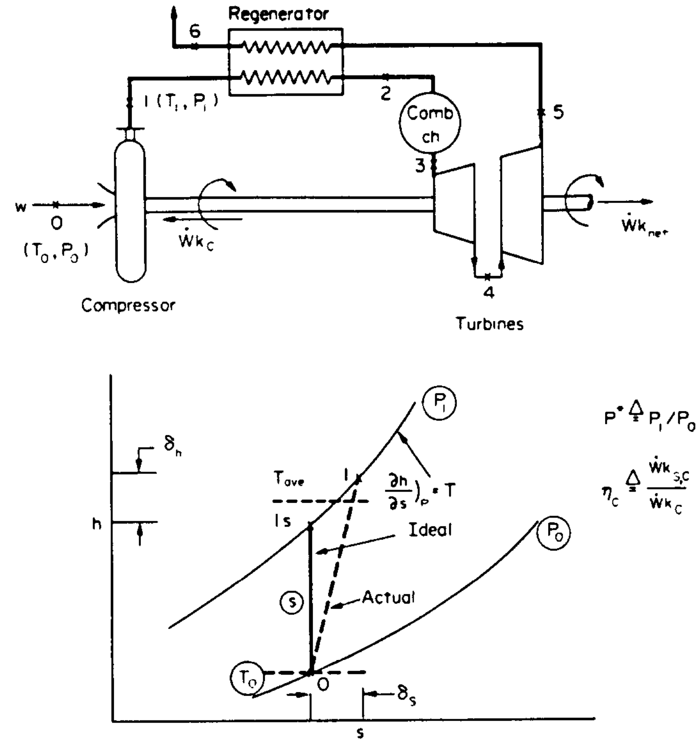

FIG. 3. The compressor in a gas turbine engine.

may well stem from the desire to keep the number of tube-to-header connections to a minimum, as these joints are expensive to fabricate and also a source of leaks that contaminate the condensate.

For the *steam-side flow-friction irreversibility* represented by ΔP_H, equation (9) yields

$$\left.\frac{\delta Irrev_E}{Irrev_E}\right|_{\Delta P_H} = \frac{\delta(\Delta P_H)}{\Delta P_H}. \tag{19}$$

In terms of the condenser example of Fig. 2 and the numerical values of Table 3, a 1% increase of ΔP_H from 0.200 to 0.202 in Hg results in a penalty at the busbar of \$6400 per year. Options for reduction include increasing the transverse tube pitch and provision for steam path lanes. These options reduce the steam velocity over the heat transfer surfaces.

Table 4 summarizes the trade-off factors as previously developed for the illustrative example of the steam power plant condenser.

At the design stage, a study of these trade-off factors could well result in a reformulation of the specifications for the condenser system. For an existing plant, these factors can be used to arrange for cleaning schedules and, possibly, for an increase in w_C by a pumping speed increase, since frequency changers may shortly become a state-of-the-art device.

In the foregoing example of the steam–electric power plant condenser, it is shown how the 'entropy law' enters into "the ordinary business of life." Whether or not the "entropy law rules supreme" is a matter of opinion. Certainly, in this example, additional physics input was required in the form of the conservation of matter and energy principles; and equations of state input was also required before the irreversibility losses, recognized as the starting point of the analysis, Fig. 1, could be priced. After pricing was accomplished, Table 3, more physics input was required in the form of heat transfer and flow friction relationships in order to consider the options for reduction and to arrive at the monetized trade-off factors of Table 4. From an engineering viewpoint, it is the 'closed loop' of considerations, described in Fig. 1, that relates entropy and economics in an operationally useful methodology.

THE TEMPERATURE-WEIGHTING FACTOR

An ambiguity that remains in this methodology is the selection of the temperature-weighting factor T_{WF} introduced in equation (6) in order to convert the entropy measure of irreversibility to an energy measure. Since this selection is a matter of judgment and since it is the energy measure that is needed to price the irreversibility in question, different analysts may arrive at different prices. The example of the condenser, Fig. 2, was selected because the function of the system was clearly to dump thermal energy into a sink at temperature T_0 so that general agreement on $T_{WF} = T_0$ could be expected. Another example will now be considered where there might be room for different opinions on the selection of a suitable T_{WF}.

Consider the compressor component of the regenerative gas turbine system described in Fig. 3. The collective irreversibilities (flow friction and throttling) will be lumped together, and the compressor will be treated as adiabatic. Entropy and mass bookkeeping, in accordance with equations (1), (2) and (5) yields the result

$$Ir\dot{r}ev_s = w(s_1 - s_0) = w\delta s. \qquad (20)$$

Introducing conservation of energy, equation of state, and the definition of compressor efficiency, η_C, yields

$$Ir\dot{r}ev_s = wc_p \ln\left\{\left[\frac{(P^{*x} - 1)}{\eta_C} + 1\right]\Big/P^{*x}\right\} \qquad (21)$$

with $Ir\dot{r}ev_E \triangleq T_{WF}\, Ir\dot{r}ev_s$

A First Law treatment for the compressor shaft power yields the conventional result

$$\dot{W}k_C = w(h_1 - h_0)$$

and for the 'ideal' (reversible adiabatic) compressor,

$$\dot{W}k_{s,C} = w(h_{1,s} - h_0).$$

The definition of compressor efficiency relates these two shaft power requirements

$$\eta_C \div \frac{\dot{W}k_{s,C}}{\dot{W}k_C}. \qquad (22)$$

The compressor designer recognizes the irreversibilities of the actual compressor as

$$\dot{W}k_C - \dot{W}k_{s,C} = w\delta h = (1 - \eta_C)\,\dot{W}k_C$$
$$= \left(\frac{1 - \eta_C}{\eta_C}\right)\dot{W}k_{s,C}\,. \qquad (23)$$

The ratio of the compressor designer's loss to the entropy measure loss [equation (20)] is given by (see Fig. 3)

$$\frac{w\delta h}{w\delta s} \approx \left.\frac{\partial h}{\partial s}\right|_P = T_{ave} \qquad (24)$$

where T_{ave} is an average of T_1 and T_{1s}. Clearly, if the designer elects T_{ave} as his T_{WF} for equation (21), the result for the $Ir\dot{r}ev_E$ measure will be accepted. However, he may prefer the following equivalent more simple formulation, derived from equation (23)

$$Ir\dot{r}ev_E = \left(\frac{1 - \eta_C}{\eta_C}\right)wc_PT_0(P^{*x} - 1). \qquad (25)$$

The essential equivalence of equation (21) with $T_{WF} = T_{ave}$ and equation (25) can most readily be shown by a numerical experiment. A symbolic proof has some difficulties in the prior specification of the appropriate averaging procedure for $T_{ave}(T_1, T_{1s})$. In fact, forcing the equivalence can be used to define the proper average. Practically, this is not an important consideration, because the ratio T_1/T_{1s} is close enough to unity that an arithmetic average will be quite adequate for a numerical evaluation.

In light of the foregoing, it is of interest to reconsider equation (12) used to evaluate the pump irreversibility in the condenser example of Fig. 2. It is recognized as paralleling the form of equation (25) for the compressor. Starting with equations (20) and (24), which are also applicable to the adiabatic pump, and treating water as an incompressible liquid yields, for the equation paralleling equation (21),

$$Ir\dot{r}ev_E = T_{WF}\, Ir\dot{r}ev_s$$
$$= wcT_{WF} \ln\left[1 + \left(\frac{1}{\eta_P} - 1\right)\left(\frac{P_0}{cT_0}\right)\left(\frac{P^* - 1}{\rho}\right)\right]. \qquad (26)$$

It is easy to show numerically, for $P^* < 100$ and $\eta_P > 50\%$, that the second term in the square bracket is very small compared to unity, so the approximation $\ln(1 + x) \approx x$ for $x \ll 1$ is applicable. Equation (12) results after noting that $T_0 \approx T_1 \approx T_{WF}$ for the pump process, unlike the compressor process, because of the difference in the equation-of-state behavior, Table 2.

A physical chemist might still prefer to use a $T_0 = T_{WF}$ weighting factor for the compressor. His argument would be that all of δh in Fig. 3 is not a loss, as a reversible cooling from T_1 to $T_{1,s}$ of the discharge air, using a 'Carnot elevator' to lower the thermal energy to T_0, will yield an incremental work term subtracting from the loss of equation (25). The engineer's preference would be the use of a reversible adiabatic compressor as the reference for comparison of actual-to-ideal, rather than a reversible adiabatic compressor, plus a 'Carnot elevator'. In any event, the engineer will be able to appreciate the physical chemist's view, and vice versa, as they will both agree on the entropy measure of irreversibility, equation (20)—namely, the strength of the inequality sign in the Second Law, equation (5). To repeat an earlier statement, T_{WF} in equation (6) is a weighting factor to be specified by the analyst, using a judgment of its relevance to the system being considered.

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L'ECONOMIE ET LA SECONDE LOI: UNE VISION D'INGENIERIE ET UNE METHODOLOGIE

Résumé—On présente une méthodologie opérationnelle pour évaluer le coût des inconvénients liés aux irréversibilités thermodynamiques qui apparaissent dans les procédés industriels.

Partant de l'inventaire des irréversibilités individuelles internes et externes, des arguments thermodynamiques sont utilisés pour formuler des mesures d'entropie et d'énergie en terme de conditions opératoires. Les mesures d'énergie conduisent à une évaluation économique reliée au coût du système énergétique. L'analyse est bouclée par des considérations reliant la réduction des irréversibilités individuelles internes en terme de facteurs économiques.

L'énergie utilisable usuelle ou l'analyse exergétique fournit une réponse aux coûts globaux des irréversibilités internes collectives. Selon la définition du système, des irréversibilités "externes" peuvent être exclues. Le manque de détails ne permet pas le développement des facteurs économiques.

WIRTSCHAFTLICHKEIT UND ZWEITER HAUPTSATZ: EINE BETRACHTUNGSWEISE UND METHODOLOGIE AUS DER SICHT DES INGENIEURS

Zusammenfassung—Eine bequem zu handhabende Methodologie wird vorgestellt, mit der man die Verluste durch thermodynamische Irreversibilitäten in Apparaten beurteilen kann. Beginnend mit einer Darstellung der einzelnen inneren und der relevanten äußeren Irreversibilitäten werden mit thermodynamischen Argumenten sowohl Entropie- als auch Energiegrößen als Funktion der Betriebsbedingungen formuliert. Die Energiegrößen führen zu einer ökonomischen Bewertung, die den Zusammenhang zu den Energieverlusten des Systems und manchmal zu den Verlustfaktoren der Energieberwertung des Systems herstellt. Der Kreis der Untersuchung wird geschlossen durch Betrachtungen zur Verminderung der einzelnen Irreversibilitäten in Abhängigkeit von Einflußfaktoren (trade-off factors).

Die übliche Exergie-Analyse liefert die Gesamtkosten aller inneren Irreversibilitäten. Je nach der Systemdefinition bleiben relevante "äußere" Irreversibilitäten dabei ausgeschlossen. Die mangelnde Betrachtung von Detailvorgängen gestattet keine Formulierung von Einflußfaktoren. Darüber hinaus verhindert das Fehlen von Detailbetrachtungen die Möglichkeit zu beurteilen, ob Energiebewertungs- außer den Energieverlustkriterien relevant sind.