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Fundamentals of exergy analysis, entropy generation minimization, and the generation of flow architecture

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Research Article

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SUMMARY

The principles of exergy analysis and entropy generation minimization (also known as thermodynamic optimization—the minimizing of exergy destruction) are presented in this work. The concepts of irreversibility, entropy production, and exergy annihilation are reviewed at the beginning of the study. Examples show how energy fluxes and buildup are accounted for in power and refrigeration plants, heat transfer processes, closed systems, and open systems. The designer searches for better thermodynamic performance under certain environmental circumstances and finite-size restrictions due to the proportionality between exergy destruction and entropy formation. Solar energy, sensible heat and latent heat energy storage systems, and the production of maximum power in a power plant model with limited heat transfer surface inventory are used as examples. It is shown that the process of global thermodynamic optimization under global constraints gives rise to the system's physical structure (geometric configuration, topology). This idea creates structure in physics, biology, and engineering (constructal theory). All rights reserved. John Wiley & Sons, Ltd., 2002.

KEY WORDS: exergy analysis; entropy generation minimization; EGM; thermodynamic optimization; constructal theory; topology optimization; self-organization in nature; self-optimization in nature

EXERGY, NOT ENERGY

Power system analysis and thermodynamic improvement have completely changed as a result of the energy crisis of the 1970s and the ongoing focus on efficiency (fuel resource conservation). Thermodynamic optimization, sometimes referred to as entropy generation minimization (EGM), is the optimization component of the new approach, which is energy analysis. The foundation of this new method is the simultaneous application of the first and second laws in design and analysis (Evans, 1969; Reistad, 1970; Nerescu and Radcenco, 1970; Brodyanskii, 1973; Haywood, 1980; Ahern, 1980). It rose to prominence in engineering education during the 1990s (Moran, 1989; Bejan, 1982, 1988; Feidt, 1987; Sieniutycz and Salamon, 1990; Kotas, 1995; Moran and Shapiro, 1995; Radcenco, 1994; Shiner, 1996; Bejan et al., 1996), and it is currently permeating every facets of engineering practice (Stecco and Moran, 1990, 1992; Valero and Tsatsaronis, 1992; Bejan et al., 1999).

For computer-assisted design and optimization, it works very well (Sciubba and Melli, 1998; Sciubba, 1999a, b). In this article, we go over the method's foundations, its present state, and several instances of how it might be used to system optimization. There have been several published evaluations of the rapidly growing topic of energy and entropy creation (Bejan, 1996a, b, 1997).

In order to prevent misunderstanding with the conventional energy-based techniques of thermal system analysis and design, we must first differentiate between exergy and energy. Heat transfer, work, and mass flow are the three ways that energy enters and exits a system (e.g. shafts, piston rods). The first law of thermodynamics is that energy is preserved rather than destroyed. Exergy is a whole other idea. It quantifies the "useful" energy, or the capacity to do or receive work—the work content—of the many different streams (heat, mass, and work) that move through the system. The ability to compare various interactions (inputs, outputs, work, heat) on a common basis is the first characteristic of the property "exergy."

Another advantage is that the degree to which the system depletes energy may be ascertained by taking into consideration all of the system's energy sources. The created entropy and the destroyed exergy are proportionate. The second rule of thermodynamics states that energy is always lost, either completely or partly. The system's lower-than-theoretical thermodynamic efficiency is caused by the created entropy or the destroyed exergy.

We may create a map of the distribution of exergy destruction throughout the engineering system of interest by carrying out exergy accounting in progressively smaller subsystems. This allows us to identify the elements and mechanisms (activities) that most severely deplete energy. This is a significant benefit in the quest to increase efficiency (usually via limited means) as it provides guidance on how to distribute engineering resources and effort from the outset. In the Section 4.1 example, we return to the ideal distribution of resources.

Today, the focus of engineering thermodynamics is on determining the mechanisms and system elements that cause thermodynamic losses, the magnitude of these losses (exergy analysis), minimizing the losses subject to the system's global constraints (entropy generation minimization), and minimizing the overall costs of constructing and running the energy system (thermoeconomics). This study does not aim to review the principles of thermoeconomics. Bejan et al. (1996) describe thermoeconomics and its applications as an evolutionary step beyond exergy analysis and entropy production reduction.

Thermodynamic optimization, often known as entropy generation minimization (EGM), is an area of study at the nexus of fluid mechanics, engineering thermodynamics, and heat transport. Figure 1 shows the field's location. The approach is based on the simultaneous application of engineering thermodynamics, fluid mechanics, and heat and mass transport concepts in order to create realistic models of installations, devices, and processes. Realistic models take into consideration the intrinsic irreversibility of engineering systems and

procedures.
In the early phases of design, thermodynamic optimization may be utilized independently (without cost reduction) to find patterns and potential optimization possibilities. The structural features and optima found via thermodynamic

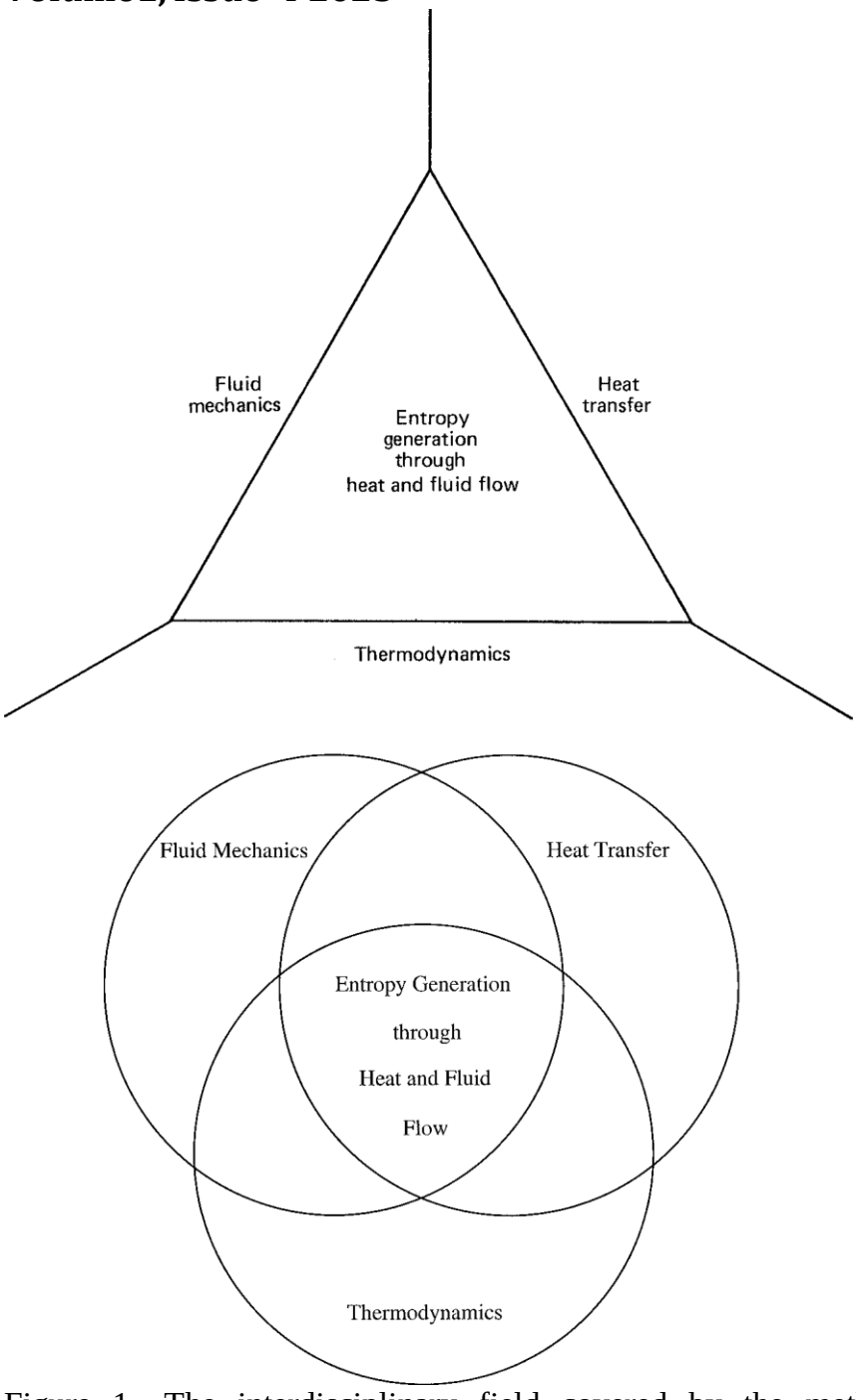


Figure 1. The interdisciplinary field covered by the method of thermodynamic optimization, or entropy generation minimization (Bejan, 1982). The lower diagram is

due to Smith (2000).
Thermoeconomics may be used to make optimization more feasible via further improvements based on global cost reduction. Due to space constraints, this approach cannot be discussed in this article; nevertheless, other articles in this issue provide instances of its use. Benelmir et al. (1991, 1992, 1997), Von Spakovsky (1994), Von Spakovsky and Frangopoulos (1993, 1994), Bejan et al. (1996), Benelmir and Feidt (1997), Tsatsaronis (1999), Valero et al. (1999), and Olsommer et al. (1999a, b).

ENTROPY GENERATION, OR EXERGY DESTRUCTION

This explains why fluid mechanics and heat transport, not only thermodynamics, are essential to thermodynamic optimization. Figure 2 illustrates the most common system-environment arrangement, which is a system operating in an unstable state. M, E, and S are its instantaneous mass, energy, and entropy inventories. Heat transfer rates (QQ °, QQ ¹,⋯, QQ') with n+1 temperature reservoirs (T°, T¹,⋯, T') and mass flow rates (mR in, mR °¹') via any number of intake and outlet ports are experienced by the system. The heat transfer rate between the system and the ambient (atmospheric) temperature reservoir, QQ °, is noteworthy in this range of interactions and will be the subject of our brief discussion.

The thermodynamics of the system consists of accounting for the first law and the second law
(e.g. Moran and Shapiro, 1995; Bejan, 1997),

$$\frac{dE}{dt} = \sum_{k''^{o}} \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \dot{W} + \sum_{in} \dot{m}_{in}\left(h + \frac{V^2}{2} + gz\right) - \sum_{o5'} \dot{m}_{out}\left(h + \frac{V^2}{2} + gz\right) \quad (1)$$

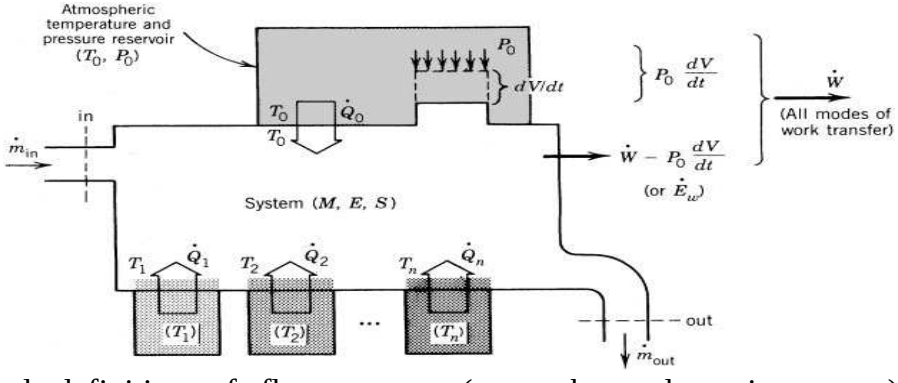


Figure 2. General definition of flow system (open thermodynamic system) in communication with the atmosphere (Bejan, 1988).

where h is shorthand for the sum of specific enthalpy, kinetic energy, and potential energy of a particular stream at the boundary. In Equation (2) the total entropy generation rate $\dot{S}_{gen}$ is simply a definition (notation) for the entire quantity on the left-hand side of the inequality sign. We shall see that it is advantageous to decrease $\dot{S}_{gen}$; this can be accomplished by changing at least one of the quantities (properties, interactions) specified along the system boundary.

We select the environmental heat transfer $\dot{Q}_0$ as the interaction that is allowed to float as $\dot{S}_{gen}$ varies. Historically, this choice was inspired (and justified) by applications to power plants and refrigeration plants, because the rejection of heat to the atmosphere was of little consequence in the overall cost analysis of the design. Eliminating $\dot{Q}_0$ between Equations (1) and (2) we obtain

$$\dot{W} = \frac{d}{dt}(E - T_0 S) + \sum_{k=1}\left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k + \sum_{in} \dot{m}(h - T_0 s) - \sum_{out} \dot{m}(h - T_0 s) - T_0 \dot{S}_{gen} \quad (3)$$

The power output or input in the limit of reversible operation ($\dot{S}_{gen}=0$) is

$$\dot{W}_{rev} = \frac{d}{dt}(E - T_0 S) + \sum_{k=1}\left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k + \sum_{in} \dot{m}(h - T_0 s) - \sum_{out} \dot{m}(h - T_0 s) \quad (4)$$

In engineering thermodynamics each of the terms on the right-hand side of Equation (4) is recognized as an exergy of one type or another (see Section 3), and the calculation of $\dot{W}_{rev}$ is known as exergy analysis. Subtracting Equation (3) from Equation (4) we arrive at the Gouy-Stodola theorem,

$$\dot{W}_{rev} - \dot{W} = T_0 \dot{S}_{gen} \quad (5)$$

In Equation (5) $\dot{W}_{rev}$ is fixed because all the heat and mass flows (other than $\dot{Q}_0$) are fixed.

Pure thermodynamics (e.g., exergy analysis) ends, and the method of entropy generation minimization (EGM) begins with Equation (5). The lost power ($\dot{W}_{lost} = \dot{W}_{rev} - \dot{W}$) is always positive, regardless of whether the system is a power producer (e.g. power plant) or a power user (e.g. refrigeration plant). To minimize lost power when $\dot{W}_{rev}$ is fixed is the same as maximizing power output in a power plant, and minimizing power input in a refrigeration plant. This operation is also equivalent to minimizing the total rate of entropy generation.

The critically new aspect of the EGM method—the aspect that makes the use of thermo- dynamics insufficient, and distinguishes EGM from pure exergy analysis—is the *minimization* of the calculated entropy generation rate. Optimization and design (the generation of structure) are the difference. To minimize the irreversibility of a proposed configuration, the analyst must use the relations between temperature differences and heat transfer rates, and between pressure differences and mass flow rates. The analyst must express the thermodynamic nonideality of the design $\dot{S}_{gen}$ as a function of the topology and physical characteristics of the system, namely, finite dimensions, shapes, materials, finite speeds, and finite-time intervals of operation. For this the analyst must rely on heat transfer and fluid mechanics principles, in addition to thermo- dynamics. Only by varying one or more of the physical characteristics of the system, can the analyst bring the design closer to the operation characterized by minimum entropy generation subject to size and time constraints. We illustrate this technique by means of a few very basic models in Section 4.

EXERGY ANALYSIS

he restricted dead is another name for the environmental condition.Determining and calculating the energies of different things involves a wealth of mathematical tools and terminology. The four categories of words shown on the right side of Equation (4) are associated with the majority of these exergy names. This terminology has to be handled carefully, particularly because the technique is being employed for the first time in regions where energy-based approaches are still the standard. The main characteristic is that, when an entity transitions from a particular state to one of equilibrium with the environment, exergy is the maximum (theoretical) work that may be extracted (or the

lowest effort that is necessary) from the entity (e.g., stream, quantity of matter). Therefore, exergy is a measure of how much a condition deviates from its surroundings; the bigger the deviation, the more work that may be done.
Examine the following examples, where the atmospheric temperature T^o and pressure P^o reflect the environmental condition, to see how exergy is calculated. The entity's exergy (measured in joules) in relation to the environment is equal to $E—E^o—T^o (S—S^o)+P^o (V—V)$ if it is a closed system (fixed mass, Figure 3) in an initial condition represented by the energy E, entropy S, and volume V (6).
The nonflow exergy of the specified mass is denoted by ξ in this formula (Moran, 1989; Bejan, 1988), and the system parameters in the condition of thermal and mechanical equilibrium with the environment are indicated by the subscript 0. T

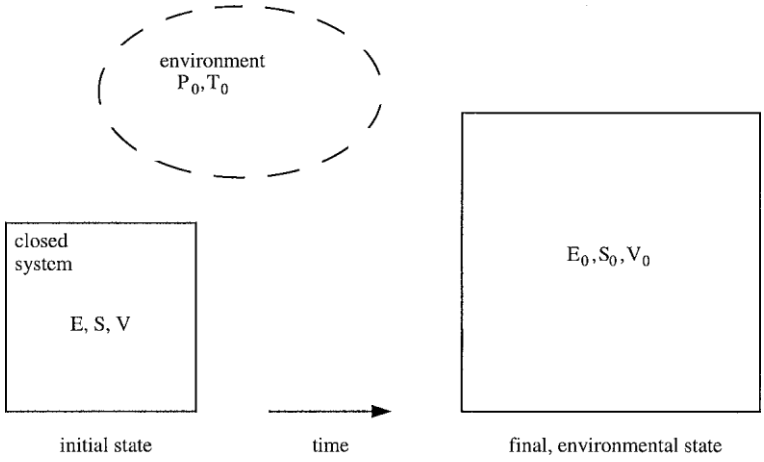


Figure 3. Closed system and process en route to thermal and mechanical equilibrium with the environment.

state—"dead" as the system is unable to provide any more labor in relation to the environment once it reaches this point. It is "restricted" because the system is not in chemical equilibrium but is in mechanical and thermal equilibrium with its surroundings. In that it does not specify the internal structure and materials (e.g., single phase vs. multi phase) of the supplied mass, equation (6) is generic. In situations where the chemical composition of the provided mass may change while reaching chemical equilibrium with the environment, equation (6) may be further extended (Moran, 1989; Bejan, 1988).
It should be noted that the first term seen on the right side of Equation (3) or Equation (4) is the origin of the non-flow exergy ξ. The amount $(E—T^oS)$ is "accumulated" by the system as potential work, and as Figure 3 moves from left to right, this work potential falls from $(E—T^oS)$ to $(E^o—T^oS^o)$. All of the work that may be generated throughout the process is

represented by the difference, which is E—E°—T° (S—S°). The system's work fraction against the atmosphere, P°(V–V), must be subtracted from this number. Equation (6) is the final expression.

Consider the steady-flow system with a stream of mass flow rate mR as an example of a second exergy calculation. The specific enthalpy h, entropy s, kinetic energy ¹V2, and gravitational potential energy gz, where z is the inlet's altitude, reflect the given (inlet) state. The particular flow exergy is measured in relation to the environment (T°, P°) and expressed in J kg¹¹:

e'=(h+¹ V2+gz)—(h+¹ V2+gz) —T (s—s) (7) 2 2 ° ° °

The characteristics of the stream that achieved mechanical and thermal balance with the surroundings are shown by the subscript 0 in Figure 4. To put it another way, e' represents the change in the group h+¹ V2+gz—T s value as it moves from the inlet to the outlet. The product mR e' represents the energy flow rate of this 2° stream. For streams that have the ability to interchange chemical species with the environment, more generic versions of Equation (7) are available (Moran, 1989; Bejan, 1988).

The third and fourth components of Equations (3) and (4), where h was a shorthand for the total (h+¹ V2+gz), already indicated the streams of flow exergy. Keeping with this abbreviated notation, we see that the group (h—T°s) denotes the stream's unique flow availability. The discrepancy between the flow availability in the specified condition (inlet, Figure 4) and the flow

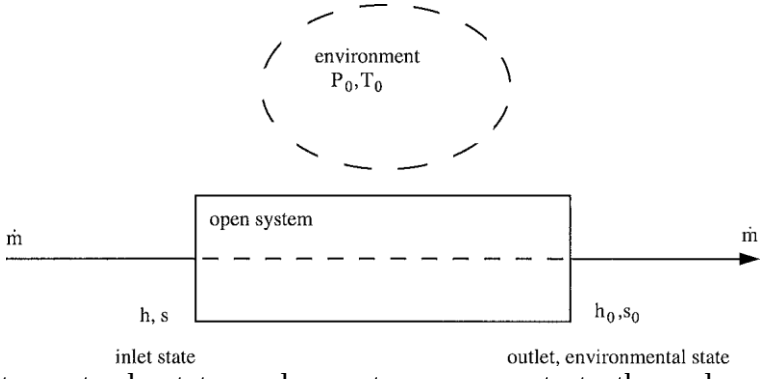


Figure 4. Open system, steady state, and one stream en route to thermal and mechanical equilibrium with the environment.

availability of the same stream at the environmental state (outlet, Figure 4) is the flow exergy, namely $e_o=(h—T_os)—(h_o—T_os_o)$, which is the same as Equation (7).

A third example is the exergy content of heat transfer. If the heat transfer rate $\dot{Q}$ enters the
system by crossing a boundary of local temperature T, then its exergy stream relative to the environment (T_o) is

$$\dot{E}_Q=\dot{Q}\left(1—\frac{T_o}{T}\right) \quad (8)$$

Alternative notations used for $\dot{E}_Q$ are $\dot{E}_A$ and $\dot{E}_{Q,A}$. The heat-transfer exergy flow rate $\dot{E}_Q$ is zero when $T=T_o$, i.e. as in the case of the heat rejected to the ambient by power and refrigeration
plants. In Figure 2 an exergy stream is associated with each of the heat inputs $\dot{Q}_1,\cdots,\dot{Q}_n$, while the
$\dot{Q}_o$ stream carries no exergy $\dot{E}_o=\dot{Q}_o(1—T_o/T_o)=0$. The exergy flows associated with heat
transfer are accounted for by the second term on the right-hand side of Equations (3) and (4).

It is important to stress that in Figure 2 and Equation (8) T_o is the temperature of those regions of the environment that are sufficiently close to the system but not affected by the discharge. The
purpose of this modeling decision is to place inside ‘the system' all the irreversibilities associated with the internal and external effect of the physical installation that resides inside the system. The system comprised the installation and the surrounding regions that are affected (e.g. heated) by the installation. In an actual power or refrigeration plant, the rejected heat current $\dot{Q}_o$ leaves the installation and enters the neighbouring environmental fluid (air, water) at a temperature somewhat higher than T_o. Further down the line, the same $\dot{Q}_o$ reaches the true environment at
T_o (i.e. it crosses the T_o boundary, Figure 2). The interaction between energy systems and their
surrounding fluids forms the subject of the environmental flows documented in several articles in this issue.

Exergy accounting and the spatial distribution of exergy destruction are illustrated in Figure 5. The top figure shows the traditional, energy-flow analysis of a simple Rankine

cycle power plant.
The heat input is QQ , and the net power output is WQ—WQ =η E . The fraction η is the
H ’ P II 9H II
second-law or *rational e*ffi*ciency* of the power plant. It is a relative measure of the combined
imperfections of the power plant. The corresponding exergy wheel for the flows through and around a refrigeration plant is illustrated in Figure 6.
The calculated widths of the exergy destruction streams indicate a ranking of the components as candidates for thermodynamic optimization. The exergy flow analysis (Figures 5 and 6, bottom) can be performed inside each component in order to determine the particular features (e.g. combustion, fouling, heat transfer, pressure drop) that dominate the irreversibility of that component. Finally, the success of the thermodynamic improvements that are implemented can only be evaluated by repeating the exergy analysis and registering the changes in exergy destruction and second-law efficiency.
The current literature reviewed in references shows that exergy principles are being applied to a wide variety of thermal/chemical processes. Avoidable destructions of exergy represent the waste of exergy sources such as oil, natural gas and coal. By devising ways to avoid the destruction of exergy, better use can be made of fuels. Exergy analysis determines the location, type, and true magnitude of the waste of fuel resources, and plays a central role in developing strategies for more effective fuel use.

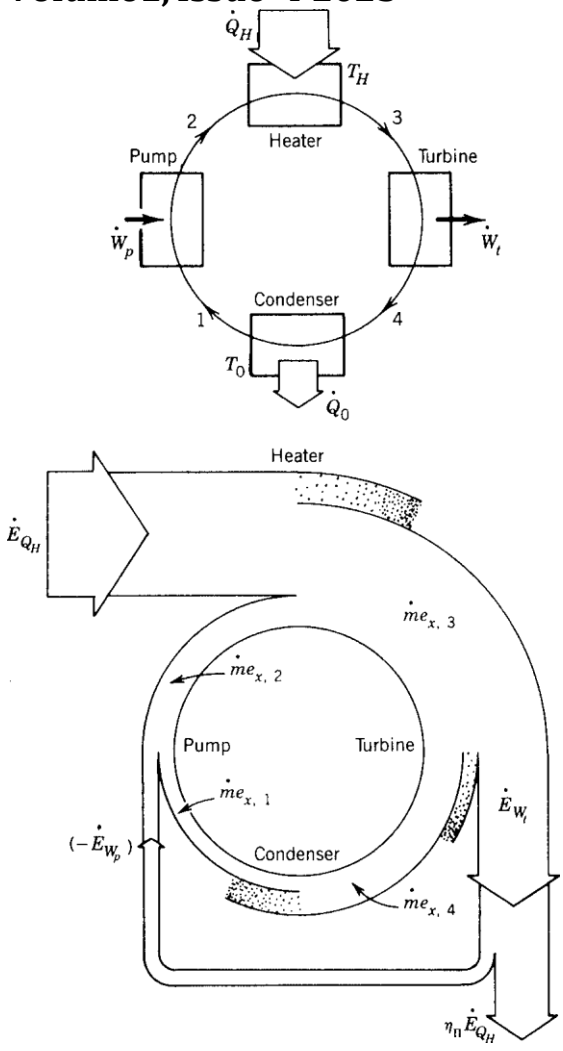


Figure 5. The conversion and partial destruction of exergy in a power plant based on the simple Rankine cycle. Top: the traditional notation and energy-interaction diagram. Bottom: the *exergy wheel* diagram (Bejan, 1988), and the definition of the second law efficiency η_{II}.

ENTROPY GENERATION MINIMIZATION

Developing solutions for the optimum allocation (configuration, topology) of resources is where the majority of the most recent work and advancement prospects are found. This work is referred to as thermodynamic optimization, irreversibility (entropy generation) minimization, or energy destruction minimization. Overall limitations like

limited sizes, finite timeframes, material kinds, and forms are often applied to it. According to recent literature studies (Bejan, 1996a, b), thermodynamic

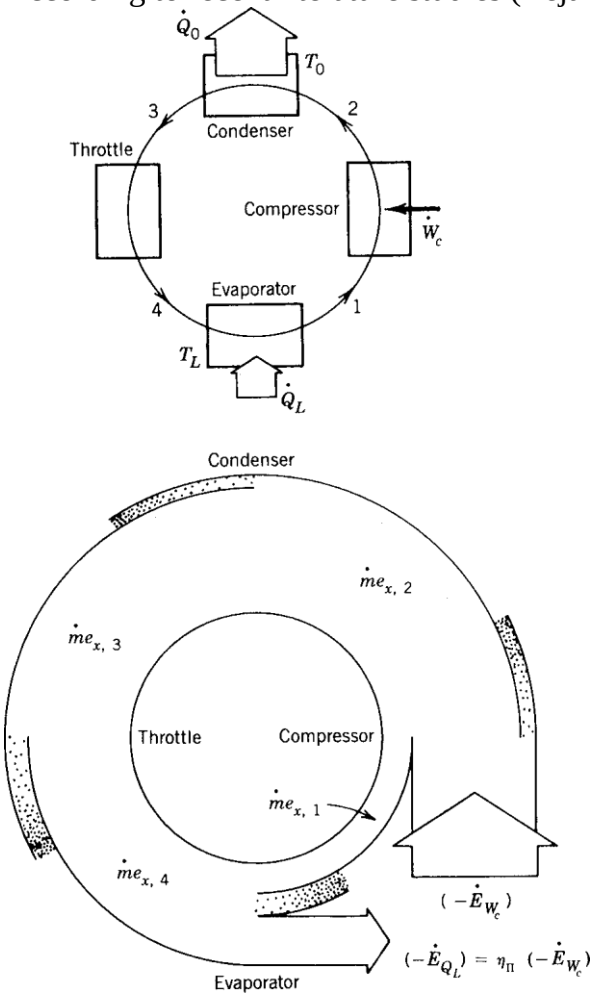


Figure 6. The conversion and partial destruction of exergy in a refrigeration plant based on the simple vapor compression cycle. Top: the traditional notation and energy-interaction diagram. Bottom: the *exergy wheel* diagram (Bejan, 1988), and the definition of the second law efficiency η_{II}.

Cryogenics, heat transfer engineering, energy storage systems, solar power plants, fossil fuel power plants, and refrigeration facilities are all seeing rapid advancements in optimization (Feidt, 1998). The prospects for creating methods of optimum allocation are shown by the examples gathered in this section.

Optimal allocation of heat transfer area

Thermodynamics and heat transport concepts help explain why energy systems are flawed and have geometric structure—that is, why their hardware is placed in certain

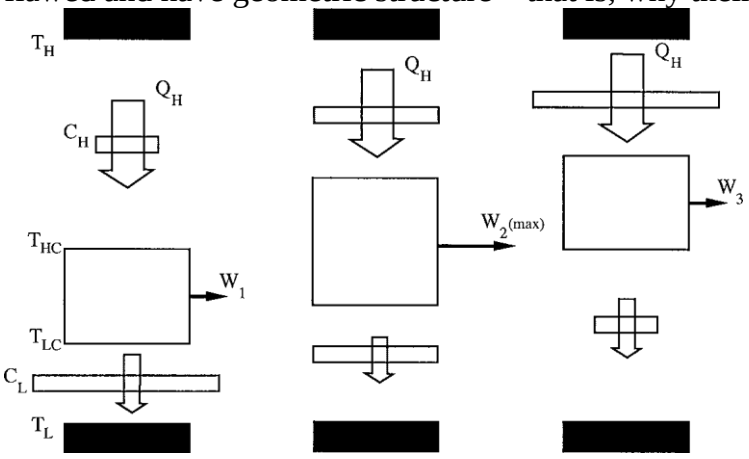


Figure 7. Model of power plant with two heat transfer surfaces, and the maximization of power output (or the minimization of entropy generation) subject to fixed heat input (Q_H) and fixed total heat

transfer surface ($C_H+C_1=C$, constant).

amounts, and in certain ways in space. A power plant owes its irreversibility to many factors, one of which is the transfer of heat across finite temperature differences. This effect has been isolated in Figure 7. The power plant is the vertical segment marked between the high temperature T_H and

the ambient temperature T_1. The heat input Q_H (fixed) and the rejected heat Q_1 must be driven by

temperature differences: the temperature gaps T_H—T_{H^3} and T_{13}—T_1 account for some of the space occupied by the power plant. Heat transfer surfaces reside in these spaces. The rest of the

space is reserved for the rest of the power plant: for simplicity, this inner space is assumed to be irreversibility free (endoreversible),

$$S_{\$^a n} = \frac{Q_1}{T_{13}} - \frac{Q_H}{T_{H^3}} = 0 \qquad (9)$$

The entire irreversibility of this power plant model is concentrated in the spaces occupied by the two temperature gaps. The simplest heat transfer model for these spaces is the

proportionality between heat current and temperature difference,

$Q_H=C_H\ (T_H—T_{H3})$ (10)

$Q_1=C_1(T_{13}—T_1)$ (11)

Each thermal conductance (C_H, C_1) is proportional to its area for heat transfer, e.g. $C_H=U\ A$

and $C_1=U\ A$, where each heat transfer area (A_H , A) is multiplied by its corresponding overall

heat transfer coefficient (U , U). When U and U are equal (or of the same order of magnitude),

H 1 H 1

the total area constraint ($A_H+A_1=A$, constant) is represented adequately by the total conduc- tance constraint (Bejan, 1988, 1997)

$C_H+C_1=C$ (12)

where C is fixed. More general constraints, valid for unequal heat transfer coefficients, can also be used (Bejan, 1996a, b).

The analytical model is completed by the first law, written for the power plant as a closed system operating in steady state or in an integral number of cycles, $W=Q_H-Q_1$. Combining this with the preceding relations, we obtain the power output as a function of the conductance allocation fraction $x=C_H/C$ (in heat exchanger terminology this dimensionless parameter is also known as C^*),

$$\frac{W}{Q_H}=1-\frac{T_1/T_H}{1-(Q/TC)_H\left(\sqrt{x}+\sqrt{1-x}\right)^2} \tag{13}$$

The minimization of entropy generation ($S_{\$^a n}$) is equivalent to the maximization of power output (W), cf. Equation (5). The W expression (13) can be maximized with respect to *x*, and the result is

$x_{oP,}=1/2$ (Bejan, 1988), or

$C_{H,^oP,}=C_{1,^oP,}$ (14)

In conclusion, there is an optimal way to allocate the constrained hardware (*C*) to the two ends of the power plant, that is, if the maximization of power output subject to fixed heat input (Q_H)

and fixed size (*C*) is the purpose. Equation (14) also holds for refrigerating machines modeled in

the same way (Bejan, 1996a, b).

The maximization of W is shown graphically in Figure 7. Small conductances strangle the flow of heat, and demand large temperature differences. The power output is small when the temperature difference across the innermost (reversible) compartment is small. The first and third frames of Figure 7 show that when the two conductances are highly dissimilar in size, large temperature gaps are present, and the power output is small. The best irreversible performance is somewhere in the middle, where the conductances are comparable in size.

Optimal latent-heat storage temperature

This is an illustration of how thermodynamic optimization may be used to choose the kind of energy-storage material. Lim et al. (1992), Figure 8, suggested a straightforward method for carrying out the thermodynamic optimization of the latent heat storage process. A single phase-change material is in contact with a hot stream of starting temperature T◉ via a finite thermal conductance UA, which is assumed to be known. A is the heat transfer area between the melting material and the stream, and U is the total heat transfer coefficient based on A. The solid or liquid phase-change substance is at its melting point (TM). The temperature of the stream that is exhausted into the atmosphere (T^o) is the same as the temperature at which the stream is thoroughly mixed.

The unsteady (cyclic) operation in which each minuscule storage (melting) stroke is followed by a brief energy retrieval stroke—mR is stopped and the recently melted phase-change material is solidified to its original state by the cooling effect provided by the heat engine positioned between TM and T^o—is explained by the steady operation of the installation modeled in Figure 8. In this sense, the whole cycle—that is, storage followed by retrieval—is represented by the steady-state equivalent model shown in Figure 8.

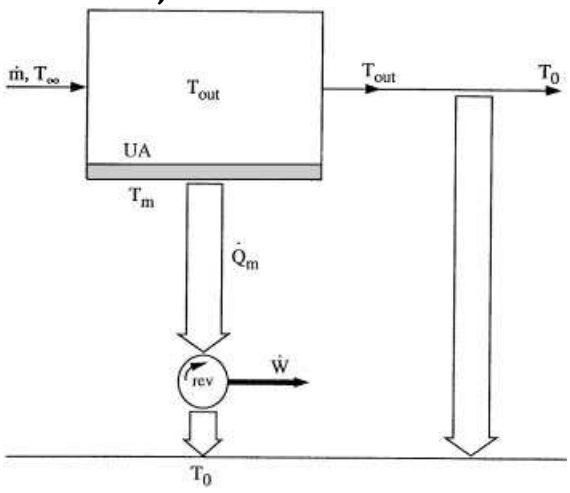


Figure 8. The generation of power using a phase-change material and a hot stream that is ultimately discharged into the ambient (Lim *et al*., 1992).

The steady cooling effect of the power plant can be expressed in two ways:

$$Q_M = UA\,(T_{o5} - T_M) \quad (15)$$

$$Q_M = \dot{m} c_a\,(T_\infty - T_{o5}) \quad (16)$$

By eliminating T_{o5}, between these two equations we obtain

$$\frac{Q_M}{\dot{m} c_a} = \frac{N_{,5}}{1+N_{,5}}\,(T_\infty - T_M) \quad (17)$$

in which $N_{,5}$ is the number of heat transfer units of the heat exchanger surface,

$$N_{,5} = \frac{UA}{\dot{m} c_a} \quad (18)$$

Of interest is the minimization of entropy generation, or the calculation of the maximum rate of exergy (useful work, W_Q in Figure 8) that can be extracted from the phase-change material. For this, we model as reversible the cycle executed by the working fluid between T_M and T_o

$$W_Q = Q_M \left(1 - \frac{T_o}{T_M}\right) \quad (19)$$

and, after combining with Equation (17), we obtain

$$W_{Q} = \dot{m} R_{c} \frac{N}{5} (T - T_{o}) \left(1 - \frac{T_{o}}{T}\right) \quad (20)$$

By maximizing W_{max} with respect to T_M—that is, with respect to the type of phase-change material, we obtain the optimal melting and solidification temperature:

$$T_{M,opt} = (T_H T_L)^{1/2} \quad (21)$$

The maximum power output that corresponds to this optimal choice of phase-change material is

$$W_{max} = \dot{m} c_p \left[(T_H)^{1/2} - (T_L)^{1/2} \right]^2 \quad (22)$$

$M^{a\wedge}$ ^{a} ◉ $1+N,_{5}$ [$(T_{◉})$]

The same results, Equations (21) and (22), could have been obtained by minimizing the total rate of entropy generation. One way to improve the power output of the single-element installation of Figure 8 is by placing the exhaust in contact with one or more phase-change elements of lower temperatures. This method is illustrated in Lim *et al.* (1992).

Optimal solar collector temperature

Two basic issues have been the focus of research on solar energy conversion optimization. One is interested in determining the exergy content of radiation or the theoretical limitations of turning thermal radiation into work. The supply of maximum power from a fixed-size solar collector is the other issue (Bejan, 1982, DeVos, 1992). This issue has also been resolved in several later applications (Bejan, 1996a), all of which share a key design characteristic: the ability to tune the collector operating temperature.

Figure 9 shows this optimization possibility. A solar collector with convective heat loss to the environment powers a power plant. It is assumed that the heat loss is proportional to the temperature differential between the collector and the ambient, $QQ^{o}=(UA)^{3}(T^{3}—T^{o})$. $QQ = (UA)i(T^{3}—T_{4})$ is a similar model for the internal heat exchanger between the collector and the hot end of the power cycle (the user). The collector and power cycle have an ideal connection that maximizes power production. The ideal collector is a representation of this design.

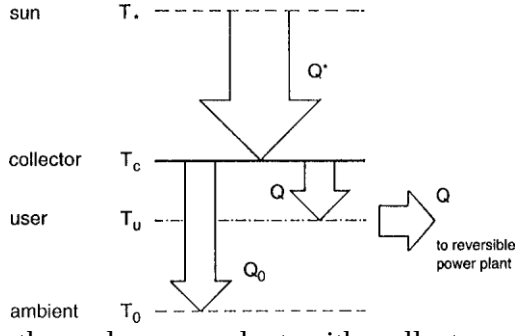


Figure 9. Model of a solar thermal power plant with collector-ambient heat loss and a heat exchanger between the collector and a Carnot energy conversion cycle (Bejan, 1982).

temperature (Bejan, 1982)

$$T_{3,op}, \quad \frac{\theta^{1}/2+R\theta}{\quad} = M^{a\wedge}$$

where $R=(UA)_3/(UA)_i$ and $\theta_{max}=T_{3,max}/T_0$ is the uppermost ('stagnation') temperature of the collector. This optimum has its origin in the trade-off between the Carnot efficiency of the reversible part of the power plant $(1-T_0/T_5)$ and the heat loss to the ambient, Q_0. The power output is the product $Q(1-T_0/T_5)$. When $T_3<T_{3,opt}$, the Carnot factor is too small. When $T_3>T_{3,opt}$, the heat input Q drawn from the collector is too small, because the heat loss to the ambient Q_0 is large.

Corresponding optimal couplings have been identified for solar-driven power plants of many power-cycle designs, extraterrestrial power plants, and refrigeration systems driven by solar power (Bejan, 1996a). These optima are obtained based on the maximization of power output, or the minimization of the total rate of entropy generation.

Optimal sensible-heat storage time interval

The opportunity for minimizing the destruction of exergy during exergy storage becomes evident if we examine the system shown in Figure 10. The storage system (the left side of the figure) contains a batch of liquid (m, c). The liquid is held in an insulated vessel. The hot-gas stream $\dot{m}$ enters the system through one port and is gradually cooled as it flows through a heat exchanger immersed in the liquid bath. The spent gas is discharged directly into the atmosphere. As time passes, the bath temperature T and the gas outlet temperature T_{o5}, approach the hot-gas inlet temperature, T_∞.

If we model the hot gas (steam, products of combustion) as an ideal gas with constant specific heat c_a, the temperature (history) of the storage system is expressed in closed form by the equations

$$\frac{T(t)-T_0}{T_\infty-T_0}=1-\exp(-y\theta) \quad (24)$$

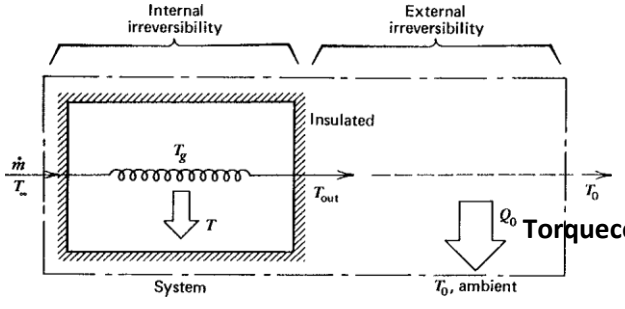


Figure 10. Two sources of irreversibility in the heating (charging) stroke of a sensible-heat storage process (Bejan, 1982).

$$\frac{T_{o5}(t)-T_o}{T_\infty-T_o}=1-y \quad \exp(-y\theta) \qquad (25)$$

where y and the dimensionless time θ are defined as

$$y=1-\exp(-N_5), \quad N_5=\frac{\bar{h}A_a}{\dot{m}c} \qquad \theta=\frac{\dot{m}c}{mc}t \qquad (26)–(28)$$

In these equations, A_a is the total heat-exchanger surface separating the stream from the liquid bath, and $\bar{h}$ is the mean heat transfer coefficient (in time and space) based on A_a. Built into the model is the assumption that the liquid bath is well mixed, i.e., that the liquid temperature (T) is a function of the time (t) only. As expected, both T and T_{o5}, approach T_∞ asymptotically. They approach T_∞ faster when N_5 is higher.

Turning our attention to the irreversibility of the energy-storage process, Figure 10 shows that the irreversibility is due to two distinct parts of the apparatus. First, there is the finite-$\Delta T(t)$ irreversibility associated with the heat transfer between the hot stream and the cold liquid bath. Second, the stream exhausted into the atmosphere is eventually cooled down to T_o, again by heat transfer across a finite $\Delta T'(t)$. Neglected in the present model is the irreversibility due to the pressure drop across the heat exchanger traveled by the stream $\dot{m}$.

The combined effect of the competing irreversibilities noted in Figure 10 is a characteristic of all sensible-heat storage systems. Because of it, only a fraction of the exergy content of the hot stream can be stored in the liquid bath. In order to see this, consider the instantaneous rate of entropy generation in the overall system delineated in Figure 10,

$$\dot{S}_{gen}=\dot{m}c\ln\frac{T_o}{T_\infty}+\frac{\dot{Q}_o}{T_o}-\frac{d}{dt}(mc\ln T) \qquad (29)$$

where $\dot{Q}_o=\dot{m}c_a(T_{o5}-T_o)$. Important is the entropy generated during the entire charging-time interval 0–t, which, using Equations (24)–(29), can be put in dimensionless form as

$$\frac{1}{\tau\eta}\int_0^\theta \dot{S}_{gen}\,d\theta=\ln\frac{T_o}{T_\infty}+\tau+\ln(1+\tau\eta) \qquad (30)$$

$mc \int_o \$^a n$ — ($T_◉$) I I

where the first-law efficiency η_I is shorthand for the right-hand side of Equation (24), and where

$\tau=(T_◉—T_o)/T_o.$,

Multiplied by T_o, the entropy-generation integral $\int_o S\Omega_{\$^a n}$ d*t* calculated above represents the destroyed exergy—the bite taken by irreversibilities out of the total exergy supply brought into

the system by the hot stream

$E , = t E Q , = t m R c_a \ln [T_◉—T_o—T_o \ln (T_◉/T_o)]$ (31)

On this basis, we define the entropy-generation number N_3 as the ratio of the lost exergy divided by the total exergy invested during the time interval 0—*t*:

$$N_3(\theta, \tau , N, _5)=E_o \int, S_{\$^a n} dt=1—\tau\eta_I—\ln \quad (32)$$

$$T \quad (1+\tau\eta_I)$$

$$— \quad \theta [\tau—\ln (1+\tau)]$$

, o

The entropy-generation number takes values in the range 0–1, the N_3=0 limit representing the elusive case of reversible operation. Note the relation N_3=1—η_{II}, where η_{II} is the second-law efficiency of the installation during the charging process.

Charts of the N_3 (θ, τ, N'_5) surface show that N_3 decreases steadily as the heat-exchanger size (N'_5) increases (Bejan, 1982). This effect is expected. Less expected is the fact that N_3 goes through a minimum as the dimensionless time θ increases. For example, the optimal time for minimum

N_3 can be calculated analytically in the limit $\tau \gtrsim 1$, where Equation (32) becomes

$$N'=1—[1—\exp\ (—y\theta)]2/\theta \qquad (33)$$

The solution of the equation $bN_3/b\theta$=0 is

$$\theta_{oP}'=1.256\ [1—\exp\ (—N'_5)]^{-1} \qquad (34)$$

In other words, for the common range of N'_5 values (1–10), the optimal dimensionless charging time is consistently a number of order 1. This conclusion continues to hold as τ takes values

greater than 1.

Away from the optimal charging time (i.e. when $\theta\to0$ or $\theta\to\infty$), the entropy-generation number N_3 approaches unity. In the short-time limit ($\theta \gtrsim \theta_{oP}'$), the entire exergy content of the hot stream is destroyed by heat transfer to the liquid bath, which was initially at

environmental temperature T_o. In the long-time limit ($\theta<\theta_{oP}'$), the external irreversibility takes over: the used stream exits the heat exchanger as hot as it enters ($T_{o5}'=T_\odot$), and its exergy content is destroyed entirely by the heat transfer (or mixing) with the T_o atmosphere. The traditional (first-law) rule of thumb of increasing the time of communication between heat source

and storage material is counterproductive from the point of view of avoiding the destruction of

exergy.

STRUCTURE SPRINGS OUT OF THERMODYNAMIC OPTIMIZATION

The overarching conclusion that unifies the examples discussed in Section 4 is that structure (configuration, topology, geometry, architecture, pattern) is the physical outcome of global optimization of thermodynamic performance.

The examples included time (Section 4.4), temperature (Sections 4.2 and 4.3), and structure in space (Section 4.1).
Further investigation of this structure-generating concept in ever more intricate system configurations is needed. Constructal theory is the idea that the same principle explains the formation of shape and structure in natural flow systems, while the construction of structure in engineering is known as the constructal approach (Bejan, 2000).

The integrative conceptual design of energy flow systems for aircraft is especially pertinent to the idea of structuring structures to collect and use maximum energy from a hot stream. The same idea holds true for systems like ships, cars, military vehicles, environmental control suits, portable power tools, etc., where every function is powered by the energy extracted from the restricted fuel supplied on board. The record on powered flight, both artificial and natural, lends further credence to this viewpoint. The cruising speeds of insects, birds, and aircraft are shown in Figure 11 alongside the potential speed attained by limiting the power (exergy rate) lost during flying (Bejan, 2000). The aircraft's speeds

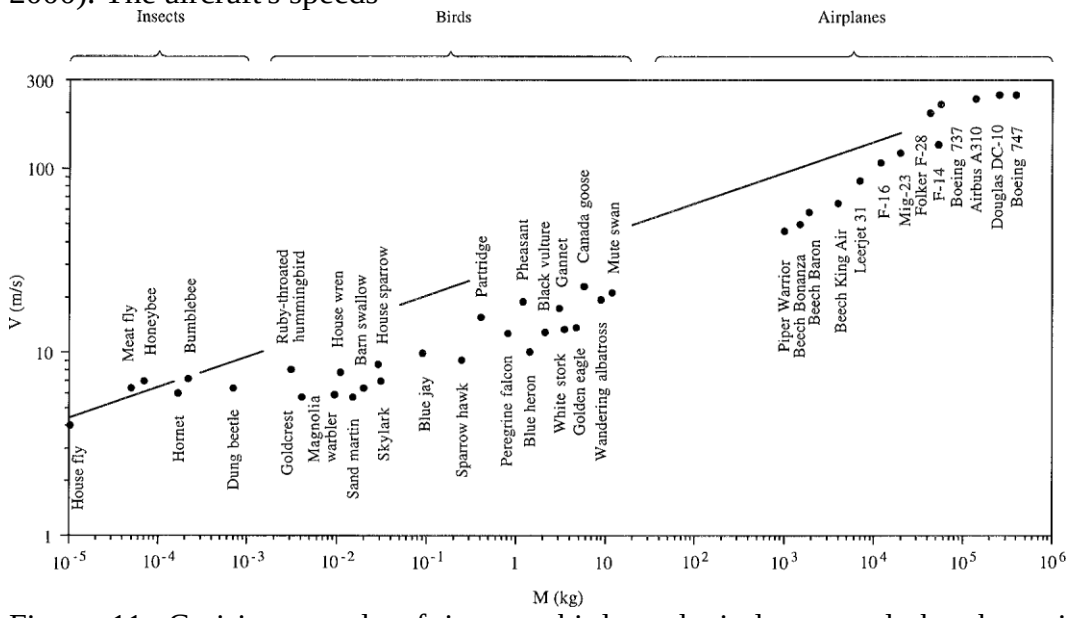


Figure 11. Cruising speeds of insects, birds and airplanes, and the theoretical speed (V&M^{1}/') for minimum rate of exergy destruction (Bejan, 2000).
Shown in this figure do not represent the aircraft's maximum speed, but rather the ideal cruising conditions—the speed for the least amount of power usage.
The natural and constructed flow systems' performance records (e.g., Figure 11) indicate that the constructal principle is significant not only in engineering but also in physics and biology in general. According to this theory, just as the body of a flying animal (such as a bat or bird) acquired its own well-suited extensions, the aircraft arises as a physical extension of man.

These extensions are all distinct points on a continuous time axis that indicate the more complicated and better. A recent book (Bejan, 2000) examines this theoretical area of investigation.

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