A NEW LOOK AT THE CLOSED BRAYTON CYCLE

F. David Doty & J. Dewey Jones

Doty Scientific, Incorporated & Simon Fraser University

The closed Brayton cycle (or 'CBC') has long been known, but has been considered impractical for most applications because of its requirement for massive and expensive heat exchangers. A novel heat-exchanger design – the microtube-strip heat exchanger – largely overcomes this problem, making it possible to take advantage of several attractive features of the cycle. Its use of external combustion, for example, leads to low emissions of hydrocarbons and NO_x, low noise and multi-fuel capability.

Design guidelines are developed for an engine designed around the closed recuperative Brayton cycle, without reheat or intercooling. Analytic expressions are derived for the effects of various design parameters on system efficiency and specific power. Taking these expressions together with considerations of manufacturability and reliability, it can be demonstrated that such an engine combines full-load efficiencies comparing favorably with current power plants with good part-load efficiency. However, several key issues must be resolved for the engine to realize these advantages. These include the design of gas bearings and suitable generators.

The Low Pressure Ratio (LPR) Closed Brayton Cycle (CBC)

The Brayton (or Joule) cycle is a single-phase thermodynamic cycle that includes the following sequence of processes: isentropic compression, isobaric heating, isentropic expansion and isobaric cooling. There are two major classes of Brayton cycles: open and closed. Historically, more attention has been given to the open cycle, because of the high heat-exchange requirements of the closed cycle. The first closed-cycle engine was designed by Ackeret and Keller at Escher Wyss in 1939, using air as a working fluid. This company has continued to work on closed-cycle engines for extracting power from coal and nuclear plants; helium is now more commonly used as a working fluid. Interest in the closed cycle elsewhere has been limited, though it is now starting to be considered as a candidate for space power and nuclear power [1,2].

The particular cycle described here has been referred to as the CBEX cycle: the closed Brayton cycle, with a recuperator, without intercool and without reheat. It has also been called the "simple" Brayton cycle, although this usually refers to the open cycle without a recuperator.

Conceptually, the closed Brayton cycle with heat recovery works as follows: the gas is compressed in a turbocompressor which decreases the volume and also results in an increase in temperature. The gas is then heated as much as possible in a recuperator that recovers available energy from the turbine exhaust. It is further heated by an external heat source to the highest temperature permitted by the heater materials. At this point, the gas has expanded and its volume flow rate has increased, since the heating has all occurred at constant pressure. The gas then expands through the turbine, doing more work than it took to compress it in the compressor. Next, the gas flows through the recuperator, exchanging its heat with the counterflowing compressed gas and dropping in temperature. Finally, the gas is cooled to the sink temperature and the cycle is repeated.

The working gas is usually helium, since compared to nitrogen its high thermal conductivity allows a fivefold reduction in the size and cost of the recuperator and almost halves the size and cost of the cooler and heater, although the turbine will be more expensive since the high velocity of sound in helium requires more turbine stages. For this reason, air, nitrogen, and mixtures of helium and xenon have also been used cost-effectively where specific work output is not critical, the increased heat-exchanger cost being offset by the reduced turbine cost.

The Micro-Tube Strip Heat Exchanger

Our new look at this unpopular cycle is motivated by the availability of an unconventional heat exchanger, known as the Micro-Tube Strip (MTS) heat exchanger. The MTS exchanger is the subject of another paper presented at this conference ("A Laminar-Flow Heat Exchanger", IECEC 900167), and will therefore be described here in outline only.

Heat exchanger design involves a trade-off between heat exchange, fluid friction losses, and exchanger cost. With respect to the first two criteria, it can be shown that very high effectiveness can be achieved at moderate levels of frictional loss by an exchanger consisting of a very large number of short, small-diameter, parallel flow paths. Designs of this sort are often found in ceramic gas-turbine regenerators and Stirling-engine regenerators, where a regenerator effectiveness of 95% is not unusual. Achieving this level of effectiveness in a gas-gas heat exchanger for acceptable flow losses and acceptable cost is more difficult.

From IECEC-90, Vol. 2

The MTS heat exchanger achieves these goals by taking advantage of the properties of modern alloys and automated manufacturing techniques. Mass-produced sub-millimeter-diameter stainless-steel tubes are finished to the required length, then automatically inserted into spacer forms. They are held rigidly by encapsulation in a low-melting-point fusible alloy, and die-cut tube strips are pressed onto each end of the array of tubes. Given the right degree of interference fit between the tubes and the die-cut holes in the tube strips (0.4 – 3 %), the assembly can then be heated to 1100 °C for a few minutes to effect diffusion welding between the tubes and header strips. We have assembled heat-exchanger modules containing over 100 tubes by this method, and found the welds to be 100% leak-tight – within 10^{-6} mm^3/s to hydrogen at one atmosphere.

A sketch of a single module is shown in Figure 1.

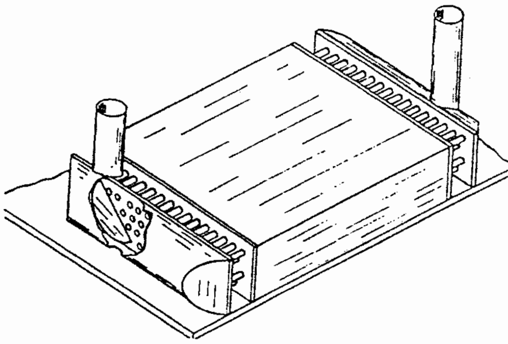

Figure 1

In a typical application, many modules would be manifolded together into banks (Figure 2.)

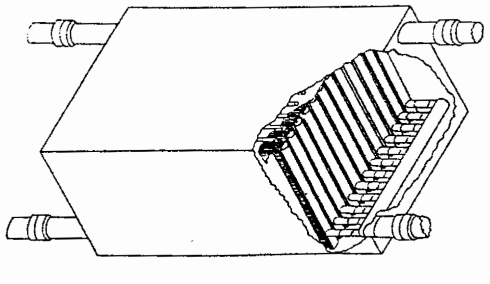

Figure 2

We have so far built and tested a three-module, 327-tube bank, and believe the current limit of technology is an 800-tube module.

In the companion paper, we calculate the dimensions that would be appropriate for an MTS heat exchanger designed to exchange 3 MW between helium streams flowing at 1 kg/s, for a total frictional loss of 30 kW. This corresponds to an MTS design employing 200,000 tubes of length 0.3 m and internal diameter 0.5 mm. This is a theoretically based calculation; practical considerations, including the losses associated with manifolding and baffling, would certainly compromise this performance. However, we believe the quoted figures are of the correct order, and that a heat exchanger of these dimensions can be manufactured with currently available technology at a reasonable price.

The heater and cooler must exchange heat with the environmental air. Because of the acknowledged sensitivity of the MTS heat exchangers to fouling, high efficiency in filtration is of crucial importance, both for cooling and for combustion. While air filtration is a very mature and well developed technology, the stringent requirements of the MTS exchangers warrant a careful evaluation of the available technology and the development of some innovative designs.

Advantages of the Closed Brayton Cycle

If the MTS heat exchanger can reduce the cost penalty associated with the closed cycle, does that cycle have any distinctive advantages to recommend it?

The most important advantage of any closed cycle is the flexibility it affords in establishing non-polluting combustion conditions with little adverse affect on efficiency. Clean fuels — especially alcohol — can be used to virtually eliminate NO_x, CO, and unburned hydrocarbon air pollution.

A second advantage of the closed Brayton cycle is its ability to achieve high efficiencies over a wide range of power demand by adjusting the gas pressure, fuel burning rate, and generator control parameters[3]. The typical demand conditions for utilities between midnight and 5:00 AM is about one fourth of peak demand. Efficiency of the closed Brayton cycle will generally be maximum at about half power and will drop to about 96% of maximum at full power and at quarter power. The efficiency of open Brayton cycles and closed Rankine cycles, on the other hand, decreases rapidly under off-design load conditions. Their efficiency is usually a maximum at full power, and it typically drops to about 80% of maximum at half power and 50% at quarter power[4].

A further advantage of the CBC is that it separates the hot corrosion resistance and hot strength requirements: hot corrosion resistance is required only in the heater, and hot specific strength only in the turbine. Uncoated high-strength superalloys are well-suited for high-efficiency turbine designs, and the new highly corrosion-resistant ductile superalloys are adequate in the heater. There have been intensive efforts over the past two decades to develop high-performance ceramics such as transformation-toughened zirconia and hot-pressed silicon nitride/carbide. However, these ceramics are still two to five times as expensive as most superalloys in finished products, and are therefore not expected to see wide use in the near future.

An engine designed around the LPR CBC also has the advantage of simplicity. High-performance heat exchangers allow high system efficiency without the complications of combined cycles, turbine-blade cooling, turbine interstage reheat, compressor interstage cooling and complex high-compression turbine designs. The system is only slightly more complex than a modern internal-combustion engine, allowing it to be scaled down below 100 kW (135 HP) without major sacrifices in performance.

Finally, the high-working-pressure LPR CBC results in more efficient utilization of turbine materials, for two fundamental reasons. First, the ratio of the radial to tangential turbine blade forces is greatly reduced. The specific power of the turbine is greatly increased, even though the pressure ratio has been reduced, owing to the high pressure and the high velocity of sound in helium. Second, the subsonic flow conditions simplify airfoil design for high polytropic efficiency in the turbines. State-of-the-art polytropic efficiencies for compressors and turbines are 83% and 90% respectively at where ρ is the gas density (kg/m^3) and μ is the dynamic viscosity (kg/ms). $\dot{W}_p$ can be reduced by increasing n, the number of tubes in the heat exchanger, or d_i, the tube internal diameter; both these strategies lead to an undesirable increase in heat-exchanger mass. Reducing $\dot{W}_p$ by decreasing L extracts a penalty in increased conduction losses along the heat exchanger. The only remaining option is to increase ρ by increasing the mean pressure. This requires the mass, and hence cost, of the heater to increase to meet creep stress requirements. Sample calculations suggest that the optimum value of mean cycle pressure lies in the range of 0.4-6 MPa.

While the heat exchangers easily scale to any size from 1 kW to 100 MW, the turbine expander is difficult to scale below 200 kW with very high efficiency, owing to the high mean pressure. Partial-admittance turbines can be scaled smaller, but they perform poorly. Highest efficiency is obtained for full-admittance, 50% reaction, axial-flow, multistage expanders. For such expanders operating at low pressure ratios with inlet pressure p_2, it can be shown that the ideal shaft power P_E is given by

$$P_E \approx A p_2 (r-1)^{1.5} \sqrt{RT/M} \qquad (13)$$

where A is the frontal area of the nozzles. For helium gas at 2 MPa, 1250 K, this is

$$P_E \approx 3.2 \times 10^9 A (r-1)^{1.5} \qquad [\mathrm{W/m^2}] \qquad (14)$$

Thus at a pressure ratio of 2, a 2-MPa, 300-kW helium expander (for a 200-kW CBC system) requires an inlet area of about 100 mm^2. The high velocity of sound (2000 m/s) in helium at these temperatures requires about 10 stages if uncooled superalloy blades are used. While such designs have been proposed in the 100-MW range, they are impractical for small systems. Expander and compressor efficiencies above 92% may be achieved in small systems with three-stage superalloy turbines if the sonic velocity can be reduced to 700 m/s. (Single-stage ceramic impulse turbines have demonstrated efficiencies above 80% at shaft power below 5 kW with sonic velocities near 700 m/s.) Changing the working gas from helium to argon will reduce the sonic velocity by a factor of 3.1, but the thermal conductivity drops by a factor of 8.6. This would require an increase in recuperator mass by the same factor and a doubling of heater and cooler masses.

A notable improvement in system performance per cost can be achieved by using a gas mixture rather than a pure gas, since the sonic velocity is determined by bulk properties – $(\gamma p/\rho)^{0.5}$ – while the thermal conductivity is primarily determined by the volume fractions of the constituents. Holland [8] gives the following expression for a reasonably accurate estimate of the thermal conductivity k of a mixture of gases far below the critical pressure:

$$\mu_m = \frac{\sum y_i k_i \sqrt{MW_i}}{\sum y_i \sqrt{M_i}} \qquad (15)$$

where y_i is the volume fraction of component i and M_i is its molecular weight. A mixture of 8% (by mass) helium and 92% argon gives a sonic velocity of 950 m/s at 1200 K, thermal conductivity being reduced by a factor of 2.7 compared with pure helium. If the alloys used in the system have sufficient carburization resistance, it is possible to use carbon dioxide in the mixture: replacing argon by carbon dioxide reduces the sonic velocity to 700 m/s and slightly increases the thermal conductivity. For aerospace applications, xenon may be used in place of argon, reducing the sonic velocity to 650 m/s. The conductivity of the helium/xenon mixture is a factor of 2.2 below that of pure helium. We have also considered mixtures containing small amounts of hydrogen; while these offer some reduction in viscosity, they do not help to reduce the ratio k/c^2.

The optimum gas mixture can be determined only when turbine and heat-exchanger costs are well-understood. We expect the turbine and compressor costs to approach $40/kW in high-volume production in the 20-100 kW range, for sonic velocities near 700 m/s.

In very small systems, optimum mixtures are expected to be close to those listed above, with total exchanger costs about $50/kW in lower-efficiency mobile applications and $100/kW in higher-efficiency stationary power plants. The cost of larger systems is dominated by the cost of the heat exchangers and it becomes practical to employ six or more turbine stages to allow smaller recuperators. The optimum mixture in CBCs above several megawatts is likely to comprise equal fractions of helium and argon, giving a mean molecular weight of about 8, a thermal conductivity about 20% below that of pure helium, and sonic velocity of 1500 m/s at 1200 K.

The CBC engine can greatly extend the load range over which high efficiency is obtained. Optimum turbine design is essentially independent of mean pressure. The design is primarily determined by the turbine materials, gas pressure ratio, and speed of sound, which depends only on temperature. Reducing the compressor-outlet pressure from 2 MPa to 0.2 MPa and reducing the combustion rate by an order of magnitude will reduce output power by an order of magnitude, reduce the T_δ's in the exchangers by an order of magnitude, increase relative conduction losses in the exchangers by an order of magnitude, and increase relative pumping power by two orders of magnitude. The net effect will be a reduction in efficiency of about 30%. (For comparison, a Rankine cycle will be down 30% in efficiency at 40% of design power.)

The maximum power P_S transmissible by a drive shaft of material with shear strength S (Pa) of radius r_s at spinning frequency f (Hz) is given by:

$$P_S \approx 2\pi S r_S^3 f \qquad (16)$$

This amounts to 100 kW for a 10-mm shaft at 1.4 kHz (surface speed 40 m/s) with a shear stress of 100 MPa. The high torque generated by a high-pressure turbine necessitates larger relative shaft diameter. We have considered using high-speed silicon nitride ball-bearings and accepting the necessity for oil-stream lubrication. However, the contamination of the gas and possible clogging of the microtubes by bearing lubricants would be unacceptable. It is therefore necessary to look at other options. Magnetic bearings have been considered, but their low pressure (0.1 MPa) makes them unsuitable for ultra-high-power-density applications.

Gas bearings are much better with respect to both stiffness (10^8 N/m on a 10-mm shaft) and pressure (2 MPa). Moreover, they eliminate lubricant contamination problems and allow higher-precision turbines since they are self-centering. The presence of a high-efficiency gas compressor in the system makes the choice of gas bearings attractive. The authors have considerable experience with nitrogen gas bearings at surface speeds up to 330 m/s at room temperature, and limited experience at temperatures up to 1100 K at lower

A short program has been written, based on Eq. 2, to calculate η_S as a function of system parameters and component efficiencies. Figure 4 shows system efficiency and specific mass for a range of values of recuperator effectiveness, and for high and low component efficiencies; curves A correspond to expander, compressor and generator efficiencies of 0.778, 0.756 and 0.886 respectively, while curves B correspond to the values of 0.9, 0.86 and 0.95 for the same parameters. Specific conductances of recuperator, heater, cooler and preheater for this figure are taken to be 150, 70, 40 and 100 W/kg.K respectively; our companion paper reports an experimentally measured specific conductance of 50 W/kg.K for a prototype MTS heat exchanger, and we believe these performances to be achievable in the short term.

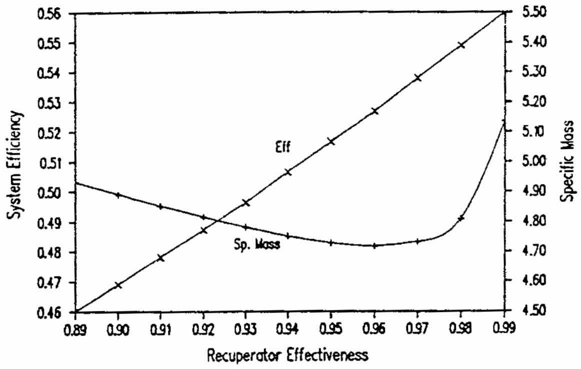

Figure 5

Figure 5 shows an optimization curve based on a longer-term extrapolation of currently available technologies. The assumed figures for expander, compressor and generator efficiencies are 0.95, 0.92 and 0.95 respectively, while the assumed values for specific conductance in recuperator, heater, cooler and pre-heater are 800, 70, 60 and 140 W/kg.K respectively.

Specific Power

An important parameter in the open Brayton cycles used for jet propulsion is the ratio of output power to gas flow. Hence, it is customary to define specific power as the dimensionless ratio $P_0/(\dot{m}c_pT_1)$, where P_0 is the power output, $\dot{m}$ the mass flow rate, c_p the gas specific heat and T_1 the maximum cycle temperature. In the CBC, this parameter is of no consequence. An extremely significant parameter, however, is the specific power P' as frequently defined for the Otto and Diesel internal combustion engines:

$$P' = P_0/m_S \tag{5}$$

where m_S is the total system mass. For the CBC,

$$m_S = m_C + m_H + m_P + m_R + m_M + m_E + m_T \tag{6}$$

where the above subscripts refer to cooler, heater, preheater, recuperator, miscellaneous, electric generator, and turbine masses respectively. They are listed in order of decreasing size.

Preliminary results from computer modeling indicate that the mass of the MTS bank manifolding and cages will be comparable to the mass of the microtubes. The mass of the high-pressure containment vessel for the recuperator will be comparable to the total mass of the enclosed MTS banks.

The number of microtubes for the cooler and heater may be calculated from the equations presented in the companion paper. The equation for the heater may also be applied to the recuperator, and the number of tubes in the preheater, n_p, may be calculated from

$$n_p \approx 2.0\frac{P_0}{LT_\delta} \tag{7}$$

where the coefficient, 2.0, depends to some extent on the choice of fuel.

A 10,000-hour design life with standard safety margins (corrosion allowances plus creep-limited stress at high temperatures, or 0.7 yield stress at low temperatures) for $p_2 =$ 50 MPa requires the following exchanger masses, including bank manifolding, (short) main manifolds, thermal insulation, recuperator containment vessel, and heater combustor.

$$m_C \approx 1.2nd^2L\rho \approx 3d^2\rho P_0/T_\delta \quad [\mathrm{m.K/W}]$$
$$m_H \approx 3.0nd^2L\rho \approx 6d^2\rho P_0/T_\delta \quad [\mathrm{m.K/W}]$$
$$m_P \approx 1.0nd^2L\rho \approx 2d^2\rho P_0/T_\delta \quad [\mathrm{m.K/W}]$$
$$m_R \approx 3.0nd^2L\rho \approx 6d^2\rho P_0/((1-\eta_X)(T_3-T_2)) \quad [\mathrm{m.K/W}] \tag{8}$$

(T_δ in the recuperator has been expressed in terms of its effectiveness.)

The above expressions are believed to be accurate within about 25% for systems between 20 kW and 100 MW. For illustration, we substitute the following values, typical for a high efficiency (55%) design:

Metal density: 9000 kg/m^3 for the high-tungsten heater alloy and 8000 kg/m^3 for the other alloys;

Microtube I.D.: 0.8 mm for the heater, 0.6 mm for the preheater, and 0.5 mm for the recuperator and cooler;

T_δ: 10 K for the cooler, 20 K for the preheater, and 75 K for the heater; 0.97 for the recuperator effectiveness, with $(T_3 - T_2) = 550$ K.

Then,

$$m_C \approx 1.3P_0 \quad [\mathrm{kg/kW}] \tag{8}$$
$$m_H \approx 0.4P_0 \quad [\mathrm{kg/kW}] \tag{9}$$
$$m_P \approx 0.3P_0 \quad [\mathrm{kg/kW}] \tag{10}$$
$$m_C \approx 0.25P_0 \quad [\mathrm{kg/kW}] \tag{11}$$

For applications where specific power is more important than efficiency, the exchanger masses can be reduced significantly by allowing a larger T_δ. If m_C and m_P are reduced by a factor of four and m_H and m_R by a factor of two, it is still possible to achieve a system efficiency of 45%.

The importance of using a gas with low molecular weight, M, follows from the fact that thermal conductivity is approximately proportional to $1/(\gamma M)$.

The selection of the optimum mean cycle pressure involves a trade-off between pumping losses and creep stress limits. Pumping loss $\dot{W}_p$ in any one of the heat exchangers is given by

$$\dot{W}_p = \frac{128\mu L}{n\pi d_i^4}\left(\frac{\dot{m}}{\rho}\right)^2 \tag{12}$$

a compression ratio of 13 (as in a typical open cycle). At a compression ratio of 2, these respective efficiencies are 94% and 94.5%.

CBC Design

We now derive analytic expressions for the efficiency and power density of the CBC as a function of the fundamental system parameters. We also consider ease of manufacture.

Efficiency

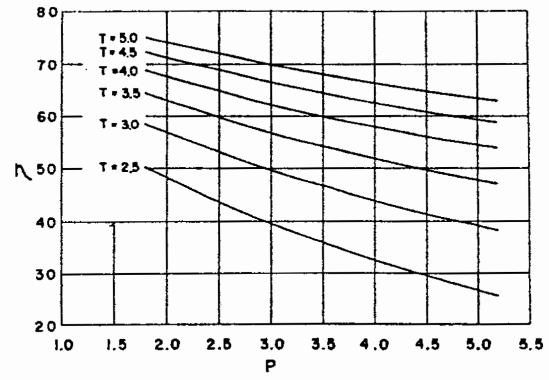

Figure 3

Fig. 3 shows theoretical efficiency curves for the ideal closed Brayton cycle, using helium as a working fluid, with perfect heat exchange for several temperature ratios ($\tau = T_4/T_1 = T_H/T_C$) as a function of pressure ratio ($p_r = p_2/p_1'$)[5]. At very low pressure ratio, the efficiency approaches the Carnot limit. The easiest conceptual explanation of the advantage of low pressure ratio (LPR) in the closed regenerative Brayton Cycle is that it increases turbine exhaust temperature and decreases compressor outlet temperature. This is obviously the opposite of what is desired without heat recovery, but with heat recovery it means that the average temperature of heat injection is higher and the average temperature of heat rejection is lower. Hence, the theoretical efficiency is increased. (This is not a novel concept. It has been well understood for decades [5,6].) Very high efficiencies can be obtained at moderate temperature ratios by using a low pressure ratio. Reducing the pressure ratio tends to reduce the power density of the engine, but this can be compensated for by increasing the mean cycle pressure. Such a design has not been optimum in the past because the heat-transfer requirements in the recuperator are increased at least five-fold and the pressure requirements in the heater are similarly increased. Using MTS heat exchangers for the recuperator and heater makes it possible to meet these requirements at reasonable cost.

The efficiency η of the ideal cycle (isentropic compression and expansion, 100% recuperator effectiveness, zero pressure loss in ducts and exchangers) is given by:

$$\eta = 1 - \frac{r^{(\gamma-1)/\gamma}}{\tau} \tag{1}$$

where r is the compressor pressure ratio, $\tau = T_4/T_1$, and $\gamma = c_p/c_v$.

Deriving an expression for the efficiency of a real cycle becomes very complicated if many effects are considered separately. The only way to approach the complete optimization problem is by computer modelling. However, it is valuable to look at major losses analytically and then discuss secondary effects.

Wilson derives a general expression for the efficiency of the real Brayton cycle that includes the following: compressor and expander polytropic efficiencies; leakage; total pressure loss; recuperator effectiveness; and mean specific heats for expansion, compression, and heat addition [7]. Adapting Wilson's results to the CBC (where mass flow through the compressor and expander are identical), assuming c_p to be independent of temperature (valid within 0.1% for He over the range of interest), and correcting a typographical error in his final equation (Eq. 3.2, p. 109) yields the following expression for the overall system efficiency η_S in generating electricity.

$$\eta_S = \frac{\eta_H \eta_e (E\tau - C)}{\tau(1-\eta_x(1-E)) - ((1+C)(1-\eta_x))} \tag{2}$$

where η_H is the heater efficiency, η_e is the generator efficiency, η_x is the recuperator effectiveness, $E = (T_4 - T_5)/T_4$ and $C = (T_2 - T_1)/T_1$. Assuming an ideal gas, we can write:

$$E = 1 - (r - \delta_r)^{(1-\gamma)\eta_e/\gamma} \tag{3}$$

$$C = r^{(\gamma-1)/\gamma\eta_C} - 1 \tag{4}$$

where r is the compression ratio, η_c is the compressor polytropic efficiency, η_e is the expander polytropic efficiency and δ_r is the normalized pressure drop in the ducts and recuperator.

Typical recuperator effectiveness will be 0.9 to 0.97; typical electrical generator efficiency will be 0.96; typical heater efficiency will be 0.85 to 0.95. Eq. 2 above ignores bearing friction losses and assumes that all ducting heat losses and viscous heating are properly considered in η_X. Polytropic efficiencies for small, low-pressure-ratio, atmospheric compressor and expander turbines are typically in the 0.85-0.90 range, while larger turbines can exceed 0.96. Efficiency is primarily a function of number of stages, precision, and Reynolds number (size). It is somewhat improved by increasing the mean pressure and reducing the pressure ratio.

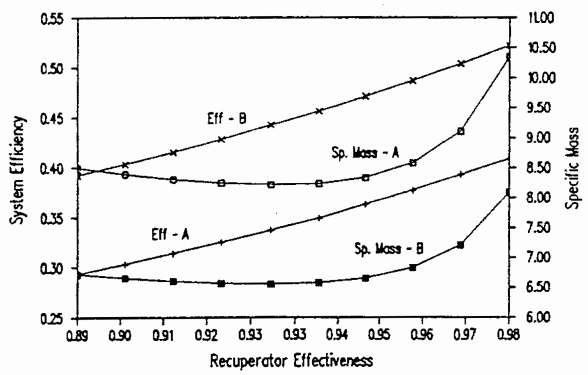

Figure 4

speeds. Total compressor power drain plus bearing friction will be about 0.02% of shaft power for high-stiffness bearings at surface speeds under 100 m/s. Some degree of bearing gas pressure control will be required to achieve stability over the wide range of mean cycle gas pressures needed to achieve flexible output power.

Fabrication

Ease of fabrication is a key factor in cost. Blade-cooling ducts are not necessary in the CBC owing to its lower turbine inlet temperature. This simplifies blade manufacture and eliminates the associated cooling losses. While 83% efficiency may be possible with a single row of blades, at least four rows of blades will be required to achieve 95% efficiency. Small turbines would be machined from solid disks for reduced cost, although this eliminates directionally solidified materials from consideration. The inert atmosphere and lower T_4 in the CBC allows considerable freedom in turbine materials selection. A number of alloys are available that will perform very well here, although it is likely that better alloys can be developed for these inert atmosphere conditions. The compressor will be made from a high-strength titanium-based alloy such as IMI 550.

High-Speed Generators

The mass and cost of the heat exchangers are expected to exceed those of the turbine and compressor. This implies the use of low-molecular-weight gases and small, high-speed turbines. Finding a generator to match these high rotational speeds is difficult. For turbines in the 10–1,000 kW range, we would like to operate at rotational speeds between 30,000 and 200,000 rpm, the higher speeds corresponding to the lower powers.

There have been several experimental generators which lie in the power and speed range of interest. One such is the Rice synchronous generator, a switched-reluctance machine which has been used with 25-kW CBCs for space power. Hyperconducting and superconducting induction generators have been used in the multi-megawatt range, and homopolar generators have been used for large DC welding supplies, railguns, and tokamak plasma heating.

We are currently exploring three technologies which we believe hold promise: inherently stable liquid-metal current collectors for homopolar generators; whisker-reinforced ferrites for high-speed permanent-magnet rotors; and efficient DC excitation for high-frequency variable-reluctance generators. Detailed comment on this work will be deferred to a later report.

Applications

Stationary Power Generation

One of the more attractive applications of the CBC engine is the static generation of electrical power. Existing non-hydro utility power-generating plants cost from \$800 to \$4,000 per kW of capacity, and most plants are in the 150 to 1,000 MW range, averaging \$1,500/kW, or about \$900 million per plant. At present about 250 plants are being designed or are under construction in the States, with construction times of 3 to 10 years. Utility power plant growth and replacement of obsolete plants is expected to continue to average more than 10 GW per year during the next 25 years. This represents a \$15 billion annual market at current prices.

The advantage of the CBC engine in this application is its combination of low capital costs with high efficiency, and hence low running costs. It also has the advantage that it can combine high efficiencies with clean operation. While combined-cycle power plants have been built with efficiencies of about 46%, these plants cost around \$4,000/kW, and their high operating temperatures lead to NO_x emissions which are far above environmental standards, necessitating the addition of costly catalytic converters.

Nuclear Power Conversion

Although nuclear energy has been quite unpopular since the Three Mile Island accident in March of 1979, nuclear energy research continues to be funded, especially for very long range tokamak fusion concepts and for inherently-safe High-Temperature-Gas-Cooled (HTGC) fast spectrum reactors. The MTS heat exchangers and LPR CBC are an ideal match for HTGC reactors, and MTS surface exchangers provide a viable solution to tokamak first-wall cooling problems.

Conclusion

Advances in heat-exchanger technology justify a re-examination of the closed Brayton cycle. Guidelines for engine design have been developed, and the areas requiring further research have been identified. Some of the more promising applications have been reviewed.

Nomenclature

A	Frontal area of nozzles (m^2)
C	Temperature ratio $(T_2 - T_1)/T_1$
c_p	Specific heat of gas at constant pressure(J/kg.K)
c_v	Specific heat of gas at constant volume(J/kg.K)
d_i	Heat-exchanger tube inner diameter (m)
E	Temperature ratio $(T_4 - T_5)/T_4$
f	Spinning frequency (Hz)
k	Conductivity (W/m.K)
L	Length of tubes in heat exchanger (m)
m_C	Mass of cold heat exchanger (kg)
m_E	Mass of electric generator (kg)
m_H	Mass of hot heat exchanger (kg)
m_M	Mass of miscellaneous system elements (kg)
m_P	Mass of the pre-heater (kg)
m_R	Mass of the recuperator(kg)
m_S	Mass of the CBC engine-plus-generator system (kg)
m_T	Mass of the CBC turbine (kg)
$\dot{m}$	Mass flow rate (kg/s)
M	Molecular weight (amu)
n	Number of microtubes in a heat exchanger
n_p	Number of microtubes in the preheater
p	Pressure (Pa)
P'	Specific system power (W/kg)
P_E	Ideal shaft power (W)
P_o	Output Power (W)
p_r	Pressure ratio (Dimensionless)
P_s	Actual shaft power (W)
p_2	Compressor outlet pressure (Pa)
r	Compressor pressure ratio
r_s	Shaft radius
S	Shear strength
T	Hoop stress (Pa)
T_C	Sink temperature (K)
T_H	Source temperature (K)

T_δ Mean temperature difference between entering hot gas stream and leaving cold gas stream
u Gas velocity (m/s)
v Circumferential speed (m/s)
y_i Mole fraction of a constituent of a mixture
γ Ratio of specific heats
δ_r Normalized pressure drop
η Efficiency of ideal cycle
η_c Compressor efficiency
η_e Electrical generator efficiency
η_H Heater efficiency
η_s System efficiency of real system
η_x Recuperator efficiency
μ Gas viscosity (kg/m.s)
ρ Density (kg/m^3)
τ Ratio of turbine inlet temperature to compressor inlet temperature

References

1 L.M. Lidsky *et al.*, "A Direct-Cycle Gas Turbine Power Plant for Near-Term Applications: MGR-GT", presented at the 10th International HTGR Conference, San Diego, CA September 19-20, 1988.

2 Z. P. Tilliette, J. Engin. Gas Tur. and Power 109, 92, 1987.

3 M. Berchtold and C. Keller, U. S. Patent #3,218,807, 1965.

4 M. R. Erbes et al, Science 237, 379 (1987).

5 H. Cohen, G. F. C. Rogers, H. I. H. Saravanamuttoo, Gas Turbine Theory, Longman, London, 1974.

6 A. H. Shapiro, Compressible Fluid Flow, Vol. I, Wiley, NY, 1953.

7 D. G. Wilson, The Design of High Efficiency Turbomachinery and Gas Turbines, MIT Press, Cambridge, Mass., 1984.

8 F. A. Holland *et al.*, Heat Transfer, Hunemann Educational Books, pp. 336, 339, London, 1970.