Isothermal Combustion for Improved Performance in Air-Breathing Engines

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This theoretical effort proposes and explains in detail the concept of isothermal combustion. The concept involves combustion at constant total temperatures in order to realize isothermal heat addition to the working fluid in typical powerplants. It is rendered practical by extracting a precise amount of work from the flowing/expanding gases; the heat addition due to combustion will be balanced out to keep the total temperature constant. In an oversimplified description, it might be said that this involves burning in the turbine stages, or burning during the expansion stroke. Isothermal heat addition enables the thermodynamic cycle to approach the Carnot cycle more closely than the state-of-the-art Brayton, Otto, or diesel cycles. A closed-form analytical expression is derived to explicitly show the cycle efficiency in terms of the pressure ratio and the overall temperature ratio. Thirty to forty percent efficiency increases are seen over the Brayton efficiency for the same overall temperature ratio. Practical aspects are qualitatively mentioned. Future prospects are outlined.

I. Introduction

THERE are strong reasons for gaining fuel efficiencies in aeropropulsion. Terrestrial gas turbine powerplants also need to be efficient, and in these nonaerospace applications, many of the restrictions are relaxed regarding the space and area limitations.

A simple analysis of the highly popular and successful Brayton cycle powerplants indicates the crucial role played by the cycle pressure ratio. The higher the pressure ratio, the higher the efficiency. What is not immediately apparent is the important effect of the maximum cycle (turbine inlet) temperature. It is a simple exercise to show that there is an optimum pressure ratio that maximizes the net work output for a fixed turbine inlet (highest) temperature. A fairly detailed discussion[1] of these facts shows the gradual evolution of the combustion temperature rise and the decreasing thrust-specific fuel consumption. The drive toward efficiency increase has projected a turbine inlet temperature of 3000 + °F in trying to reach the stoichiometric limit of 4000°F for JP4 fuel with air. State-of-the-art temperatures are in the 2400–2800°F range accompanied by a thrust specific fuel consumption (TSFC) of 0.6-lb/$lb_f \cdot$h. The E^3 technology objective was to achieve, by 1990, 0.54 lb/$lb_f \cdot$h for the TSFC, which is still far removed from what the stoichiometrically limited temperature values promise.

Two facts seem obvious. State-of-the-art and immediate projections show efficiencies substantially less than the ultimate; the progress toward the goal has been slow, evolutionary, and strongly dependent on high-temperature material technologies. It would seem worthwhile to inquire if there exists some new technology that may be able to deliver vast improvements without depending on advances in critical materials. In addition to the efficiency advantages, such approaches would also minimize the risks associated with advanced materials. (Such risks always accompany new materials until a large number of qualification hours accumulate sufficient service life data.)

This paper presents a concept for improved performance with state-of-the-art materials use; the concept also indicates a way of actually decreasing our demands on the materials' temperature capabilities while realizing superior cycle performance. The concept uses a cycle that approaches the Carnot cycle. The fundamental attempt is to burn at constant total temperature. This can be accomplished by extracting work while liberating combustion energy, or burning in the turbine stage(s). The concept may be said to be similar to the interburn (reheat) cycle, where one also attempts to approximate isothermal expansion through a series of staged heating and work extraction. However, the present concept is fundmentally different in that work is continuously extracted during heat addition and not in series. Thermodynamically, the two cycles may look similar on the T-s diagram, but they are inherently different in the realization of the isothermal heat addition. The concept of isothermal expansion with heat addition is not new. In addition to the famous Carnot cycle, at least two other thermodynamic cycles with isothermal heat addition have been treated in detail in the literature[2]: these are the Stirling cycle and the Ericsson cycle. Their realization appears to have involved external combustion, or heat addition. This paper presents an internal combustion cycle, which minimizes many complications besides rendering the power plant compact. Also, the author is not aware of any other

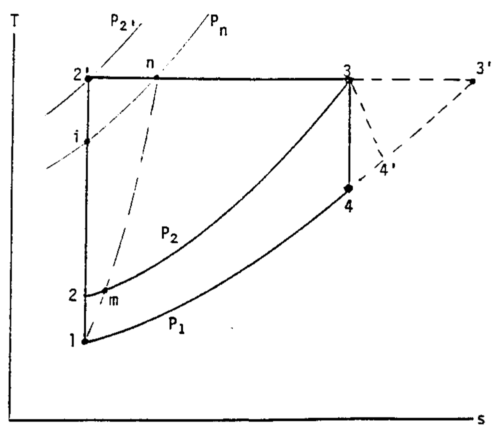

Fig. 1 Cycle(s) on the T-s diagram.

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work treating an isothermal heat addition cycle in the specific context of gas turbine power plants (see Blazowski[3] and Zukoski[4]). Several of the cycle details are somewhat unique to gas turbine and jet propulsion engines. These involve some of the performance optimizations, the concept of isentropic (or polytropic, or adiabatic) component efficiencies, and the maximum temperature in the cycle.

Several powerful new technologies involving control of the "nature-prescribed" combustion time scales are available with the use of free radicals.[5-7] Hence, the distinct possibility exists that we may be able to tailor the combustion time scales to suit the turbine flow needs. If the new engine is viewed as a modification of the state-of-the-art gas turbines, numerous problems present themselves; these involve losing many of the carefully optimized performance values, blade shapes, annulus shapes, cooling passages, and related variables that have seen literally millions of hours of evolution. Alternatively, if the new turbine is recognized as an inherently different machine, many of these problems disappear and a whole new design challenge confronts the creative engineer.

II. Cycle Analyses

This section considers the thermodynamic analysis of the cycle performance. It is compared with the Brayton cycle at every stage. Referring to Fig. 1, 1-2-3-4 represents the ideal Bryton cycle operating between the pressures P_1 and P_2, with the standard nomenclature

$$\pi_c \equiv (P_2/P_1) = (P_3/P_4)$$

and

$$\tau_c \equiv (T_2/T_1), \qquad \tau_\lambda \equiv (T_3/T_1), \qquad \pi_N \equiv (P_{2'}/P_1)$$

With the isentropic relation between pressure and temperature ratios, it is easily shown that the Brayton cycle efficiency

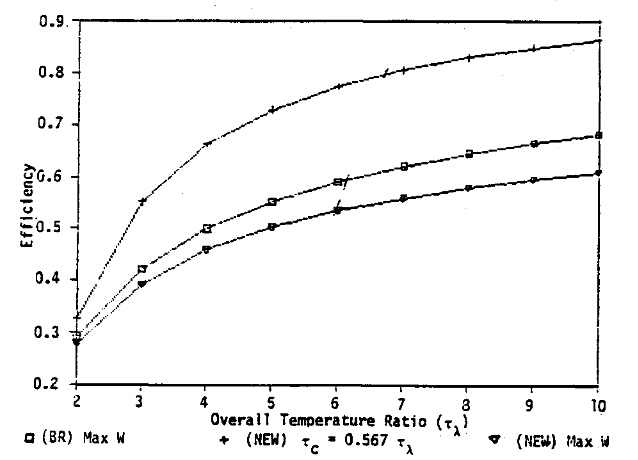

Fig. 2 Ideal cycle efficiencies.

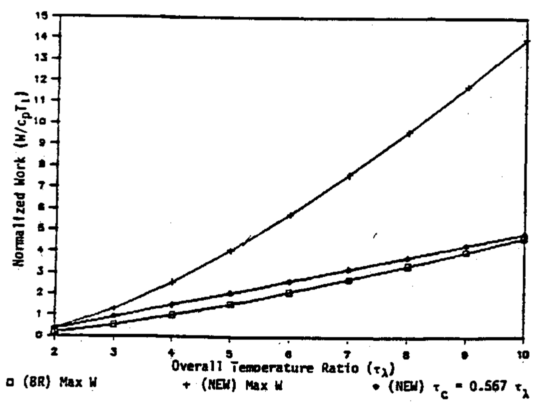

Fig. 3 Ideal cycle net work output.

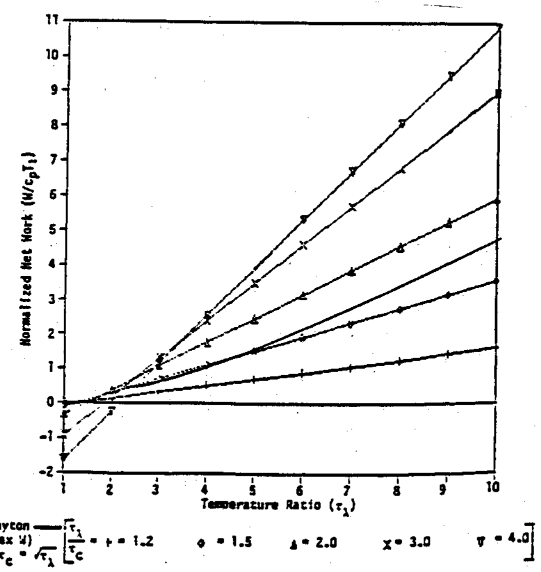

Fig. 4 Ideal cycle(s) net work for various (τ_λ/τ_c).

is

$$\eta_B = 1 - (1/\tau_c) = 1 - \frac{1}{\pi_c^{(\gamma - 1)/\gamma}} \tag{1}$$

where $\gamma = c_p/c_v$ has been assumed to be constant. The new cycle, represented by 1-2′-3-4, has the efficiency

$$\eta_N = 1 - \frac{(1/\tau_c) - (1/\tau_\lambda)}{\ell n(\pi_N/\pi_c)^{(\gamma-1)/\gamma}}$$

$$\eta_N = 1 - \frac{(1/\tau_c) - (1/\tau_\lambda)}{\ell n(\tau_\lambda/\tau_c)} \tag{2}$$

Equation (2) tends to Eq. (1), the Brayton cycle efficiency, in the limit of $\tau_\lambda \to \tau_c$ (use of L'Hospital rule).

Before we plot the results to compare the two cycles, some meaningful bases of comparison must be established. The Brayton cycle efficiency shows no maxima or minima with π_c (i.e., τ_c). However, the net work output can be shown to be a maximum at $\tau_c = \sqrt{\tau_\lambda}$. Hence, the maximized work and the cycle efficiency at this maximum work have been plotted in Figs. 2 and 3.

Simple differentiation of Eq. (2) indicates that there is a maximum in the efficiency for fixed τ_λ; however, this leads to zero work because the maximum occurs when $\tau_c = \tau_\lambda$. Alternatively, $\tau_c = 1$ leads to maximum net work, which is also obvious from the fact that this condition leads to maximum area on the T-s diagram; namely, 1-2′-3′-1 in Fig. 1. For the purposes of numerical computations, a reasonable value for τ_c was arbitrarily chosen as 0.567 τ_λ. It would also be useful to have the variation of η and W with τ_λ/τ_c, and these are plotted as Figs. 4 and 5. It is easily shown that the ideal Brayton cycle net work has the maximum value

$$(W/c_pT_1) = (\tau_\lambda^{1/2} - 1)^2$$

and the corresponding efficiency is

$$\eta = 1 - \tau_\lambda^{-1/2}$$

with the compressor pressure ratio

$$\pi = \tau_\lambda^{1.75} \qquad \text{(recall that } \tau_c = \sqrt{\tau_\lambda}\text{)}$$

The new cycle, similarly, can be shown to have the following values:

$$\left.\begin{aligned} \frac{W}{c_pT_1} &= \tau\ \ell n\tau_\lambda - (\tau_\lambda - 1) \\ \eta &= 1 - \frac{\lambda_\lambda - 1}{\tau_\lambda\ \ell n\tau_\lambda} \end{aligned}\right\} \begin{array}{l}\text{at maximum work} \\ \text{i.e., } \tau_c = 1\end{array}$$

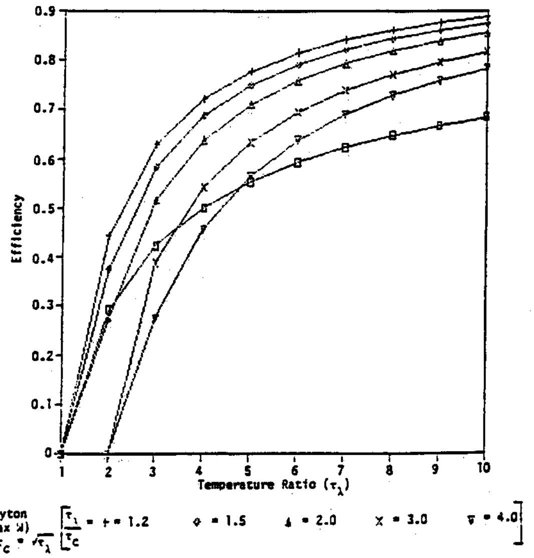

Fig. 5 Ideal cycle(s) efficiency for various (τ_λ/τ_c).

and Eq. (2) at $\tau_c = 0.567\ \tau_\lambda$ becomes

$$\eta = 1 - (1.3457/\tau_\lambda)$$

along with

$$(W/c_pT_1) = 0.567\ \tau_\lambda - 0.763$$

For most practical cases, $(\tau_\lambda/\tau_c) \cong$ 2-3 or

$$\ell n \left(\frac{\tau_\lambda}{\tau_c}\right) \sim 1$$

Hence,

$$\eta_N \sim 1 - \left(1 - \eta_B - \frac{1}{\tau_\lambda}\right) \sim \eta_B + \frac{1}{\tau_\lambda}$$

This shows that the advantages of the new cycle are more pronounced at the lower τ_λ values. Viewed another way, η_B for a higher τ_λ can be matched by η_N with a lower τ_λ with obvious implications to materials.

A. Nonideal Processes

As a first step, conventional analyses with nonideal compressor and turbine performances are shown. The compressor is considered to have an isentropic efficiency η_c, defined as the ratio of isentropic (ideal) work to actual work, while compressing the working fluid from T_1 to $T_{2'}$ (new cycle point n) and P_1 and P_2 (Brayton cycle point m). Similarly, the expansion from 3 to 4′ represents the nonideal process in both the new and Brayton cycle with that turbine efficiency η_t. Nonideal processes in going from 2 (or 2′) to 3 are deferred to a later study.

1. Brayton Cycle [1-m-3-4′-1 on Fig. 1]

The net work and the efficiency are readily shown to be

$$\left(\frac{W}{c_pT_1}\right)_B = \left[\pi_c^{(\gamma-1)/\gamma} - 1\right]\left[\tau_\lambda \frac{\eta_t}{\pi_c^{(\gamma-1)/\gamma}} - \frac{1}{\eta_c}\right] \tag{3}$$

$$\eta_B = \frac{\pi^{(\gamma-1)/\gamma} - 1}{\tau_\lambda} \frac{\tau_\lambda \dfrac{\eta_t}{\gamma^{(\gamma-1)/\gamma}} - \dfrac{1}{\eta_c}}{\left\{1 - \dfrac{1}{\tau_\lambda}\left[\dfrac{\pi^{(\gamma-1)/\gamma} - 1}{\eta_c} + 1\right]\right\}} \tag{4}$$

It should be remembered that the isentropic simplification $\pi_c^{(\gamma-1)/\gamma} = \tau_c$ is no longer applicable.

2. New Cycle [1-n-3-4′-1 on Fig. 1]

The main points to remember are 1) the compression process does not have to reach $P_{2'}$ from P_1 but will reach the maximum temperature $T_{2'} = T_n$ at a lower pressure P_n and 2) the isentropic efficiency is defined as

$$\eta_c \equiv (T_i - T_1)/(T_n - T_1)$$

Recognizing that

$$T_i = T_1[\eta_c(\tau_\lambda - 1) + 1] \tag{5}$$

we may write

$$\eta_N = 1 + \frac{\eta_t\left(1 - \dfrac{1}{\tau_c}\right) - \left(1 - \dfrac{1}{\tau_\lambda}\right)}{\ell n\ \{[\eta_c(\tau_\lambda - 1) + 1]/\tau_c\}} \tag{6}$$

By putting $\eta_t = \eta_c = 1$ in Eq. (6), we satisfy ourselves that we recover the earlier result in Eq. (2) for the ideal cycle.

Similarly, the net work of the new cycle is

$$\therefore\ \frac{W}{c_pT_1} = \tau_\lambda\left[\ell n(\pi_N/\pi_c)^{(\gamma-1)/\gamma} + \eta_t\left(1 - \frac{1}{\tau_c}\right) - \left(1 - \frac{1}{\tau_\lambda}\right)\right] \tag{7}$$

At first glance, it may appear surprising that the real cycle new work does not have η_c anywhere. It is, however, a matter of simplified algebra that we have chosen to deal with the real temperature ratio $\tau_\lambda = T_n/T_1$ across the compressor instead of the ideal temperature rise T_i/T_1 and the attendant η_c.

B. The Compressor Pressure Ratio

The real compressor pressure ratio is lower than the ideal for a constant τ_λ. The relation is easily derived from Fig. 1:

$$\eta_c \equiv \frac{T_i - T_1}{T_n - T_1} = \frac{(T_i/T_1) - 1}{(T_n/T_1) - 1} = \frac{\pi_{\text{actual}}^{(\gamma-1)/\gamma} - 1}{\pi_{\text{ideal}}^{(\gamma-1)/\gamma} - 1}$$

or

$$\pi_{\text{actual}} = [\eta_c(\tau_\lambda - 1) + 1]^{\gamma/(\gamma-1)} \tag{8}$$

All of the above relations are plotted in Figs. 6–11 for representative values of the parameters.

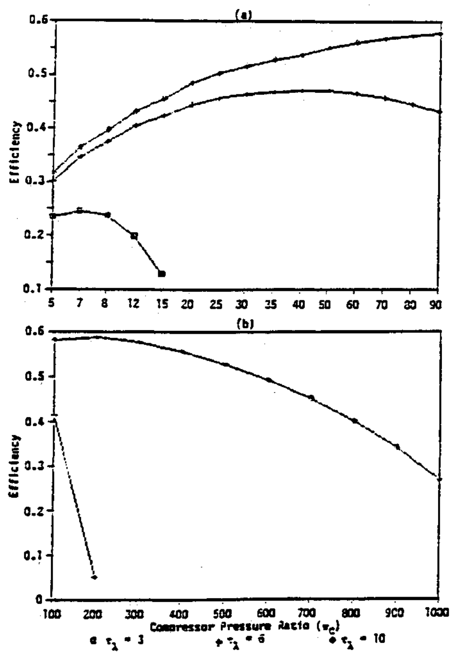

Fig. 6 Brayton cycle efficiency (nonideal).

The extensive performance charts prepared for representative values of the parameters show the anticipated superiority of the new cycle over the state-of-the-art Brayton cycle. This conclusion holds both for the ideal and real cycles. It is especially important that both the efficiency and the net work can be improved at the same time, and one is not obtained at the expense of the other. To have a direct comparison of the advantages, a cross-plot is presented in Fig. 12; value of the τ_c (secondary turbine temperature ratio) was chosen as 1.5 (other values can also be used but would diminish the clarity of the plots), the compressor adiabatic efficiency was taken as 0.85, and the turbine adiabatic efficiency was assumed to be 0.9. A constant value of 1.4 was chosen for the ratio of specific heats,

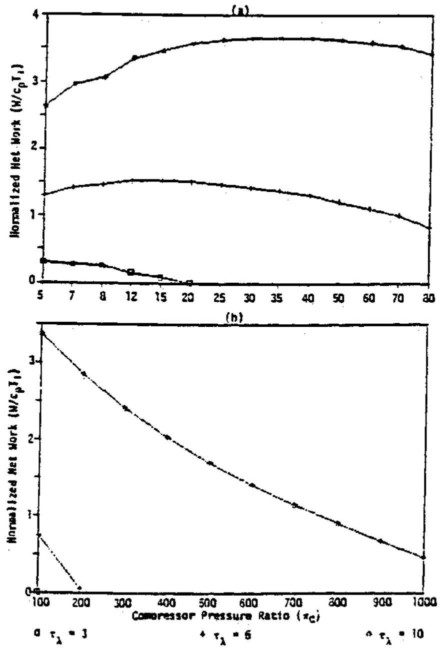

Fig. 7 Net work from nonideal Brayton cycle.

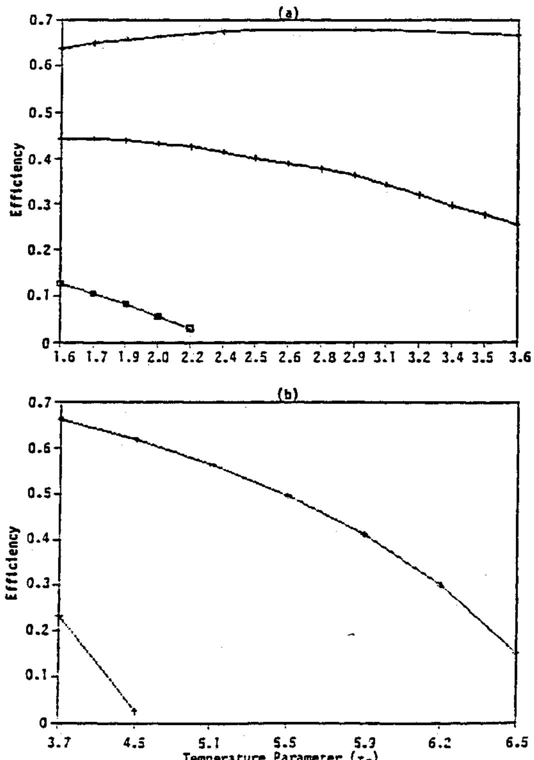

Fig. 8 New cycle efficiency (nonideal).

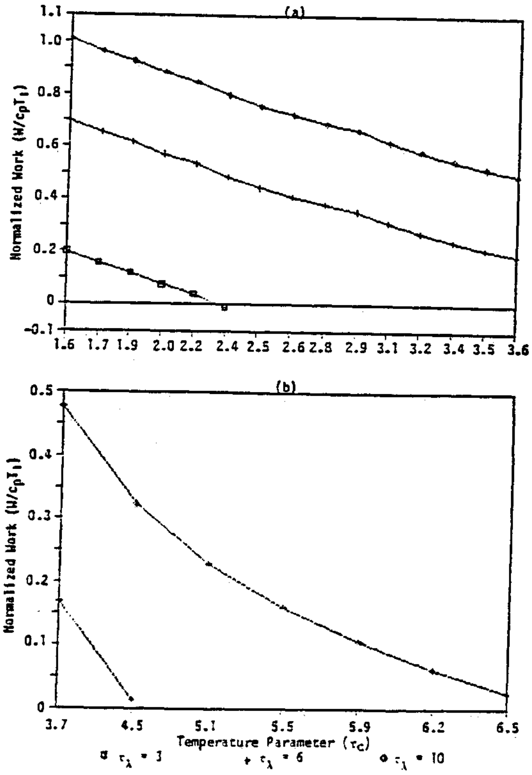

Fig. 9 New cycle net work (nonideal).

γ. Other values can be used, but the point should be clear. From the performance data, efficiency and network for the nonideal Brayton cycle at various values of the overall temperature ratio τ_λ are plotted. The compressor pressure ratio yielding these values is also noted in Table 1. For the nonideal new cycle, the efficiency and net work are read for $\tau_c = 1.5$. In the case of the new cycle, the overall temperature ratio is directly related to the compressor pressure ratio, as shown earlier. It is clear that the advantages are sufficiently substantial to provide a strong motivation for actually realizing the isothermal heat addition in the turbine stages. The physical (and chemical) factors are discussed in the next section.

III. Mechanics of Isothermal Combustion

Working solutions to the problem of isothermal combustion must await further theoretical and experimental research. Here, it is the aim to explore if there exist any fundamental reasons why such a process could not be realizable. It is also the aim to see if there are any obvious faults in the general idea. Characteristic time scales associated with the component (sub)processes are estimated from the values in the literature. It is shown that the general idea seems to be technically feasible.

Combustion heat libration involves fuel injection, atomization, evaporation, macro- and micromixing within the ignition/flammability limits, ignition delay, ignition, and conversion to products. If work is extracted simultaneously, it must be done at the requisite time step to prevent unintended temperature variations. Simple gasdynamic considerations suggest that the initial Mach number be low in order to minimize total pressure losses. With a Mach number of 0.15 at compressor exit, the flow velocity will be approximately 400 to 500 ft/s for the new cycle, with a τ_λ value around 6. This leads to characteristic residence times of 2 ms, with a characteristic length scale of 1 ft. This number may seem rather arbitrary at this time, but it provides us with a reasonable working scale for the rest of the discussions.

The characteristic times associated with evaporation, ignition, and initial combustion (energy release) are estimated from data in the literature (primarily from Refs. 3 and 4). The

Table 1 The compressor pressure ratio requirements ($\eta_c = 0.85$, $\eta_t = 0.9$, $\gamma = 1.4$)

	Brayton		New
	Max η	Max W	$\tau_c = 1.5$
$\tau_\lambda = 3$	$\pi_c = 7$	$\pi_c = 5$	$\pi_N = 32$
6	35	12	332
10	200	35	1904

initial conditions, i.e., at the exit from the compressor, are assumed to be T = 2760 R (which results from a τ_λ of 6 and a compressor inlet temperature of 460 R; these are representative of a flight Mach number of 0.86 at an altitude where the ambient conditions are 400 R and 2 psia pressure) and 663 psia (for the new cycle, with a compressor efficiency of 0.85, the $\pi_N = 332$ for a $\tau_\lambda = 6$). The self-ignition temperature (SIT) is 470 K or 846 R for JP-4 fuel. Hence, no external ignition sources are needed. The evaporation time is shown to be $t_e = r_0^2/(\rho_v \mathfrak{D}/\rho_\ell)$. The numerical value for a 20-μ-rad droplet is 1.5×10^{-4} s. The combustion time is estimated from the expression

$$-\frac{\mathrm{d(fuel)}}{\mathrm{d}t} = \frac{5.52 \times 10^8 T}{p^{0.825}} \mathrm{(fuel)}^{1/2}\,\mathrm{(O)}^2 \exp(-24{,}400/RT)$$

and gives approximately 10^{-7} s. This is only the time for pyrolysis and partial oxidation, and the full combustion time will be longer. Simple flame speed expressions taken with the characteristic flame thickness for the length scales suggest a combustion time scale of 10^{-4} s. Considering that these individual time scales are cumulative, the overall characteristic time scale for heat release is estimated to be of the order of 10^{-3} s.

So far, we have considered classical combustion mechanisms and time scales. A new technology that promises considerable advantages is the technology of free-radicals influenced combustion (and ignition). It has been experimentally demonstrated[5,6] that classical flammability limits and ignition energies can be considerably altered by injecting free-radicals into the reaction zone. The complications of these injection schemes (free-radicals are extremely short-lived and reactive) were greatly reduced by the concept of tailored free-radicals release in situ.[7] Fifty percent extension of the lean limit was demonstrated in a methane/enriched-air turbulent flame by premixing the reactants with a small (0.6%) quantity of a donor that decomposed in the reaction zone, thereby releasing a burst of free-radicals precisely where they were needed. Other possibilities exist, along similar lines, for increasing the reaction times (slowing the reactions). The main point to note is that the characteristic combustion times need no longer be taken as nature prescribed; there exist modern technologies to tailor these time scales to suit our needs. It is thus concluded that the flow times and combustion times are of the same general magnitude, which renders the concept of isothermal combustion technically realistic.

A. Mechanical Realization

There are several variations in the mechanical design for achieving the injection, mixing, and combustion. The fuel can be injected as a liquid at the inlet to the turbine stage(s). The fuel can be injected as a liquid, or a vapor, through perforations in the first stator stage. The injection can also be through perforations in the rotor blades, which also will serve to cool the blades. The details of fluid dynamics and combustion in the roto machinery will have to be worked out to arrive at the optimal arrangement. It is important to note that the geometry of the isothermal combustion turbines may be substantially different from the state-of-the-art turbines. This removes the usual skepticisms expressed about the possible success of injecting fuel in the turbine stages.

B. Blade Cooling

If the new cycle is run at a low value of the maximum temperature, the cooling problems are inherently less severe; recall that the new cycle can deliver efficiencies and work higher than those of the Brayton cycle even with the substantially lower turbine inlet temperatures. If the new cycle is run with high (>3000°F) temperatures, the blade cooling needs careful study. In the conventional Brayton cycle, larger areas with gas expansion are also accompanied by lower temperatures. Thus, the larger blade areas can be cooled more easily.

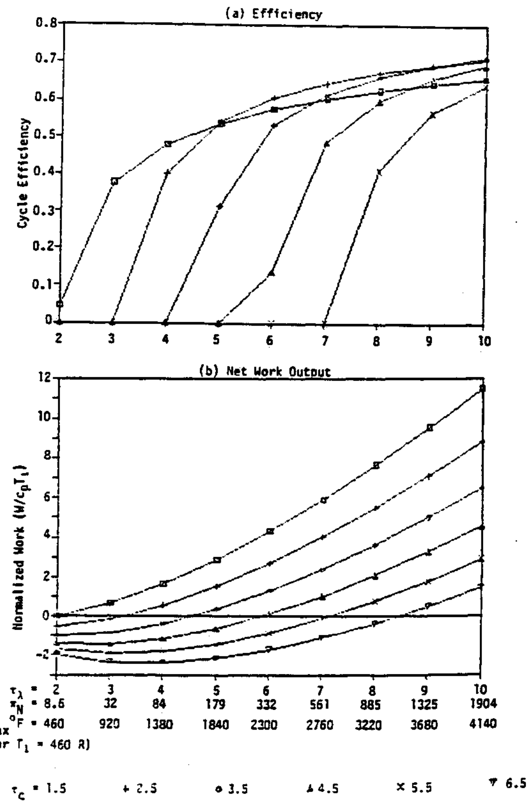

Fig. 10 a) Efficiency of and b) new output from nonideal new cycle (compressor efficiency = 0.85; turbine efficiency = 0.9; $\gamma = 1.4$).

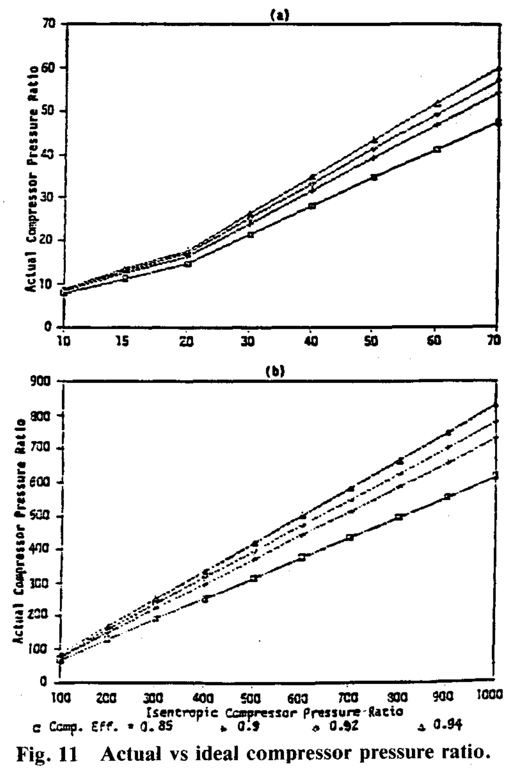

Fig. 11 Actual vs ideal compressor pressure ratio.

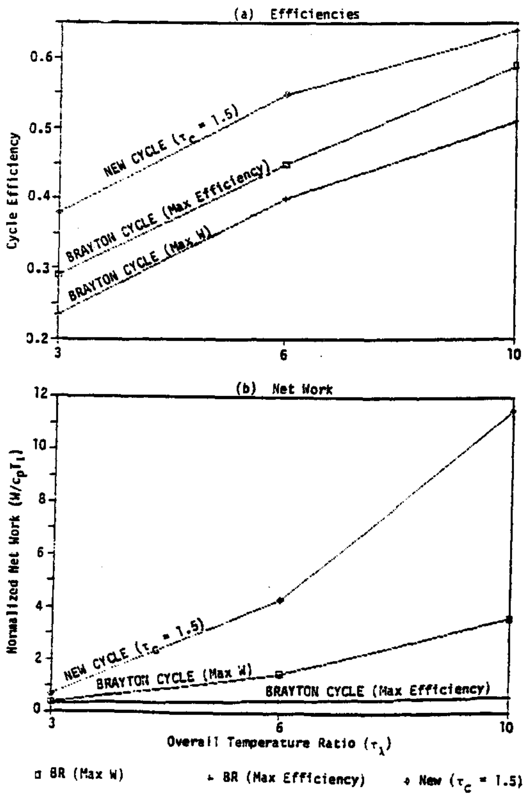

Fig. 12 a) Efficiencies of and b) net work output from nonideal cycles.

In the new cycle, the larger areas continue to see the same high temperature. Hence, the natural advantages are no longer present. This may not be a very severe problem in stationary power plants that have fairly loose restrictions on the geometrical parameters (e.g., hub-to-tip ratios).

C. Reciprocating Engines

The isothermal combustion cycle is not limited to the gas turbines. In fuel injection engines, the injection rate can be programmed to match the cylinder free volume in order to realize constant temperatures while the piston is moving. Thus, work can be extracted while heat is being liberated in the cylinder due to combustion. With modern control technologies and microcircuitry, along with advances in fuel injectors, it is felt that isothermal combustion is applicable to reciprocating engines, too. Diesels, with their inherently high compression ratios, should be the first candidates to test the idea.

IV. Summary

This paper considered the theoretical development of isothermal combustion in internal combustion engines; the gas turbine engine was treated in some detail. Substantial advantages were shown in the cycle efficiency and the net work from gas turbine engines over the conventional Brayton cycle engines. The conclusion holds both for the ideal cycle and real cycles.

The advantages appear sufficiently important to pursue the idea further. A simple table top turbine stage, which can extract work isothermally, could prove the practicality of the method. It is hoped that the work can be continued.

Acknowledgments

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