Regenerative Internal Combustion Engine Part I:Theory

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A zero-order thermodynamics analysis is presented for an engine cycle in which part of the thermal energy of the exhaust gases is extracted, stored, and released to the flammable mixture at the end of the compression stroke. The ideal efficiency rises by at least 50% of the value without regeneration. Regeneration benefits engines based on the Otto and atmospheric (no compression) cycles more than it does those based on the Diesel cycle. It is expected that the temperature after regeneration will be higher than the autoignition temperature, so that no igniter will be required, except for starting. It is recommended to control the power by varying the density at the intake, maintaining a constant equivalence ratio of the mixture.

Nomenclature

A = regenerator total area
A_0 = regenerator flow area
D = hydraulic diameter
f = friction coefficient
h = heat-transfer coefficient
k = C_p/C_v = gas constant
M = mass of regenerator
Nu = Nusselt number
ΔP = pressure loss across regenerator
P_i = cylinder pressure at point i
r = V_1/V_2 = compression ratio
Re = Reynolds number
T_i = temperature (K) at point i
V_i = cylinder volume at point i
δ = $\Delta T/T_i$ = relative heat-release factor
ϵ = regenerator effectiveness
η = thermodynamic efficiency
λ = reduced length of regenerator
μ = viscosity
π = reduced period
τ = T_1/T_3 = temperature ratio

Introduction

IN the average spark ignition, internal combustion engine, more energy goes to the exhaust gases (35%) than work is produced by the engine (25%). Several approaches have been followed to recover that energy in internal combustion engines[1]:

1) Exhaust afterexpansion in a turbine. If the exhaust gases are allowed to expand in an external turbine, about an extra 10% power can be added to the shaft. This power is usually employed in a supercharging compressor, which has a multiplying effect in the power delivered by the engine. The efficiency increases by as much as 4% of the initial value for diesel engines, but will remain about the same for turbocharged spark ignition engines, due to the need to lower the compression ratio to avoid knock.[2] Gearing the turbine to the output shaft—and not to a compressor only—has not been practical, in general, due to the wide difference of rotational speed between crankshaft and turbine (3,000 rpm vs 60,000 rpm, typically), which requires a complex and costly reducer.

2) Bottoming cycles. The hot exhaust gases can be used as a heat source in a steam or hot-air engine, which is adjacent to the main engine. Reductions of fuel consumption slightly over 10% have been reported.[2] Again, the problem is the additional complexity and cost of the system, which are difficult to justify for such a small gain.

3) Regeneration. Regeneration consists of the recovery of heat of the exhaust within the cycle itself. Heat is removed from the exhaust gases and supplied to the working gases at the end of the compression phase, before combustion takes place. This is the only phase when regeneration effects an improvement of the efficiency. Putting the heat back before compression, for instance, has no effect on the ideal cycle efficiency, because the minimum cycle temperature is raised proportionally to the maximum temperature. In a real cycle, the maximum temperature will not rise as high, due to enhanced heat-transfer losses, and the efficiency will drop with an increase of intake temperature.

Regeneration is an integral part of all automotive gas turbine engines and is responsible for an increase in fuel economy from the basic configuration in excess of 20%. Regeneration is relatively easy to achieve in gas turbines because the exhaust and compressed gases are separated in space, not it time. In the usual piston engine, this separation occurs in time, not in space. The second paper of this series[3] deals with different mechanisms that are proposed to achieve this effect. A common characteristic of these designs is a divided combustion chamber, whose only communication is through a porous regenerator. A similar arrangement is found in practical Stirling cycle machines.[4]

Figure 1 represents the temperature-entropy (T-S) diagram of a typical Otto cycle. The shaded areas are the additional work that could be obtained with an exhaust turbine geared to the engine (expansion to atmospheric pressure), or with a bottoming cycle (of efficiency 33%), and the heat saved with regeneration. The gain due to regeneration is substantially larger than that due to a geared turbine and even than that of a bottoming cycle.

Ideal Cycle Analysis

In this section, the ideal regenerative Otto, Diesel, and atmospheric (no compression) cycles are discussed. A general analysis was presented by Walker,[4] following the work of Rallis et al.,[5] to model real Stirling engines. The relations presented here can be considered as particular cases of that analysis. A similar analysis was carried out by Keating et al.,[8] also including cases of regeneration during compression and expansion. What follows is, of necessity, a very simplified view of how the real engine will perform. Important effects,

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Presented as Paper 88-3063 at the AIAA/ASME/SAE/ASEE 24th Joint Propulsion Conference, Boston, MA, July 11-13, 1988; received Aug. 2, 1988; revision received March 27, 1989.

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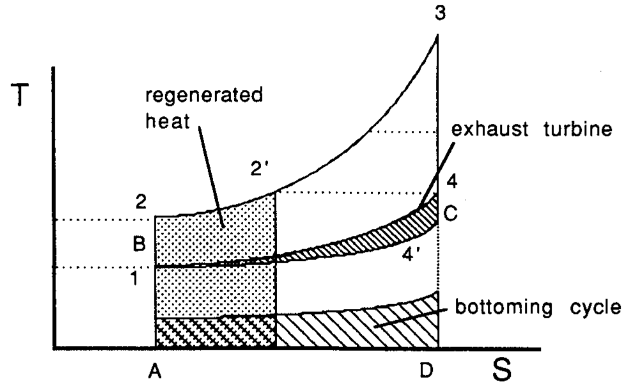

Fig. 1 Cycle comparison for proposed energy-saving solutions.

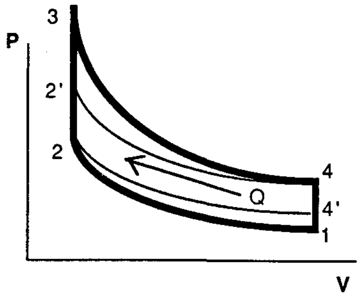

Fig. 2 Regeneration in the Otto cycle.

such as variable specific heats, finite reaction rates, heat transfer, and friction, have been entirely neglected. The virtue of the analysis presented below is to point out the effects of several important variables and to show the potential gains of regenerative engines over conventional engines.

Otto cycle

Figure 2 represents the typical pressure-volume diagram of an Otto cycle of compression ratio $r = V_1/V_2$. For the standard Otto cycle, from the ideal relations

$$T_2 = r^{k-1} T_1 \tag{1}$$

$$T_3 = T/\tau \tag{2}$$

$$T_4 = T_3/r^{k-1} \tag{3}$$

one can deduce the efficiency:

$$\eta_s = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - r^{1-k} \tag{4}$$

If there is a perfect regenerator, this becomes

$$\eta_r = 1 - \frac{(T_4 - T_1) - (T_4 - T_2)}{(T_3 - T_2) - (T_4 - T_2)} = 1 - \tau r^{k-1} \tag{5}$$

where $\tau = T_1/T_3$ represents the degree of heating. Observe that the efficiency increases as r increases in the ideal Otto cycle, whereas it decreases in the regenerative cycle. The effect of regenerative is strong enough to reverse the functional variation of the efficiency with the compression ratio.

If the regenerator is imperfect, having an effectiveness ϵ, then

$$\epsilon = \frac{T_{2'} - T_2}{T_4 - T_2} \tag{6}$$

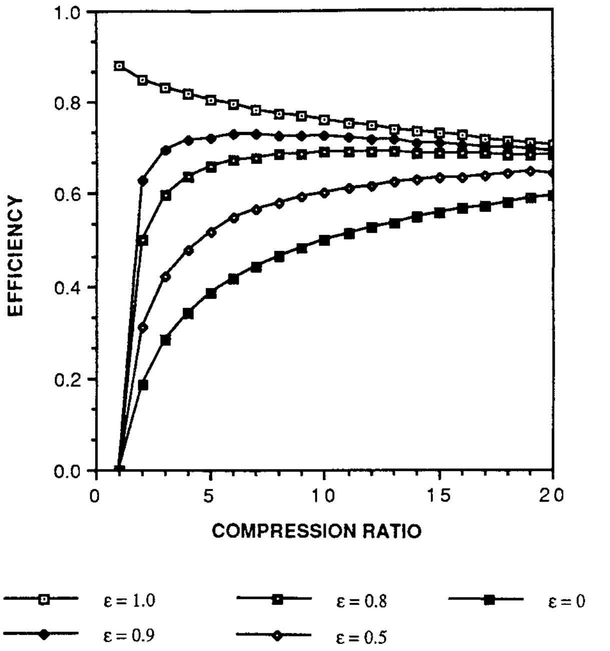

Fig. 3 Efficiency curves of the regenerative Otto cycle; constant $\tau = 0.12$.

And the efficiency is

$$\eta = 1 - \frac{(T_4 - T_1) - \epsilon(T_4 - T_2)}{(T_3 - T_2) - \epsilon(T_4 - T_2)} \tag{7}$$

which finally yields

$$\eta = \frac{(1 - r^{1-k}) - \tau(r^{k-1} - 1)}{(1 - \epsilon r^{1-k}) - \tau(1 - \epsilon) r^{k-1}} \tag{8}$$

Observe that Eq. (8) becomes Eq. (5) for $\epsilon = 1$ and Eq. (4) for $\epsilon = 0$ (no regeneration).

It is interesting to plot Eq. (8) as a function of r for several values of ϵ. A value of τ typical of spark ignition engines ($\tau = 300$ K/2500 K $= 0.12$) is used in these graphs (Fig. 3). Similar graphs are presented in Rallis et al.[5] for $\tau = 0.35$.

All of the constant ϵ lines intersect at the point $r = r_{max}$, where $r_{max} = \tau^{1/(2-2k)}$ is given by the condition $T_4 = T_2$, so that no regeneration is possible. For $\tau = 0.12$ and $k = 1.3$, $r_{max} = 34.2$, which is much higher than any practical value of r. The curves also intersect at the zeros of $\eta(r)$: $r = 1$ and $r = \tau^{1/(k-1)}$. The second intersection, that for $\tau = 0.12$, is $r = 1173$, which is far from any practical case. Regeneration increases the overall efficiency for all of the practical range of compression ratios found in current engines.

The mean effective pressure is not affected by the regeneration and is given the work parameter ξ:

$$\xi = \frac{\text{MEP}}{P_1} \tag{9a}$$

$$\xi = \frac{r}{(k-1)(r-1)} \left[\frac{1}{\tau} (1 - r^{1-k}) - (r^{k-1} - 1) \right] \tag{9b}$$

where MEP is the mean effective pressure.

The work parameter is a monotonically decreasing function of r, with maximum $\xi_{max} = (1/\tau - 1)$ at $r = 1$. If the regeneration is imperfect, the maximum efficiency is achieved at a certain compression ratio, r_{opt1}, which is obtained by derivating Eq. (8) and given by

$$r_{opt1}^{k-1} = \frac{1 - 2\epsilon - (1 - \tau)\sqrt{\epsilon(1 - \epsilon)/\tau}}{(1 - \epsilon)(1 + \tau) - 1} \tag{10}$$

For instance, if $\tau = 0.12$ and $\epsilon = 0.80$ (and $k = 1.3$), the maximum efficiency is obtained at $r_{opt1} = 11.5$ and has a value $\eta_{opt} = 0.689$, whereas the standard Otto cycle would have given $\eta_s = 0.520$. In this case, regeneration increases the efficiency by almost 17%, which is about a 33% improvement in relative fuel economy. One can see in Fig. 3 that the maximum of η as a function of r is very flat: large variations of compression ratio do not substantially change the efficiency near the maximum. This is very interesting for the design of a practical engine because a very good efficiency can be achieved regardless of the compression ratio. It may be possible to choose the compression ratio without worrying about deterioration of the efficiency.

When the maximum temperature is not limited, then the regeneration can produce a temperature increase over the standard cycle. If the temperature rise due to combustion, $\Delta T = T_3 - T_{2'}$, is maintained constant and we call $\delta = \Delta T/T_1$, then

$$\delta = T_3/T_1 - T_2/T_1 - \epsilon(T_4/T_1 - T_2/T_1) \tag{11}$$

and, using relations (1-3),

$$\delta = 1/\tau - r^{k-1} - \epsilon[r^{1-k}/(\tau - {}^{k-1}) \tag{12}$$

and finally,

$$\tau = \frac{1 - \epsilon r^{1-k}}{\delta + r^{k-1}(1-\epsilon)} \tag{13}$$

After substituting this into Eq. (8), the equation for the efficiency becomes

$$\eta = \frac{(1 - r^{1-k}) + \epsilon(2 - r^{k-1} - r^{1-k})/\delta}{1 - \epsilon r^{1-k}} \tag{14}$$

A normal value of δ for a stoichiometric mixture of octane is of the order of $\delta = 1900\ \text{K}/300\ \text{K} = 6.33$. For an equivalence ratio of 1.2 (rich) or 0.8 (lean), one get approximately $\delta = 6$. The plot of Fig. 4 has been generated using $\delta = 6$. The maximum efficiency is found at the compression ration $r = r_{opt2}$, given by

$$r_{opt2}^{k-1} = \epsilon + \sqrt{(1-\epsilon)(1 - \epsilon + \delta/\epsilon)} \tag{15}$$

For $\epsilon = 0.8$, $\delta = 6$, we get $r_{opt2} = 10.78$. For this compression ratio, $\eta = 0.7224$ and $\tau = 0.095$. As δ grows, the optimum compression ratio becomes larger. At partial loads, δ will be smaller (if no throttling is used), and the maximum efficiency occurs at a higher compression ratio than at full load. The variation of η with r is exceptionally flat near the maximum; it could be convenient to build the engine with a lower compression ratio than optimum ($r \approx 6$), in order to reduce the average pressure within the cylinder. A lower compression ratio corresponds also to a higher power density, according to Eq. (9).

The regenerative effect is lost when $T_4 = T_2$. This condition corresponds to a value $\delta = \delta_{min}$, given by

$$\delta_{min} = r^{k-1}(r^{k-1} - 1) \tag{16}$$

If the δ falls below this level, the regenerator absorbs heat from the compressed mixture and becomes detrimental to the performance of the engine. For $r = 10$, Eq. (16) gives $\delta_{min} = 2$, about one-third of the stoichiometric value.

Observe that the ideal efficiency can reach very high values at a low compression ratio. This is due to the fact that τ decreases as r decreases when δ is constant [Eq. (13)]; thus, T_3 can become very high at low r.

Diesel Cycle

Let us consider now a cycle in which the heat release occurs at constant pressure. For this case, the analysis of Rallis et al.,[5] following similar steps outlined earlier, yields

$$\eta = \frac{k(1/\tau - r^{k-1}) - (\tau^{-k} r^{k(k-1)} - 1)}{k(1/\tau - r^{k-1}) - \epsilon(\tau^{-k} r^{k(k-1)} - r^{k-1})} \tag{17}$$

and

$$\xi = \frac{r^k(1/\tau - r^{k-1}) - (\tau^{-k} r^{k(k-1)} - 1)}{(k-1)(r-1)} \tag{18}$$

The parameter ϵ is limited to $\epsilon_{max} = C_v/C_p = 1/k$ for a perfect regenerator because the heat is absorbed by the regenerator from the exhaust at constant volume but is released to the mixture at constant pressure. For example, for $\tau = 0.12$, $\epsilon = 0.8$, and $r = 11.5$ (optimum conditions for the regenerative

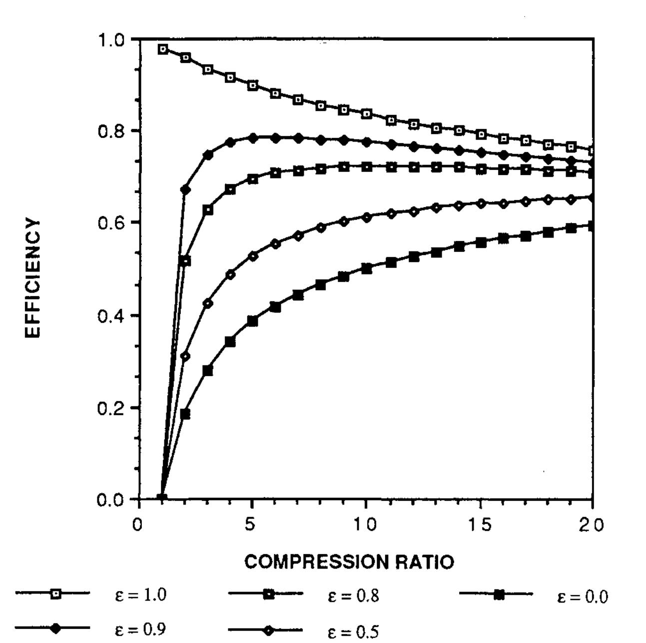

Fig. 4 Efficiency curves of the regenerative Otto cycle; constant $\delta = 6$.

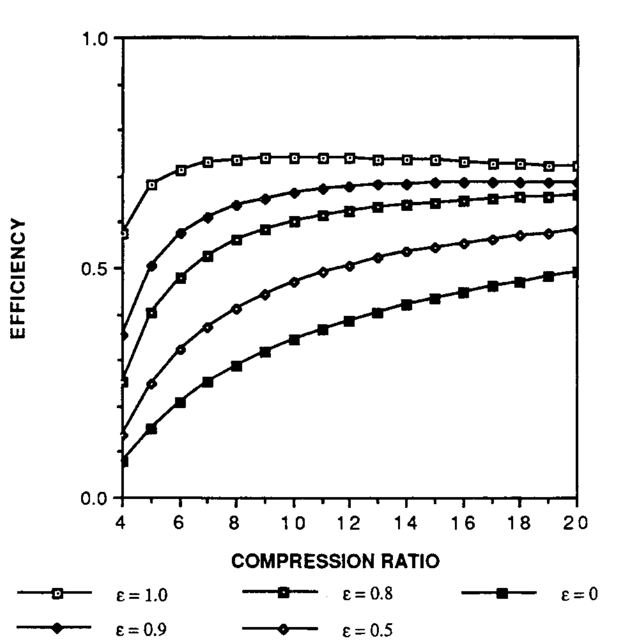

Fig. 5 Efficiency curves of the regenerative Diesel cycle; constant $\tau = 0.12$.

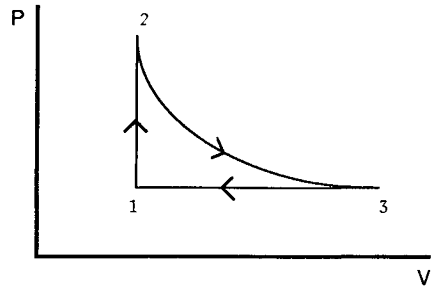

Fig. 6 Pressure-volume diagram of the atmospheric cycle.

Otto cycle), Eq. (17) gives $\eta = 0.6195$, which is a little smaller than the value $\eta = 0.689$ of the regenerative Otto cycle of the same parameters. In the regenerative Otto cycle, the high temperatures are maintained for a larger part of the cycle than in the regenerative Diesel, hence the higher efficiency.

The curves for $\tau = 0.12$, $k = 1.3$ have been plotted in Fig. 5. Regeneration seem advantageous for $r < r_{max} = \tau^{k/(1-k^2)}$ (given by $T_4 = T_2$). In our example, $r_{max} = 28.17$. The ideal case corresponds to $\epsilon = 1/k = 0.77$. The optimum efficiency occurs at a compression ratio r_{opt} given by the equation.

$$(r_{opt}^{k-1})^{k+1}(\epsilon - k - \epsilon k/\tau)\tau^k + r_{opt}^{k-1}(k+1)[\epsilon(k+1) - k] + k^2(1-\epsilon)/\tau + \epsilon k = 0 \tag{19}$$

which has to be solved numerically. For $\tau = 0.12$ and an 80% effective regenerator ($\epsilon = 0.8\epsilon_{max} = 0.615$), there is no real solution for r_{opt} in Eq. (12). There is no optimum compression ratio, except for very effective regenerators.

Atmospheric Cycle

The atmospheric cycle was used in the very first internal combustion engines by Lenoir and Otto and Langen[2] in the 1870's, but was eventually abandoned for more efficient compression cycle. In this cycle, represented in Fig. 6, the mixture is admitted and burned at constant volume, when the piston is at top dead center. Then the products are expanded down to atmospheric pressure and finally exploded during the return stroke. There are only three identifiable "end states," related by

$$T_2 = T_1/\tau \tag{20}$$

$$T_3 = T_2(P_3/P_2)^{(k-1)/k} = T_1/\tau(P_1/P_2)^{(k-1)/k} \tag{21}$$

but

$$P_1/P_2 = T_1/T_2 = \tau \tag{22}$$

Thus,

$$T_3 = T_1\tau^{(k-1)/k-1} = T_1\tau^{-1/k} \tag{23}$$

Hence, the efficiency of the atmospheric cycle without regeneration is

$$\eta_s = 1 - \frac{C_p(T_3 - T_1)}{C_v(T_2 - T_1)} = 1 - k\,\frac{\tau^{(k-1)/k} - \tau}{1-\tau} \tag{24}$$

In any case, the expansion ratio is

$$r = (P_2/P_1)^{1/k} = \tau^{-1/k} \tag{25}$$

The atmospheric cycle is interesting because of its simplicity (it does not need a compression stroke). Heat regeneration is particularly suitable to the atmospheric cycle. The use of regeneration produces a large increase of efficiency from that of the standard cycle, even for low maximum temperatures (high τ). When a perfect regenerator is added, the efficiency becomes

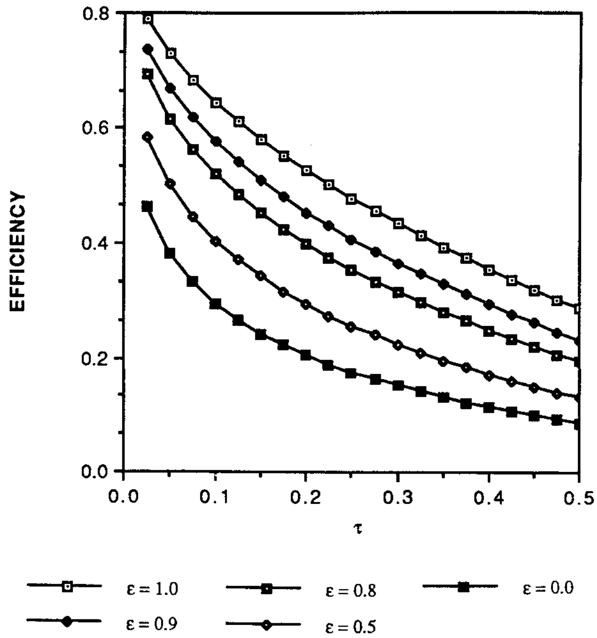

Fig. 7 Efficiency curves of the atmospheric cycle.

Table 1 Efficiency comparison of different cycles, with $\tau = 0.12$ and an 80% effective regenerator

Cycle type	r	Efficiency, %
Otto	11.5	52
Regenerative Otto	11.5	68.9
Diesel	11.5	37.6
Regenerative diesel	11.5	54
Atmospheric	6.4	27.2
Regenerative atmospheric	6.4	61.8

$$\eta_r = 1 - \frac{C_p(T_3 - T_1) - C_v(T_3 - T_1)}{C_v(T_2 - T_1) - C_v(T_3 - T_1)} \tag{26a}$$

$$\eta_r = 1 - (k-1)\,\frac{\tau^{(k-1)/k} - \tau}{1 - \tau^{(k-1)/k}} \tag{26b}$$

For $\tau = 0.12$, we obtain

$$\eta_s = 0.2716$$

$$\eta_r = 0.6177$$

$$r = 6.41$$

For a nonideal regenerator, Eqs. (20-23) give

$$\eta = \frac{1/\tau - 1 - k(\tau^{-1/k} - 1)}{1/\tau - 1 - \epsilon(\tau^{-1/k} - 1)} \tag{27}$$

of which Eqs. (24) and (26) are just particular cases, for $\epsilon = 0$ and $\epsilon = 1$, respectively. For $\tau = 0.12$ and $\epsilon = 0.8$, Eq. (27) gives $\eta = 0.4922$. The atmospheric cycle draws great advantage from regeneration and, consequently, is quite sensitive to the regenerator effectiveness. For a perfect regenerator, the efficiency of the atmospheric regenerative cycle lies between those of the

Otto and the regenerative Otto cycles of the same expansion ratio. Figure 7 represents the variation of η with τ and ϵ.

The atmospheric cycle has a power parameter of

$$\xi = \frac{(1/\tau - 1) - k(\tau^{-1/k} - 1)}{(k-1)(\tau^{-1/k} - 1)} \tag{28}$$

Table 1 compares the efficiencies obtained by the different cycle seen here for $\tau = 0.12$ ($k = 1.3$) and a regenerator that is 80% effective at the optimum compression ratio (for the Diesel cycle $\epsilon = 0.8/k = 0.615$; $\epsilon = 0.8$ for the other cycles). Otto cycle, followed by that of the regenerative atmospheric cycle. The smallest efficiency is that of the standard atmospheric cycle.

Combustion Characteristics

In the regenerative Otto cycle of the example ($\tau = 0.12$, $\epsilon = 0.8$), the temperature of the gases after compression (starting from $T_1 = 300$ K) is $T_2 = 624$ K. After regenerative heating, the temperature is

$$T_{2'} = T_2 + \epsilon(T_4 - T_2) = T_2 + \epsilon(T_3 r^{1-k} - T_2) = 1086 \text{ K}$$

This temperature is high enough to provoke the autoignition of the fuel for most of the fuels commonly used.[2] A spark plug will not be necessary, except for starting. The timing of the spark can be fixed to a few degrees after top dead center.

All of the fuel-air mixture passes through the regenerator and reaches the temperature $T_{2'}$ and is brought to autoignition conditions in a continuous way. The combustion will occur downstream of the regenerator as a time-delayed, stratified "explosion," rather than as a deflagration (using these terms as defined by Kuo[6]). In the schematic representation of Fig. 8 consider the mixture flowing out of the regenerator and burning. The regenerator heats up the mixture to autoignition conditions. The reaction starts downstream, after a finite induction time, and proceeds as in a one-dimensional plug-flow reactor. The temperature profile would resemble that of a premixed flame, with the difference that no upstream heat conduction is required to maintain the ignition of the mixture. The process would be very similar to the aftershock combustion of a detonation wave, according to the ZND model.[6] Also, the process would be stable at all flow velocities: a larger velocity would only stretch the temperature profile and decrease its slope, whereas a lower velocity would compress it. Since the fuel is subject to a controlled autoignition, the problems associated with it in spark ignition and diesel engines (knock) are not expected in this engine.

Catalytic effects have been successfully used in constant flow burners to extend the limits of flammability of the mixture.[7] It has been found that even a simple uncoated alumina matrix can have a significant catalytic effect, above 800 K. Exceedingly lean mixtures can be ignited by this means. This effect could be used to reduce the combustion temperature and hence the formation of oxides of nitrogen, NO_X, by feeding the engine with a mixture of high air-to-fuel ratio, or with a substantial amount of recirculated exhaust gases. A catalytic regenerator can also be used to reduce the concentration of NO_X, CO, and HC pollutants in the exhaust gases as they go through it, during the exhaust phase of the cycle.

An alternative to homogenous charge is fuel injection into the combustion chamber, past the regenerator. In this case, only air would be compressed and forced through the regenerator. The combustion will start when the air heated by the regenerator comes in contact with the fuel, which had been injected on the other side of the regenerator during the compression but was unable to burn in the absence of oxygen. A highly turbulent diffusion flame will be established within the combustion chamber as the air mixes and reacts with the fuel. The rate of heat release is expected to be slower than in the premixed case, which can be beneficial to the smoothness of the combustion and the inhibition of NO_X pollutants. Certain fuels (coal slurries, for instance) will have to be burned in this way to prevent severe clogging of the regenerator.

Power Control

To control the power of the engine for part-load operation, as we have seen, two main alternatives exist:

1) Reduce the density at the intake, by throttling, but maintaining a constant equivalence ratio.

2) Maintain the density at the intake (no throttling), but reduce the amount of fuel (variable equivalence ratio).

If the relative fraction of fuel in the mixture is varied, the parameter δ will vary accordingly, and so will τ, by Eq. (7). The benefit of regeneration will become smaller at part loads and will disappear altogether when $\tau = r^{2(1-k)}$ (in the regenerative Otto cycle). For the sample values being considered ($r = 11.5$, $\epsilon = 0.8$, $k = 1.3$), this condition corresponds to $\tau = 0.231$, which means $\delta = 2$, about one-third of the usual value $\delta = 6$. If the fuel heat release is reduced to less than one-third of the normal value, the regenerator would become detrimental to the efficiency, rather than beneficial. From this, it can be concluded that unthrottled power control will not be as effective as the use of a throttle, even if pumping losses are avoided. For best results, the fuel-to-air ratio should be maintained nearly constant. Other alternatives to throttling, such as variable boost or variable intake timing, can be used if throttling losses are to be reduced. Variable inlet timing

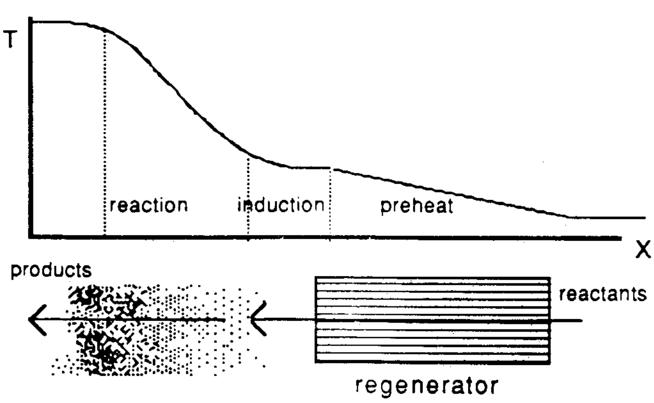

Fig. 8 Schematic of regenerator-ignited premixed combustion.

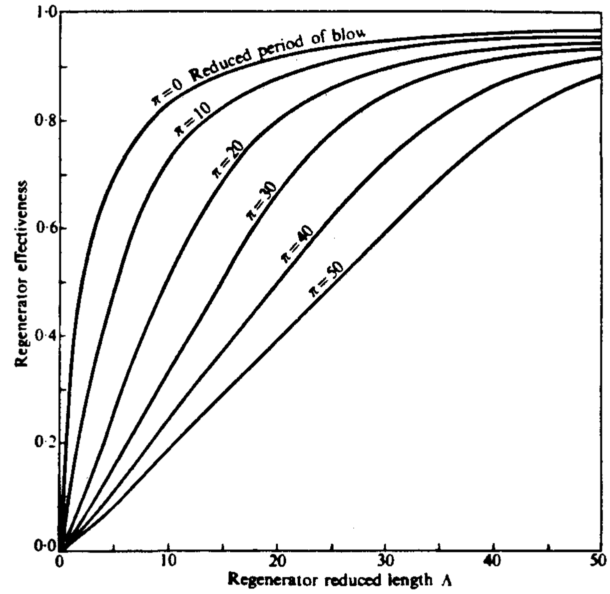

Fig. 9 Regenerator effectiveness vs λ and π (from Walker[4]).

should be easy to achieve with the engine configurations of the flat-piston family if the intake is controlled by a rotary valve.[3]

Pressure Losses in the Regenerator

The regenerator will tend to be bulkier than that of a Stirling engine. The main reason is the short time available for regeneration at the end of the compression stroke. The regenerator's effectiveness (and hence the efficiency that the engine can achieve) is controlled by the dimensionless parameters λ and π defined by

$$\lambda = hA/\dot{m}C \tag{29}$$

$$\pi = hAt/MC \tag{30}$$

The dependency of the effectiveness on λ and π is represented in Fig. 9 (from Ref. 4). For the short period of blowing of these regenerators, ϵ is almost a function of λ alone. The larger the λ, the larger the effectiveness. This parameter λ depends on the total surface area of the regenerator and on the heat transfer coefficient that can be achieved. The best characteristics are those of packed-wire matrix generators, the wire being as fine as possible, producing a very small average hydraulic diameter (0.1 mm or less). The problem is that the pressure loss across the regenerator can be very high unless the total flow cross section, A_0, is also large. Given a certain value of λ and of the time available for the flow, the following design equations are obtained:

$$\frac{A_0}{A} = \frac{Nu}{RePR}\frac{1}{\lambda} \tag{31}$$

$$\frac{\Delta P}{P_1} = \frac{1}{2} f \frac{A}{A_0} Re^2 \frac{\mu^2}{P_1 D^2 \rho_m} \tag{32}$$

where

$$Nu = \frac{hD}{k} \tag{33}$$

$$Re = \frac{\dot{m}D}{A_0 \mu} \tag{34}$$

and $f = f(Re)$ depends on the type of regenerator. For thin wire mesh in laminar flow, $f \propto Re^{-1/6}$.

It is convenient to size the regenerator for a low Reynolds number (around 2000), which gives, as a preliminary design, a "pancake" regenerator of roughly the same cross section as the cylinder. At its peak, the pressure loss is less than 10% of the compressed gas pressure.

Despite their simplicity, these equations provide some useful information. One can see in Eq. (32) that the pressure loss is proportional to the 11/6 power of the engine speed (through the Reynolds number). There will be a speed limitation of the engine, due to excessive pressure loss at the regenerator. The same limitation applies to the charge density. The performance of the engine will decay rapidly as the speed is raised above its design level.

Concluding Remarks

1) Regeneration of heat from the exhaust gases to the preignition gases could be an effective way to recover exhaust energy.

2) The ideal cycle efficiency rises significantly when regeneration is applied to the Otto cycle, a little less for the Diesel cycle, but more than doubles for the atmospheric cycle.

3) The fuel equivalence ratio should be maintained within fixed bounds in part-load operation, not due to combustion problems, but to maximize the benefit of regeneration.

4) Ignition can be achieved through the regeneration process, without a spark plug, except for starting. The combustion process should not be affected by knock problems, although it is likely to have its own special problems.

5) The design of the engine should pay particular attention to the pressure losses through the regenerator. For a given design, the flow restriction will impose an operational limit to the engine speed.

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